Adaptive-Passive Quarter-Wave Tube Resonator Silencer

Carl Q. Howard and Richard A. Craig

School of Mechanical Engineering, The University of Adelaide, Adelaide, S.A. Australia

ABSTRACT

An adaptive-passive quarter-wave tube resonator silencer was experimentally tested using recorded in-duct reciprocating engine exhaust noise. The exhaust noise was re-played through a loudspeaker attached to one end of a duct, and the adaptive passive system was able to track changes in engine speed and provide attenuation of the noise at the fundamental firing frequency of the engine and the third harmonic. The control system used a novel method of determining the phase angle of the transfer function between the pressure in the quarter wave tube and the main duct by using a sliding-Goertzel algorithm to calculate the Fourier coefficients.

INTRODUCTION

The two common types of adaptive-passive acoustic resonators are based on a quarter-wave tube and a Helmholtz resonator. These adaptive devices have the capability of altering their resonance frequency so that they can be tuned to attenuate tonal noise in an acoustic duct. One attractive feature of an adaptive-passive acoustic resonator device is that they do not inject acoustic power into the system (such as an active noise control system) to provide noise reduction. In addition, because they are passive devices, there is little potential for acoustic feedback or instability compared with an active noise control system.

This paper describes an adaptive quarter wave tube side-branch resonator that can be used to attenuate tonal noise over a range of frequencies. Experiments were conducted on a 3.5 litre, 6G74(M), V6 Mitsubishi petrol engine to:

1. Measure the sound pressure levels in the exhaust duct.
2. Manually tune a quarter wave tube to show that tones in the exhaust can be attenuated.
3. Recorded the exhaust noise to a computer.

The recorded exhaust noise was then replayed into a loudspeaker in another experimental rig that had an adaptive quarter wave tube attached to a duct, and was used to show that the adaptive system was capable of tuning and attenuating the noise at the engine firing frequency.

PREVIOUS WORK

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 and gas turbines. Neise and Koopman (1980) and Koopman and 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.

Active Helmholtz resonators, which include a loudspeaker, have been considered by a number of researchers (Birdsong and Radcliff 2001, Utsumi 2001, Yuan 2007). However these devices are not suitable for exposure to hot exhaust gases.

Lamancusa (1987) developed an adaptive Helmholtz resonator with a variable volume that was used to attenuate the noise at the firing frequency noise of a reciprocating engine. He implemented two configurations: a volume variable where the cavity length was varied by moving a piston using a lead screw and a DC motor, and second, in which an adjustable internal partitions altered the volume. The paper suggests that the resonators were automatically tuned with the engine speed, however details about the control system were not discussed.

de Bedout et al. (1997) developed an adaptive Helmholtz resonator that altered the volume of the Helmholtz resonator using a movable internal partition. This concept is similar to McLean (1998) that also used a moveable internal partition.

Kostek and Franchek (2000) presented an adaptive-passive Helmholtz resonator with a variable volume that was implemented using a piston. The tuning method that was implemented involved sweeping the piston over the entire length to determine the minimum sound pressure level and then the piston moved back to that location. This system is suitable for a laboratory environment. In a real system it would mean that any changes to the excitation frequency would require the system to conduct another "sweep" of the piston and would temporarily provide no acoustic attenuation, and hence not suitable for tracking changes in excitation frequency.

Some researchers have implemented adjustable Helmholtz resonators that have the capability to vary the geometry of the throat (Izumi et al., 1991, Nagaya et al. 2001, Esteve and Johnson 2004, Cheng et al. 1999, Kotsun et al. 2004, Ciray 2005).

There are few reports in the research literature of implementations of adaptive-passive quarter wave tubes. This paper presents some experimental results demonstrating their effectiveness.

BACKGROUND

A quarter wave tube 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 of air is given by (Bies and Hansen, 2009)

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

where γ=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 temperature in Kelvin.

These quarter wave tube resonators provide acoustic attenuation at odd multiples (i.e. 1x, 3x, 5x, …) of the fundamental (1x) resonance frequency, when they are attached to a main acoustic duct.

For a four-stroke reciprocating engine, the piston firing frequency occurs at half the crankshaft speed times the number of pistons in the engine, hence

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

For example, in the case of the V6 Mitsubishi engine under investigation here, the firing frequency at 1500rpm is 75Hz.

For an engine rotating at constant speed, the firing frequency will remain constant. However, the appropriate length for a quarter wave tube side-branch resonator will vary depending on the temperature of the exhaust gas, as this alters the speed of sound of the gas as indicated in Eq. (2). Therefore a fixed length quarter wave tube will only provide attenuation for a particular engine speed and exhaust gas temperature. An adaptive quarter-wave tube can be tuned to attenuate tonal noise for a range of engine speeds and exhaust gas temperatures.

ENGINE TESTING

This section describes the experimental apparatus used to measure the sound pressure level in the exhaust of 3.5 litre V6 Mitsubishi engine. Figure 1 shows a photograph of the experimental setup in the School of Mechanical Engineering engine test cell facilities. A quarter-wave tube side-branch resonator was attached to the exhaust pipe on the Mitsubishi engine after the catalytic converter and before the muffler.

The acoustic pressure in the exhaust duct was measured using water cooled PCB 106B pressure sensors. Figure 2 shows a close-up photograph of the custom-made water-cooled jackets for the pressure sensors. Thermocouples were installed along the length of the quarter-wave tube to measure the exhaust gas temperature.

Manual Tuning of Quarter Wave Tube

The Mitsubishi engine was operated at a constant speed of 1500rpm for 30 minutes to allow the exhaust gas temperature to stabilise at 507°C immediately after the catalytic converter. A plunger was inserted into the quarter wave tube and was manually positioned until the noise at the engine firing frequency (75Hz) was minimised, which resulted in approximately 15dB of attenuation, as shown in Figure 3.

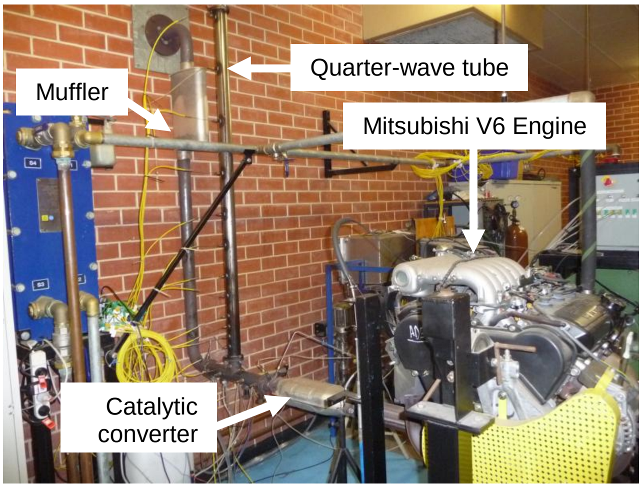


Figure 1: Photograph of the Mitsubishi engine and quarter wave tube.

Figure 2: Photograph showing the custom made water-cooled jackets for PCB microphones.

Also note that the third harmonic was attenuated, however the second harmonic was not, which is to be expected as quarter-wave tube resonators only provide attenuation at odd-multiples of their fundamental resonance frequency. The effective length of the quarter wave tube was 1.29m.

Figure 3 also includes a curve labelled "background", which was measured after the engine was switched off. This curve indicates the electrical noise floor of the instrumentation, which has an equivalent sound pressure level of around 80dB re 20uPa. The measured sound pressure levels when the engine was operating was greater than the electrical noise floor, and less than the maximum range of the pressure sensor which is around 190dB re 20uPa.

The speed of sound at 507°C can be estimated using Eq. (2) as 560m/s. Hence a prediction of a suitable quarter wave tube using Eq. (1) would have been

$$L = c/(4 f_{QWT}) = 560/(4 \times 75) = 1.87\text{m} \quad (4)$$

whereas the actual length was 1.29m. The discrepancy occurs because there is a significant temperature gradient in the quarter-wave tube and hence the speed of sound varies along the duct. The gas temperature at the piston in the quarter-wave tube was approximately 30°C. By using the known length of the quarter-wave tube, the "average" gas temperature equates to 95°C.

Having proved that a quarter-wave tube can attenuate exhaust noise at the engine firing (1500rpm), further measurements were made to record the exhaust noise, for subsequent play-back into another test rig with an adaptive quarter-wave tube, at several engine speeds as described in the following section.

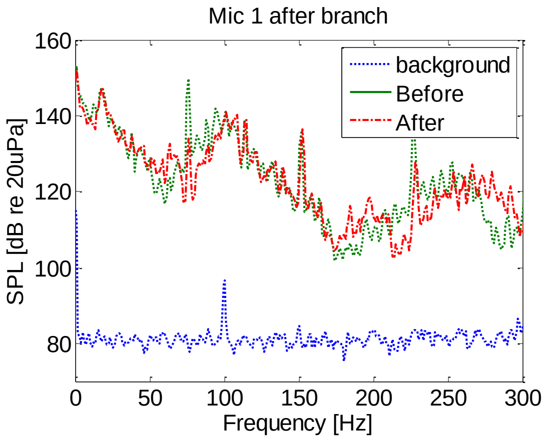


Figure 3: Sound Pressure Level at the microphone after the side branch.

Recording of Exhaust Noise

The side-branch section was replaced with a straight section of pipe that had two water-cooled pressure sensors, as shown in Figure 4. The Mitsubishi engine was operated at several engine speeds and the sound in the duct was recorded to a computer using Labview SignalExpress software and a National Instruments NI 9234 analog-to-digital converter. A magnetic tachometer sensor was installed on the engine crankshaft ring gear and the signal from this sensor was recorded at the same time as the pressure signals.

These recorded signals of the exhaust noise and the synchronised tachometer were then replayed into the experimental rig shown in Figure 5, to demonstrate the effectiveness of an adaptive quarter wave tube in attenuating tones in engine exhaust noise, and is described in the following section.

ACOUSTIC DUCT AND ADAPTIVE QUARTER WAVE TUBE

Apparatus

A steel duct with an internal diameter of 155mm, wall thickness of 5mm and length of 3m was used to simulate an exhaust duct. A loudspeaker at one end of the duct was used to provide an acoustic noise source. Plastic pressure piping was used to form a quarter wave tube that was attached to the main duct. A pneumatic piston head was attached to the end of a rod and formed part of the adaptive quarter wave tube. The position of the piston head could be adjusted using a linear actuator (Linear Actuators Australia, Square Linear Actuator 762mm stroke 50kg force). A custom-made motor controller was constructed to drive the linear actuator. Calibrated microphones were attached to the duct to measure the sound pressure levels throughout the system: a microphone was mounted in the face of the piston, two microphones at fixed positions were attached to the main duct, and a microphone that could traverse the length of the main duct, as shown in Figure 5.

Figure 6 shows the instrumentation used in the adaptive control experiments. The acoustic duct has three microphones that are connected to an instrumentation amplifier.

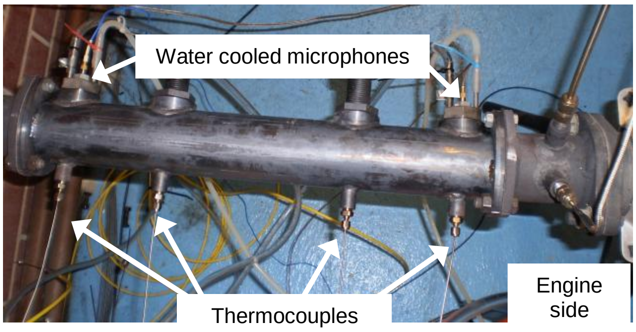


Figure 4: Photograph of the straight-through pipe section used on the Mitsubishi engine tests with the water-cooled PCB pressure sensors and thermocouples to measure the temperature of the exhaust gas.

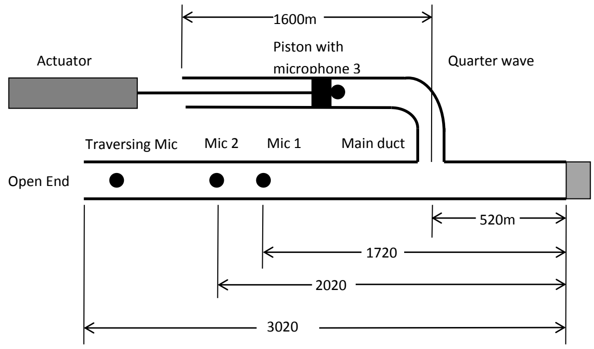


Figure 5: Schematic diagram of the adaptive quarter wave tube and acoustic duct.

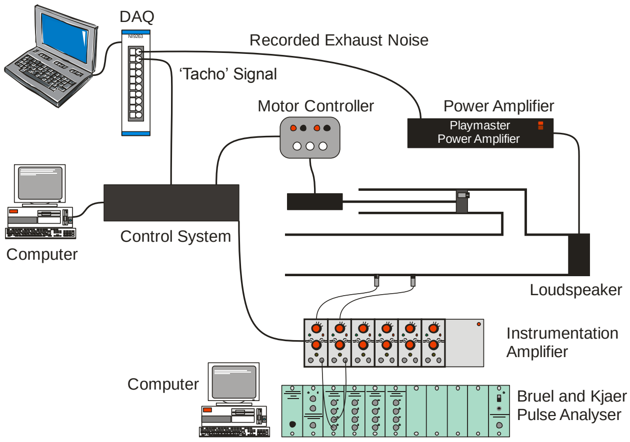


Figure 6: Instrumentation used in the adaptive control experiments.

These microphone signals are connected to both the Bruel and Kjaer Pulse spectrum analyser, which provides and independent measurement of the noise attenuation, and to a DSpace 1104 controller. The DSpace 1104 has 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 which moves a piston in the quarter wave tube. In addition a digital tachometer signal from the crankshaft was recorded that had 110 pulses per crankshaft revolution. The recorded exhaust noise and tachometer signal could be replayed using a digital-to-analog converter (National Instruments NI 9263) and replayed through a loudspeaker attached to the acoustic duct, and the tachometer signal was supplied to the DSpace 1104 control system.

Control Algorithms

The control algorithms used in this system comprises two parts: (1) a cost-function evaluation, and (2) an adaptive algorithm.

The cost-function evaluation for this system was based on the phase angle of the frequency response function between a microphone in the face of the piston and a microphone installed in the main duct, downstream from the side-branch labelled Mic 2 in Figure 5. The phase angle was determined using a novel method that implemented a sliding-Goertzel algorithm (Chicaro 1996, Garcia-Retegui 2007) , as described in Howard (2011).

The adaptive algorithm that was implemented adjusted the position of the piston in the quarter wave tube until the phase angle, described in the previous paragraph, was -90 degrees.

For the system under investigation here, there are three phase states that can be defined where the phase angle is:

- close to 0 degrees = State 1,
- close to -180 degrees = State 3, and
- in the region of the (near) step change between 0 to -180 degrees = State 2.

When the phase angle was in states 1 or 3, the linear actuator was driven at its maximum speed (1.6 cm/s). When the phase angle was between 70 and 110 degrees, it was in state 2, and was driven at a slower speed (0.2 cm/s).

Results

The acoustic pressure and tachometer signals were recorded on the Mitsubishi engine for two cases, where the engine speed was constant at 2000rpm, and when the engine speed was step changed from 2100rpm to 1900rpm. These recordings were replayed through the loudspeaker in the test rig, although the same sound pressure level amplitude (150dB) could not be achieved. Replaying the recorded engine exhaust noise enabled comparison of the acoustic attenuation by the adaptive quarter wave tube when adaptive control was implemented and when the adaptive control was switched off and the piston was set at one position. Comparison of the results from these two test will indicate the attenuation that can be achieved using an adaptive quarter wave tube.

For the first case, the engine speed was constant at 2000rpm (100Hz). The piston in the quarter wave tube was initially fully retracted. Figure 7 shows the sound pressure level at 100Hz over time at the three microphones positions shown in Figure 5. The upper graph in Figure 7 shows the results for adaptation. Initially the adaptation was switched off and then after 10 seconds the adaptation was turned on and the quarter wave tube tuned itself to attenuate the noise by 25dB at the fundamental firing frequency about 40 seconds later. The slow speed of the linear actuator limits the speed of convergence. The lower graph in Figure 7 shows a repeat of the test when the piston was fully retracted and the adaptation was switched off. Comparison of the two results shows that the adaptive system provides significant acoustic attenuation.

Figure 8 shows the power spectral density of the sound pressure level at Microphone 1 when the piston was fully retracted and adaptation was turned off (labelled "No Control"), and when the adaptive control system was turned on (labelled "With Control").

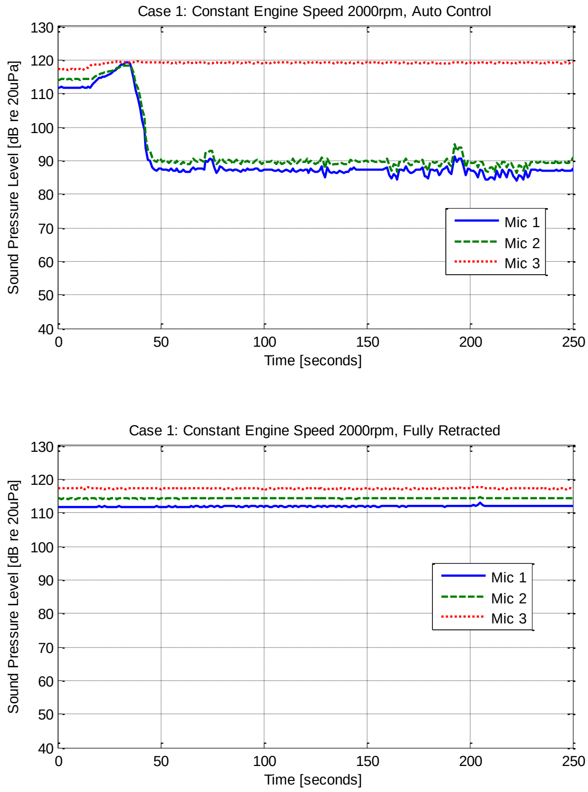


Figure 7: Sound pressure level in the duct when the engine speed was constant at 2000rpm. Top graph shows when adaptive control was implemented, bottom graph shows when the piston was fully retracted.

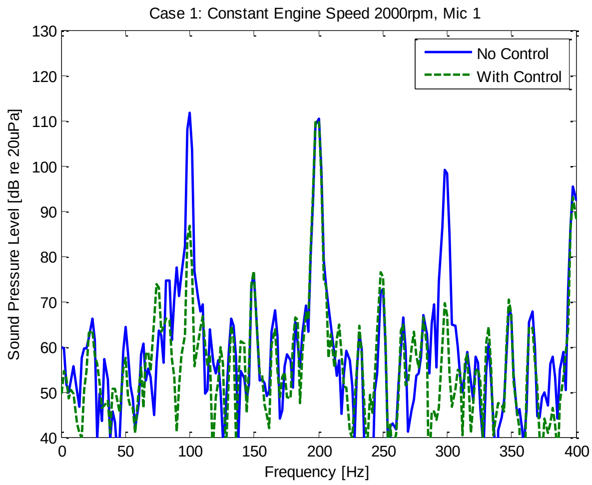


Figure 8: Sound pressure level spectrum at Mic 1 when the engine speed was constant at 2000rpm, when the piston was fully retracted (No Control) and with adaptive control (With Control).

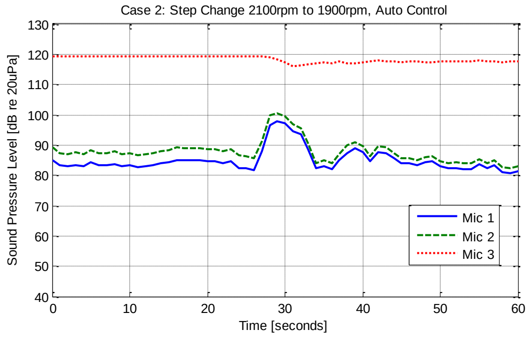
The results show that the sound pressure level at the 1x and 3x harmonics of the fundamental firing frequency (100Hz) were attenuated by the adaptive quarter wave tube, which is to be expected as a quarter wave tube only provides attenuation at odd-multiples of its fundamental resonance frequency.

For the second case, the speed of the Mitsubishi engine was step changed from 2100rpm (105Hz) to 1900rpm (95Hz). Figure 9 shows the variation over time of the sound pressure level, integrated over the frequency range from 94Hz to 108Hz for the three microphones positions shown in Figure

5. The test was conducted by initially tuning the adaptive quarter wave tube such that it was tuned to attenuate the exhaust noise at 105Hz. The engine speed remained constant for about 25 seconds and was then step changed to 1900rpm (95Hz). The upper graph in Figure 9 shows the change in the sound pressure level at the three microphones when the adaptive control was turned on. The lower graph shows the results when the experiment was repeated with the adaptive control turned off and the piston was fully retracted. The results show that the adaptive quarter wave tube provided significant attenuation at the fundamental engine firing frequency, and that the adaptive control system is capable of tracking a step change in the engine speed. After the step change in engine speed, the adaptive control system re-tuned to the correct speed within about 10 seconds. The speed of convergence of the adaptive control system is currently limited by the slow speed of the linear actuator.

Figure 10 shows the spectrum of the sound pressure level at the initial engine speed of 2100rpm (105Hz) for the case when the adaptive controller was turned off and the piston was fully retracted (labelled "No Control") and when the adaptive controller was turned on (labelled "With Control"). It can be seen that the quarter wave tube attenuates the fundamental and 3x harmonic. There was a 25dB reduction at the engine firing frequency.

Similarly Figure 11 shows the spectrum of the sound pressure level at the final engine speed of 1900rpm (95Hz) for the case when the adaptive controller was turned off and the piston was fully retracted (labelled "No Control") and when the adaptive controller was turned on (labelled "With Control"). There was a 30dB reduction in the tone at the engine firing frequency.



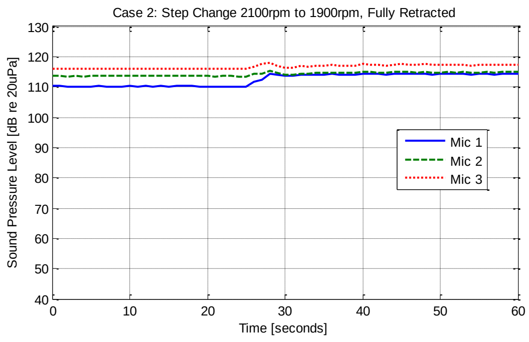


Figure 9: Sound pressure level in the duct when the engine speed was step changed from 2100rpm to 1900rpm. Top graph shows when adaptive control was implemented, bottom graph shows when the piston was fully retracted.

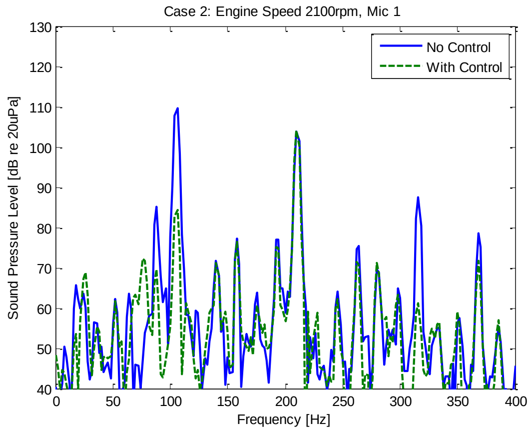


Figure 10: Sound pressure level spectrum at Mic 1 when the engine speed was 2100rpm when the piston was fully retracted (No Control) and with adaptive control (With Control).

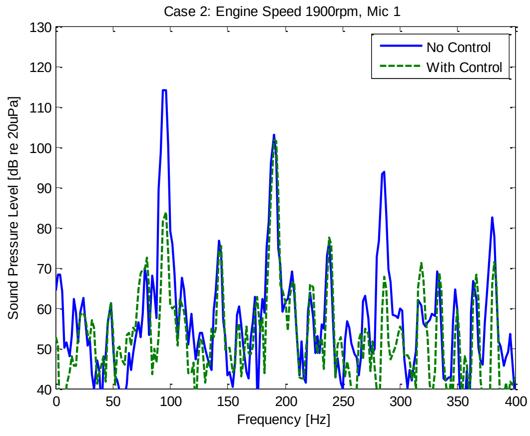


Figure 11: Sound pressure level spectrum at Mic 1 when the engine speed was 1900rpm when the piston was fully retracted (No Control) and with adaptive control (With Control).

SUMMARY

This paper described the implementation of an adaptive quarter wave tube and demonstrated its effectiveness using recorded exhaust noise from a reciprocating engine. The adaptive control system implemented a novel method of determining the phase angle of the transfer function between the pressure at the piston in the quarter wave tube and the pressure in the main duct, by using a sliding-Goertzel algorithm to calculate the Fourier coefficients.

Measurements of the in-duct exhaust noise from a 3.5 litre V6 Mitsubishi engine were recorded using water-cooled pressure sensors to protect them from the exhaust gas that was at temperatures greater than 500°C. A quarter-wave tube was attached to the exhaust system and was manually adjusted to attenuate noise at the fundamental and third harmonic of the engine firing frequency. It was shown that it is difficult to estimate the appropriate length of a quarter-wave tube because of temperature gradients in the exhaust system, and that an adaptive quarter-wave tube can be used to provide high levels of attenuation of a tone (and odd harmonics) over a range of frequencies.

REFERENCES

Bies, D.A. and Hansen, C.H. (2009), *Engineering Noise Control*, Fourth Edition, Spon Press.

Birdsong, C. and Radcliffe, C.J. (2001), "An Electronically Tunable Resonator for Noise Control", *Proceedings of the 2001 Noise and Vibration Conference* (NOISE2001CD), Traverse City, Michigan, April, SAE Technical Paper Series 2001-01-1615.

Cheng, C. R., McIntosh, J. D. Zuroski, M. T. and Eriksson, L. J. (1999) "Tunable acoustic system", United States Patent Number 5,930,371.

Chicaro, J.F. and Kilani, M.T. (1996), "A sliding Goertzel algorithm", *Signal Processing*, **52**, p283-297.

Ciray, M. S., (2005) "Exhaust processor with variable tuning system", United States Patent Number 6,915,876.

de Bedout, J.M., Franchek, M.A., Bernhard, R.J., Mongeau, L. (1997), "Adaptive-passive noise control with self-tuning helmholtz resonators", Journal of Sound and Vibration, 202 (1), p109-123.

Estève, S. J. and Johnson, M. E. (2004) "Development of an adaptive Helmholtz resonator for broadband noise control", *Proceedings of IMECE 2004*, Anaheim, CA, November. ASME International Mechanical Engineering Congress.

Garcia-Retegui, R., Gonzalez, S., Funes, M., Maestri, S. (2007), "Implementation of a novel synchronization method using sliding goertzel DFT", in: *IEEE International Symposium on Intelligent Signal Processing. WISP 2007*.pp. 1–5.

Howard, C.Q. (2011), "A new cost function algorithm for adaptive-passive vibration and acoustic resonators", *Proceedings of Acoustics 2011*, 2-4 November 2011, Gold Coast, Queensland, Australia

Izumi T, Takami H, Narikiyo T. (1991) "Muffler system controlling an aperture neck of a resonator. Journal of the Acoustical Society of Japan, 47(9), p647–52.

Koopmann, G. H., Neise W. (1982), "The use of resonators to silence centrifugal blowers", Journal of Sound and Vibration, 82 (1) p17-27.

Kostek, T.M., Franchek, M.A. (2000) "Hybrid noise control in ducts", Journal of Sound and Vibration, 237 (1), p81-100.

Kotsun, J.D., Goenka, L.N. Moenssen , D.J., and Shaw, C.E. (2004) "Helmholtz resonator", United States Patent Number 6,792,907.

Lamancusa, J. S. (1987), "An actively tuned, passive muffler system for engine silencing". Proceedings of Noise-Con 87, p313–318.

Nagaya K, Hano Y, Suda A. (2001), "Silencer consisting of a two stage Helmholtz resonator with auto tuning control. Journal of the Acoustical Society of America, 110(1), p 289–95.

Neise, W., Koopmann, G.H. (1980), "Reduction of centrifugal fan noise by use of resonators", Journal of Sound and Vibration, 73 (2), p297-308.

Stuart P. E. R., Variable resonator, United States Patent Number 6,508,331 (2003).

Utsumi, M. (2001), "Reduction of Noise Transmission in a Duct by Termination Impedance Control of a Sidebranch Resonator", Journal of Vibration and Acoustics, 123, p289-296.

Yuan, J. (2007), "Active Helmholtz Resonator With Positive Real Impedance", Journal of Vibration and Acoustics, 129, p94-100.