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ACTIVE EXHAUST SILENCING SYSTEM FOR THE MANAGEMENT OF AUXILARY POWER UNIT SOUND SIGNATURES

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ABSTRACT

We propose a system for the active cancellation of exhaust sound power where the desired outcome is a compact and lightweight solution to reduce exterior noise levels to inaudible operation at 20 meters. We have identified two challenges in developing this solution. The first is the integration of COTS technology to provide the signal processing for the active system, and the second is the development of a novel noise source and sensors which can withstand the extreme environment within a vehicle exhaust.

INTRODUCTION

Auxiliary power units (APU) offer potential to improve operational efficiency and provide significant tactical advantage for surveillance missions conducted from land vehicle platforms. Current installations have resulted in higher than acceptable noise levels, adversely impacting the ability to perform a "silent" watch. In particular, noise created by the combustion process presents a problem due to its relatively low frequency. Low frequency noise propagates efficiently through the atmosphere resulting in audible detectability at unacceptable range. Low frequency noise can be reduced using passive silencing techniques, but these devices are typically large and heavy and present objectionable integration burdens.

In the first section of this paper, the theory of noise cancellation within a duct will be discussed. An ideal noise source will be analyzed, but considerations will be made for different boundary conditions. A series of Finite Element models of an ideal system will be presented which help to establish the expected maximum performance capabilities of an active system, as well as identifying significant features of the system, such structural and acoustic resonances. A model for the noise source will also be shown, and the importance of various features in the model will be considered.

In the second section, two separate control algorithms will be analyzed. The two algorithms can be classified as a 'broad band' approach and a 'narrow band' approach, respectively. The relative benefits and problems associated with each method will be discussed, and data will be presented on the relative effectiveness of both. Finally, the performance of the system on an idealized lab source and a small engine will be compared though recorded data.

THEORY OF SOUND CANCELLATION IN A DUCT

The goal of an active noise cancellation system is to find a pressure field to be injected into an exhaust duct which destructively interferes with the downstream traveling source wave. This produces a net quieting effect at the exhaust port. This approach is complicated by the impedance mismatches at the primary and secondary sources, and exhaust port which cause multipath reflections. These reflections superimpose with the primary wave traveling in pipe and must be taken into account by the canceling system.

An ideal model for the system is necessary to develop in order to analyze the system behavior. To develop a model, a similar approach to [1] is taken. Two conceptual mass-less pistons are introduced into the system before and after the secondary source, such that they will move exactly with the plane wave particle velocity fluctuations associated with either the primary or secondary source. It is assumed that the pistons are 'close' together relative to the wavelength of the wave, that the pipe has an anechoic termination on either end, and that the secondary source has infinite internal acoustic impedance. This model is then analyzed for an understanding of exactly how injecting flow into the middle of a duct affects the pressure and velocity fields within the duct.

The behaviors that are derived in this ideal system can be considered valid in the real system because the additional effects created by the source are mainly linear in nature, so superposition holds.

When flow is forced into the left side of the pipe by the primary source, it causes both pistons to move identically. The pressures induced by the flow on the pistons do not affect the flow generated by the secondary source because of the infinite impedance of the secondary source. This means that when the cancelling source is not active, it is assumed to not affect the flow or pressures in the duct.

When flow is injected between the pistons, they will move in opposite directions in equal magnitudes. By plotting the pressure and velocity fields from the outside of each piston, it can be seen that the pressure field will always be continuous, but that the velocity field exhibits a discontinuity. The magnitude of the discontinuity is proportional to the flow injected between the pistons.

Let $U(y_+)$ be the movement of the right hand piston, and $U(y_-)$ be the movement of the left hand piston, *q(x)* be the volume velocity of the gas particles in the pipe, and *S* be the surface area of the secondary source. ρ_0, c_0, and k are the density of air, speed of sound, and wave number. Then, in the upstream pipe, the pressure *p(x)* and particle velocity *u(x)* within the pipe can be written as follow:

$$\begin{aligned} p(x) &= \rho_0 c_0 U(y_+) e^{-jk(x-y)} \\ u(x) &= U(y_+) e^{-jk(x-y)} \end{aligned} \tag{1}$$

and that downstream, the following is true:

$$\begin{aligned} p(x) &= -\rho_0 c_0 U(y_-) e^{jk(x-y)} \\ u(x) &= U(y_-) e^{jk(x-y)} \end{aligned} \tag{2}$$

Let *q(y)* be the volume velocity injected into the pipe. By recognizing that the velocity of the pistons is dependent on the injected flow, then the following equation can be written:

$$U(y_+) = -U(y_-) = q(y)/2S \tag{3}$$

Combining these observations, equations of motion of the particles within the system can be derived. Then, the pressure field in the pipe can be written as follows.

$$p(x) = q(y) \frac{\rho_0 c_0 e^{-jk|x-y|}}{2S} \tag{4}$$

This important equation describes how the pressure field within the pipe is related to the injected volume velocity. Given a source model and the pressure already within the pipe, it is now possible to understand how the secondary source must act to produce a canceling effect. Note that the factor of 2 in the denominator of equation (4) comes from the injected volume velocity splitting such that half the velocity flow flows upstream, and the other half downstream.

Secondary Source Model

Once the relationship between the injected volume velocity and pressures in pipe is understood, it is necessary to develop a model of how the secondary source produces the volume velocity. The secondary source is assumed to be a second order mass-spring-dampener system with an attached mass-less piston. This is a good model to choose because it correlates well with data at low frequencies and because it lends itself well to analysis. More accurate higher order models would require computer simulation.

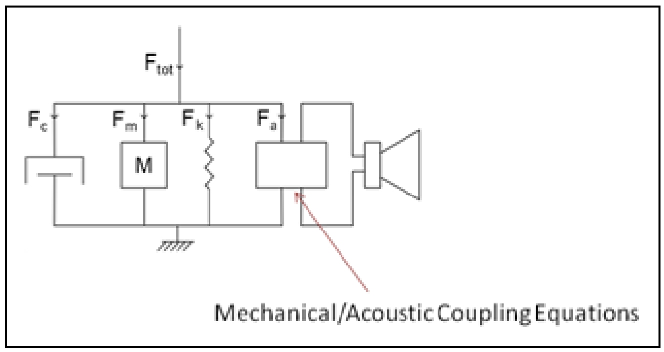

Figure 1: Circuit representation of secondary source

The dynamic model of the secondary source can be written in the following equation. Let *M* be the mass of the bellows, *C* be the mechanical damping, and *K* be the spring constant of the source. *F* is the applied force on the source, most likely from an electromagnetic coil and *x* is the position of the mass/spring/damper/piston.

$$M\ddot{x} + C\dot{x} + \frac{1}{2}\rho_0 c_0 S\dot{x} + Kx = F \tag{5}$$

In (5), there is an additional term which does not appear in the standard mass spring damper model. This term ($\rho_0 c_0 S\dot{x}$) is called the acoustic coupling term and describes how energy is transferred from the mechanical system into the acoustic domain. Note that it is a damping term, being proportional to $\dot{x}$ and that it depends upon the area of the source, which checks with intuition.

This acoustic coupling term is derived by looking at equations (3) and (4) and knowing that the volume velocity is related to the motion of the source piston by the following equation.

$$\dot{x} = \frac{q(y)}{S} \qquad (6)$$

By combining equations (3), (4), and (6) and evaluating at the point x=y=0 (where the interface to the piston exists), it can be shown that:

$$p(t) = \frac{\dot{x}\rho_0 c_0}{2} \qquad (7)$$

Finally, by using the definition of force ($F = p\,S$), the acoustic coupling term can be derived (Where F_a is the force due to the acoustics pressures in the duct).

$$F_a(t) = \frac{1}{2}\rho_0 c_0 S\dot{x} \qquad (8)$$

This equation then becomes one of the terms in the dynamic model of the injected source.

CHOOSING A SECONDARY SOURCE

With a functioning secondary source model, it is possible to evaluate different source parameters to find a well behaved source for our expected pressures and frequencies.

There are several considerations to take into account when choosing a secondary source. The largest is that it must survive in exhaust temperature environments. While a loudspeaker would be ideal to use because it is already designed to transfer power from an electrical system into the acoustic domain, we are not aware of any speakers built with materials that could survive in such a hostile environment.

Instead, a mechanical bellows attached to an electro-mechanical shaker was chosen. A bellows has good properties because it can be made of high durability materials like stainless steel, and unlike a piston system, there is no sliding interface which could be a potential source of problems.

Limitations of the bellows/shaker system include the total power which can be produced by the shaker/electrical drive system and the total travel range of the shaker/bellows. Both of these constraints can be evaluated using the source model derived in the previous section.

Power transfer between the electrical and acoustic domains can also be characterized using this model. There are two real power sinks within the secondary source, which are mechanical dampening and the acoustic coupling. Either energy will dissipate from the system as heat or acoustic radiation. Our goal is to pick a system such that the acoustic radiation term is dominant over the heat loss term.

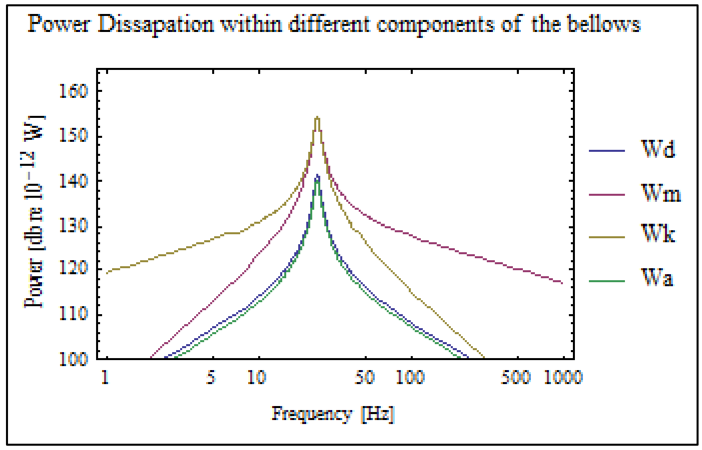

Figure 2: Magnitude of power dissipation for a 10cm diameter bellows in each of the bellows model components when excited with a uniform force spectrum. Wd – Dampener, Wm – Mass, Wk – Spring, Wa – Acoustic Dampening. Notice that the power dissipated due to mechanical dampening is similar to the power transferred into the acoustic domain.

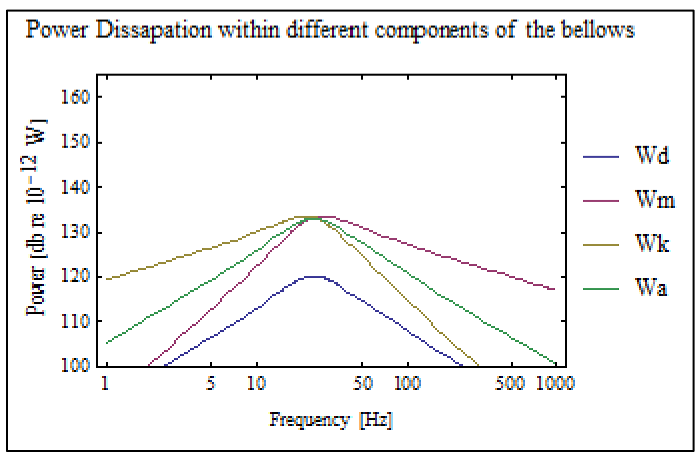

Figure 3: Graph is for a 50 cm diameter bellows. Notice that increasing the bellows diameter increases the total dampening of the system, which decreases the resonance peak. Also, more power is being lost to acoustic energy than to mechanical losses.

Figure 4: Example of mechanical bellows used as secondary source

MODELLING THE SYSTEM DYNAMICS

Once a bellows and shaker are picked out, they must be evaluated for performance. The maximum theoretical performance of the system should also be found, to compare against actual results.

To perform this analysis, a 3D vibro-acoustic finite element simulation is used from within a program named VA One. The physical model is set up in the program and a source representing the engine is attached to the exhaust manifold. The magnitude and phase of the pressure wave in pipe is measured at a sensor downstream of the secondary source, known as the cancelling point or error sensor.

The engine source is then disabled, and the force necessary to create a wave equal in magnitude but with a 180 degrees phase shift at the cancelling point is found. When both sources are active, these wave superimpose at the cancelling point producing a very low sound pressure level.

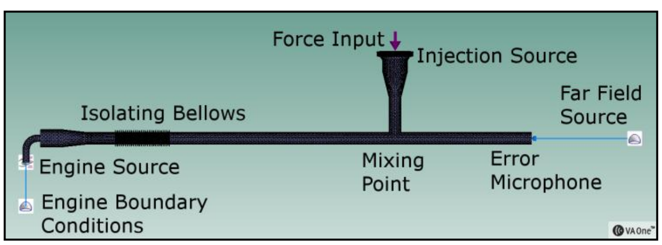

Figure 5: 3D vibro-acoustic model of canceling setup

Because of reflections occurring at each of the boundaries (engine manifold, secondary source, and outlet) the canceling effect is not constant with location downstream of the secondary source as one would expect. However the effect is still quite pronounced. To evaluate the effects that will be observed by any listeners, VA One is used to take a measurement located at an arbitrary distance away from the exhaust port in free space. This measurement is repeated both when the bellows force is enabled and disabled, and the difference is found. This value represents the maximum potential cancellation possible by the bellows system.

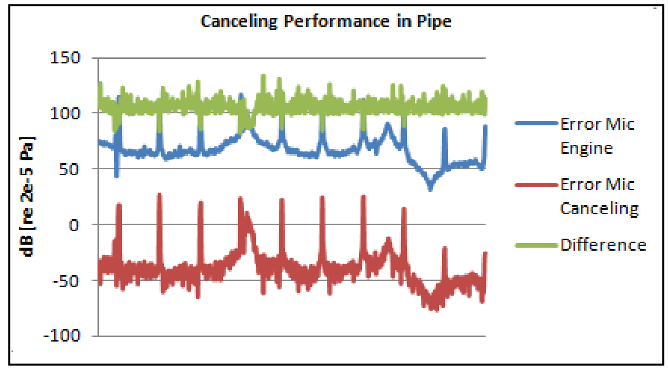

Figure 6: Canceling model results on an undisclosed engine at the canceling point in pipe.

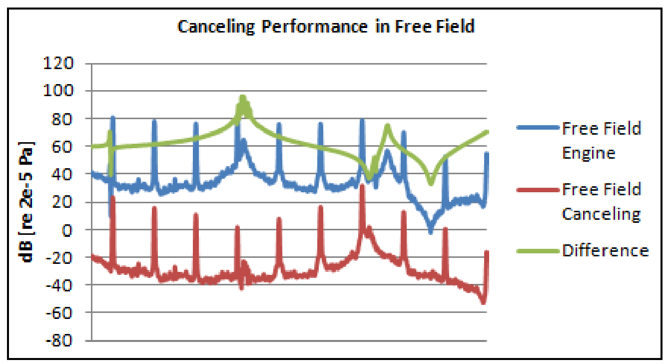

Figure 7: Canceling model results on an undisclosed engine at the far field measurement point in free space.

Using this canceling model, the forces applied to the bellows and the velocity of the bellows can also be measured. These must be below the rated specifications for the shaker/amplifier maximum force and the shaker/bellows maximum travel.

CANCELING ALGORITHM

The remaining effort falls on creating a system that is capable of producing a pressure field that is equal in magnitude but opposite in phase of a detected signal. To be robust the system must adapt to a changing environment. There are many variations of adaptive algorithms, but they all center around two different types of iterative processes, which are filtering and adaptation.

The filtering process consists of a series of multiply and accumulate operations that occur between the discrete values of the incoming reference signal and the set of gains that represent the adaptive filter. The output of the filtering process is then compared with a desired signal *d(k)*. The subtraction of *y(k)* from *d(k)* creates an error term *e(k)* which is then passed into the adaptation process.

The adaptation process uses a minimization function that alters the adaptive filter gains in order to minimize the value of *e(k)* such that *y(k)* approximates the phase and opposite magnitude of *d(k)*. The type of minimization function used separates adaptive algorithms into three general categories

- Newton
- Quasi-Newton
- Steepest Descent

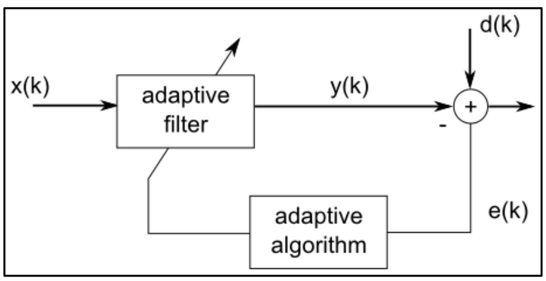

Figure 8: Generic form of an adaptive filtering process. X(k) is referred to as the reference signal, d(k) is called the desired signal, y(k) is the output signal, and e(k) is the error signal.

Figure 8 shows a high level depiction of the filtering and adaptive update process. It is helpful to understand what each of the signals represents. *X(k)* is called the *reference signal*. In the classic broadband active noise control (ANC) applications the reference signal is usually detected as a pressure wave being emitted from the *primary source* using a microphone. *Y(k)* is usually called the secondary signal because it gets directly routed to the *secondary source,* which is a noise generation device, usually a speaker.

One of the most popular adaptive algorithms in use today is called the Least Mean Squares (LMS) algorithm. The objective function of the LMS algorithm can be seen in Equation 9 [2]. ***W(k)*** is a vector of adaptive weights, μ is the step size of the algorithm, ***x(k)*** is a vector of the current and past inputs, and *e(k)* is the mean square error.

$$\boldsymbol{w}(k+1) = \boldsymbol{w}(k) + \mu e(k)\boldsymbol{x}(k) \tag{9}$$

where

$$e(k) = d(k) - \boldsymbol{x}(k) * \boldsymbol{w}(k) \tag{10}$$

$$y(k) = \boldsymbol{x}(k) * \boldsymbol{w}(k)$$

The reasons for the popularity of the LMS algorithm are many, including low computational complexity, stability when working with finite word lengths, and proven ability to converge in non-stationary environments [2].

In many ANC applications there isn't the ability to directly access the secondary signal, *y(k)*. The diagram in Figure 8 must be modified to include what is known as a secondary path which resides between the output of the adaptive filter and the point where the signal *e(k)* is detected. Figure 9 shows the updated representation that is used.

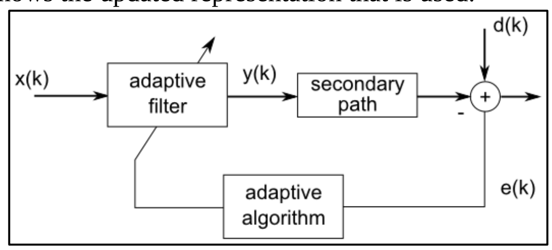

Figure 9: Adaptive algorithm with y(k) being observed through secondary path

There are two methods that can be used in order to account for the observation of *y(k)* through the secondary path. The first involves taking the inverse of the estimated transfer function of the secondary path. As long as the secondary path is slowly time varying, it is also possible to process the reference signal through the secondary path prior to passing it into the adaptive algorithm. In order to support either arrangement, an estimate of the secondary path transfer function must be obtained. The modified algorithm, also known as the Filtered-X Least Mean Squares (FXLMS) algorithm, is represented by the following set of equations:

$$\boldsymbol{w}(k+1) = \boldsymbol{w}(k) + \mu e(k)\boldsymbol{x}'(k) \tag{11}$$

where

$$\boldsymbol{x}'(k) = \hat{\boldsymbol{s}}(k) * \boldsymbol{x}(k) \tag{12}$$

Where $\hat{s}(k)$ is the secondary path estimate. The FXLMS algorithm was chosen for the APU noise control application outlined in this paper.

IMPLEMENTATION

One of the downsides of the standard broadband approach using the FXLMS algorithm is the effects of feedback from the secondary source to the reference microphone. The secondary source emits sound in both the downstream and upstream traveling directions. The downstream traveling wave mixes with the primary wave in order to cancel it, but if the upstream traveling wave is not properly accounted for it can corrupt the signal that gets read at the reference microphone. In order to account for it, a representation of the feedback path must be correctly modeled so that the output of the secondary source can be processed through it before being subtracted from the reference signal. This process becomes more difficult to handle in exhaust applications because of the fact that the feedback path is non-stationary and can vary significantly with engine load. The main driver affecting the variability of the feedback path is engine exhaust temperature, which has a significant effect on the speed of sound. If the error in the estimate of the feedback path becomes too large, poles can be introduced into the controller transfer function which can affect system stability [2].

The problems described above were addressed by modifying the reference signal so that it is synthesized from a non-acoustic sensor. The non-acoustic sensor in our application was in the form of a tachometer signal. Since the reference signal is now synthesized, it adds the capability of only targeting the strong harmonic components that an engine usually emits. This simplifies the adaptive filter structure shown in Figure 9 to a set of two gains per target frequency. Figure 10 shows a pictorial representation of what is known as the adaptive notch algorithm.

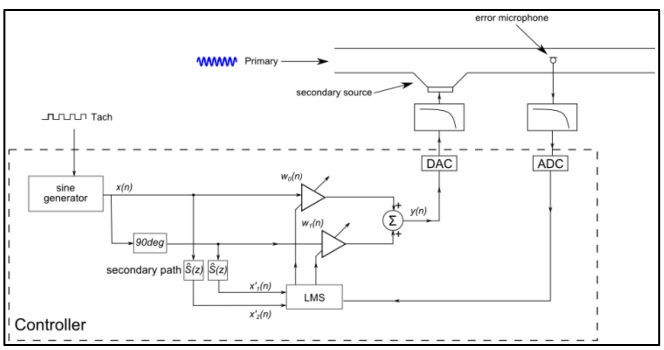

Figure 10: The adaptive notch algorithm with FXLMS as the adaptive update mechanism.

In order to target multiple frequencies the adaptive notch algorithm can either be run in parallel or cascade form. Due to resource constraints on the hardware that was being used to implement the adaptive notch algorithm it was decided that the cascaded algorithm would be the best implementation approach. The modified algorithm is shown in Figure 11.

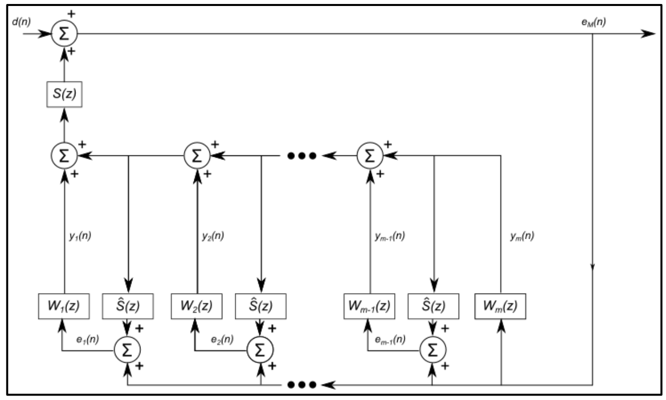

Figure 11: The cascaded adaptive notch algorithm.

Before the FXLMS algorithm can be used, an estimate of the secondary path must first be obtained. During this secondary path learning process white noise is played out of the secondary source while a LMS algorithm that is independent of the ANC algorithm is used to alter the tap values of a 100 tap FIR filter that is used to represent the secondary path. After the learning process is complete the filter taps are saved for later use in the FXLMS algorithm.

For the application outlined in this paper the adaptive notch algorithm was implemented on the National Instruments CompactRIO data acquisition hardware. The bulk of the algorithmic processing was synthesized onto the FPGA chip that is onboard the CompactRIO chassis. The RTOS on the CompactRIO handled all parameters manipulation and supervisory control.

Testing is currently being performed on a 205cc Briggs and Stratton engine with future plans of moving to the Marvin Land Systems Auxiliary Power Unit.

RESULTS

The preliminary engine noise cancellation results look very promising. When targeting the second third and fourth harmonics of the firing frequency an overall sound power level reduction of 12 dB at the targeted frequencies can be achieved. Attempting to include more harmonics for cancellation causes the secondary source to draw past its power limits and the system automatically shuts down.

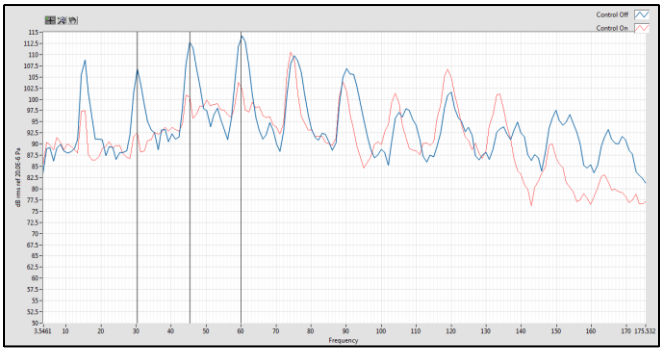

Figure 12: Adaptive notch cancellation performance targeting second, third, and fourth harmonics

Target Frequency	SPL Reduction
30 Hz (2nd Harmonic)	14 dB
45 Hz (3rd Harmonic)	12 dB
60 Hz (4th Harmonic)	12 dB

Table 1: Adaptive notch cancellation performance at engine idle targeting second, third, and fourth harmonics

As Table 1 indicates, the system is capable of targeting the first four harmonics of the Briggs engine that was used for testing. The data in Figure 12 and Table 1 was collected during the engine idle condition which is where the system was most capable. As the engine speed is increased the imaginary impedance due to the mass of the bellows end cap acts as a drag on the cancellation system. This effect starts to reduce the effectiveness of the control system.

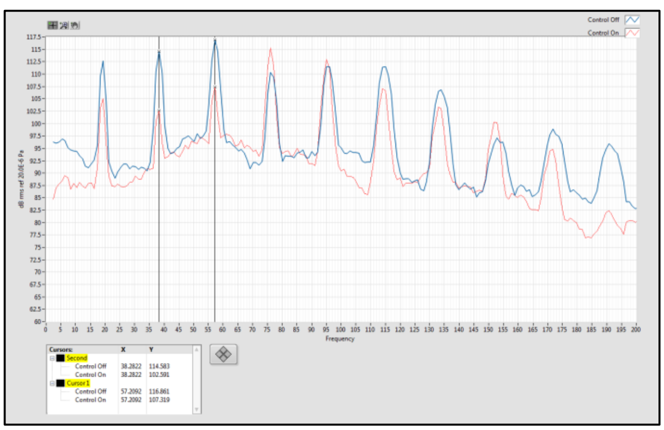

Figure 13: 2300 RPM engine cancellation targeting the second and third harmonics.

Figure 13 shows the cancellation ability at 2300 RPM of the Briggs engine. The cursors indicate the targeted harmonics.

Target Frequency	SPL Reduction
38 Hz (2nd Harmonic)	12 dB
57 Hz (3rd Harmonic)	9 dB

Table 2: Adaptive notch cancellation performance at 2300 targeting second and third harmonics

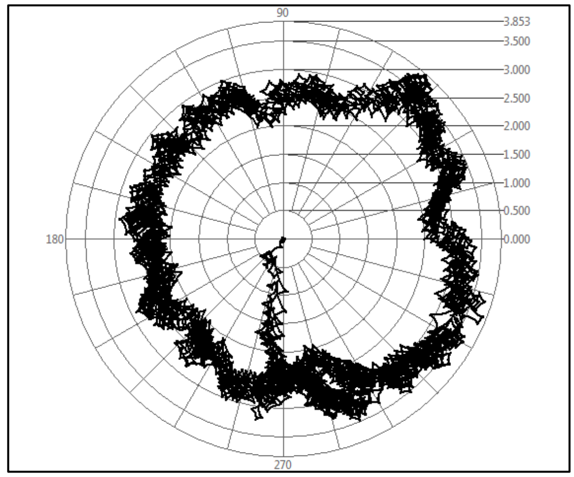

Figure 14: Adaptive notch phase compensation due to errors in estimate of firing frequency

LESSONS LEARNED

There were many practical aspects of ANC that were learned throughout the implementation of the adaptive notch algorithm. The biggest hurdle was implementing the full algorithm onto an FPGA chip. LabVIEW eases the process by allowing a graphical approach to the development, but there are still slight nuances that can have a huge effect on the ability of the code to execute correctly. For example, since the majority of the code can execute in parallel it is very important that the arbitration options are set correctly when reading and writing to block memory. If the arbitration is not set correctly it is possible to miss a write cycle due to memory access conflicts. Memory access conflicts do not show up in the simulation environment, so they can be very difficult to track down if one does not know what they are looking for. It was found that memory access was the major source of complications during the algorithm development process.

Some of the aspects of synthesizing a reference signal also caused issues early on in the project. Initially it was thought that in order to create a phase locked reference signal the phase drift between the synthesized reference and the primary signals would need to be reset every time a tachometer pulse was detected. The thought was that this would prevent the drift between the signals from getting too large. In reality the resetting of the phase at each tachometer pulse caused audible discontinuities in the control signal due to rollover effects that couldn't be compensated for using this approach. The final implementation leaves it to the adaptive notch algorithm to compensate for the buildup in phase error. This puts a lower bound on the size of the value of μ which is known as the step size of the algorithm. This phase compensation phenomenon can be seen as a circular pattern when plotting the gains for a specific frequency on a polar plot, this can be seen in Figure 14. If the step size of the algorithm becomes too small the gain polar plot will be noticeably off center.

REFERENCES

[1] A. Nelson , P.and Elliott, Active Control of Sound, Academic Press, 1992

[2] Kuo, S.M. and Morgan , D.R., Active Noise Control Systems Algorithms and DSP Implementations, John Wiley & Sons Inc, 1996