

## Active Control of Energy Density in a Mock Cabin

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### 1. INTRODUCTION

Acoustic energy density's dependence on particle velocity as well as acoustic pressure makes it a good candidate for an error signal in an adaptive noise control algorithm. It has been shown that minimizing acoustic energy density often provides more global attenuation of an enclosed sound field than minimizing squared acoustic pressure.<sup>1</sup> Nodal regions in three-dimensional enclosed sound fields comprise entire planes when looking at squared pressure and lines when looking at energy density. This means that for most locations in the sound field, an energy density sensor will have a much lower probability of being at a nodal position than a pressure sensor for the natural modes of the enclosure.<sup>2</sup>

Active noise control has been applied to tractors in order to reduce operator noise exposure.<sup>3-5</sup> The research discussed in this paper applies active minimization of acoustic energy density to the problem of noise attenuation in enclosed tractor cabins. Previous approaches to this problem which based themselves on minimization of squared pressure achieved attenuation in a relatively small area localized around the operator's head and required that the error sensors be placed near the operator's ears, thus restricting the operators range of movement to either side.<sup>3-5</sup> However, the results discussed later in this paper were obtained with an energy density sensor located directly above the operator's head against the ceiling of the cab.

### 2. CONTROL SYSTEM DEVELOPMENT

Both acoustic pressure as well as particle velocity must be measured in order to obtain the acoustic energy density. In order to do this, an energy density sensor was developed, which consists of four inexpensive electret microphones arranged in two orthogonal pairs on the face of a short cylindrical enclosure. Each pair of microphones estimates the particle velocity at the mid point between them in the direction established by a cartesian axis passing through both microphones. The pressure at the center of the sensor is estimated by averaging the pressures measured by all four microphones. Only two microphone pairs are used for the sensor, because of the sensor's placement against the ceiling of the cab. The particle velocity in the direction normal to the ceiling will be zero at the ceiling's surface if that surface is perfectly rigid. The assumption used in this research is that the particle velocity normal to the ceiling is small enough to be neglected in the measurement of acoustic energy density. The current sensor is therefore referred to as a two-dimensional energy density sensor, or 2D sensor. The orientation of the 2D sensor is such that the two directional components of particle velocity are measured in directions normal to the sides and back of the tractor cabin. As part of this research, numerous tests have been conducted to verify experimentally that the 2D sensor performs as well as a three-dimensional energy density sensor in a similar location. A photo of the sensor is shown in Figure 1.

The algorithm used in the active noise control system is a filtered-x LMS adaptive filter, modified for use with an energy density error signal.<sup>6</sup> In this research, a feed forward approach was used to attenuate tonal noise at the engine firing frequency. Even though engine speed can change significantly in a short period of time during the normal work cycle of a tractor, the active noise control system only attempts to attenuate a single tone corresponding to the engine firing frequency at any point in time. The reference noise signal comes from a tachometer signal, which is correlated with the engine speed. Currently, a single 2D sensor provides an error signal to the control system and two control channels drive a set of loudspeakers to provide an acoustical control signal. The loudspeaker set consists of a subwoofer and two full range drivers. For control signal frequencies below 90 Hz, the two control signals are summed electronically and fed to the subwoofer. Above 90 Hz, each control channel drives

a separate full range loudspeaker. The reason for the subwoofer/satellite approach is twofold: full range drivers cannot produce sufficient sound pressure levels at low frequencies without creating unacceptably large amounts of harmonic distortion, and low frequencies have large enough wavelengths that a single loudspeaker can provide significant noise reduction from anywhere inside the cab.

A Motorola DSP96002 processor handled the implementation of the algorithm at a sampling rate of 2 kHz. Prior to running the control algorithm, a system identification algorithm was run to determine transfer functions which could then be used to generate the filtered-x signals for the energy-based control filter. An adaptive system identification algorithm could also be run simultaneously with the adaptive control algorithm, but was not used for the measurements discussed in this paper. Tradeoffs had to be made between higher sampling rates which would allow the control system to adapt more quickly to changes in the tractor's engine speed and system identification filter lengths which, if too short, would prevent the control system from converging to an optimal solution very quickly.

### 3. MEASUREMENTS

In order to simulate the noise to which an operator is exposed in an enclosed tractor cab, an audio signal recorded in an actual tractor cab is played back inside of a mock tractor cab. The mock cab consists of a metal frame approximately 1 meter wide and 1.5 meters tall. The length of the mock cab is 1 meter at the top and 1.2 meters at the bottom. The ceiling, floor, sides and back of the mock cab are made of 1/2-inch plywood and the front is thin plexiglass. A photo of the cab can be seen in Figure 2. The audio recording was made originally on Digital Audio Tape (DAT), after which it was transferred to compact disk for easy track selection. A Mackie HR 824 loudspeaker, situated beneath the seat in the mock cab, reproduced the recorded cab noise. The frequency response of the Mackie loudspeaker remains quite flat ( $\pm 1.5$  dB) between 40 Hz and 20 kHz, which suggests that the original tractor cab noise should have been reproduced with high fidelity. The active noise control system was run inside of the mock cab simultaneously with the recorded noise, and measurements were made of the achieved noise attenuation. For these measurements, each of the two full range loudspeakers was placed to one side of the operator's head against the side wall of the mock cab. The subwoofer sat on the floor in the front, left corner of the cab.

Recordings of tractor noise were made for various machine conditions. The tonal noise associated with the engine speed of the real tractor lies between 46 and 120 Hz. With the machine at idle, the engine tone is approximately 46 Hz and at full throttle the tone is near 120 Hz. Cab noise with engine tones of 46, 90, 110, and 119 Hz were played back inside the mock cab in order to obtain the results discussed in this paper. Tractor noise with engine tones at the lower three frequencies were recorded with the machine in a stationary position. The higher engine tone was obtained with the tractor moving forward in first gear at full throttle. Control system performance was also measured using a recording of the tractor in more dynamic conditions. For the dynamic recording, the tractor was used in a standard dig and dump cycle, which is intended to simulate truck loading. In this case, measurements of the noise spectrum inside the cab were time averaged in order to obtain an overall noise reduction for several dig and dump cycles.

In all measurements, a total of 15 microphones were used to get a picture of noise reduction throughout the mock cab. Twelve of the microphones were arranged to map out a three-dimensional grid inside the cab to provide a measure of global noise attenuation. The twelve microphones were divided into two horizontal planes, each consisting of two rows of three microphones with the rows parallel to the cab sides. The two planes were located such that the upper one was higher than a nominal operator ear location and the lower plane was below ear height. Two additional microphones were then mounted near the operator's ears and one of the microphones in the 2D error sensor was also monitored. An operator sat inside the mock cabin for each measurement.

### 4. RESULTS

Performance of the active control system was measured in terms of the amount of noise reduction achieved at the frequency of the engine tone corresponding to the engine speed as well as the amount of overall A-weighted sound level reduction. Table 1 presents the reduction of the engine tone for the tractor noise recordings in which the engine speed was held constant. For most engine speeds, the greatest reduction of the tone occurred at the error sensor itself, as would be expected. For the low frequency case, the microphones at the ear locations actually show slightly more reduction. Wavelengths around 46 Hz are significantly larger than the dimensions of the cab so the attenuation is quite global in this case, as can be seen in the 12-microphone spatially averaged attenuation. In all cases, global attenuation of the engine tone was significant and the greatest difference in attenuation between the 12-

microphone average and the error sensor location occurred at the highest engine tone frequency. Poorest attenuation occurred at an engine speed of 1800 RPM, which corresponds to a tone of 90 Hz. Attenuation was poorest at this frequency largely because in this case, the uncontrolled engine tone was not as loud as the tones at the other engine speeds. Attenuation near the operator's ears was greater than 20 dB for both ears in nearly all cases. For some measurements a significant difference in noise reduction was seen between the two ear microphones. This may have been influenced by the operator's seat location, which was not perfectly centered between the sides of the mock cab. Figures 3 and 4 compare the noise spectra measured at the error sensor and right ear locations for uncontrolled and controlled tonal noise. At both measurement locations, the engine tone was almost completely eliminated and the rest of the noise spectrum unchanged as would be expected for a feed forward control algorithm with a tonal reference input. These results were achieved toward the upper end of the engine tone frequency range, which suggests that placing an energy density sensor against the ceiling of the cab where it can be completely out of the operator's way can still produce significant noise reduction in the region around the operator's head.

Overall A-weighted sound level reductions are shown in Table 2. The second column of the table shows overall reduction for several dig and dump cycles at the various microphone locations in the cab. A spatially averaged reduction of approximately 1 dBA was achieved, with 1.5 dBA reductions at the error sensor and ear locations. The time-averaged spectrum of the noise at the operator's right ear is shown in Figure 5. When looking at the comparison of the spectra with and without the control system running, it becomes apparent that the control system actually added noise in the frequency range between 60 and 90 Hz even though it reduced noise levels near the high and low extremes of the engine tone range. This discrepancy is likely due to the speed at which the control system adapted to the changes in the tractor's engine tone. During a standard dig and dump cycle of the tractor, more time is spent at the higher and lower engine speeds than at the intermediate speeds. Higher engine RPM's facilitate digging and dumping and lower engine RPM's occur during direction changes, three of which take place during a standard cycle. The engine speed changed more slowly during digs, dumps, and direction changes, allowing the control system more time to converge on an optimal solution for cancellation of the engine tone. Generally, changes from high RPM to low and from low RPM to high happened very quickly. Therefore, the added noise in the intermediate frequency range may be due to the limited adaptation speed of the control system in the configuration used for these tests. For example, during the ramp up from low RPM to high, the actual engine tone may be at 80 Hz by the time a 70 Hz control signal is generated. Rather than canceling the engine tone that actually exists then, the active noise control system is adding a control tone for an engine tone that has already passed.

The last four columns of Table 2 show the sound level reductions (in dBA) achieved with constant tractor speeds. Except at 1800 RPM, global reduction was greater than 3 dBA in each case. At 1800 RPM, as mentioned before, the uncontrolled engine tone was not as loud as at other speeds. Additionally, the engine tone at this frequency was no longer the dominant peak in the spectrum, which lessened its contribution to the overall sound level. Additionally, the dominant peak was at a higher frequency, which also meant that reducing the 90 Hz tone did less to affect the overall level because of the A weighting. Again, in all cases the amount of attenuation tended to be significant throughout the cab, and not just near the error sensor.

## 5. CONCLUSIONS

Targeted tonal noise inside of the mock cabin was cancelled almost completely through active minimization of the acoustic energy density. Cancellation tended to be significant throughout the cab, even with the error sensor placed against the ceiling toward the back of the cab. Overall A-weighted sound level reductions of several dBA were achieved for most situations in which the tractor engine speed remained constant. Measured noise reductions for tractor dig and dump cycles were approximately 1 to 2 dBA. It appears that the overall attenuation in dynamic machine conditions may be improved by increasing the speed with which the control system adapts to changes in the engine speed. This could possibly be verified by introducing well-defined swept sine signals with different sweep rates as noise into the mock cab and as reference signals into the control system. To actually increase the adaptation speed of the current noise control algorithm would likely require a processor with greater computing power than the DSP96002 used for these tests. Even with the existing processor, the performance of the control system could be further optimized by exploring the effects and tradeoffs of altering filter lengths, sampling rates, and convergence parameters.

## REFERENCES

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	idle	1800 RPM	2200 RPM	Full throttle forward
<b>12-Mic Avg</b>	33.47	16.55	23.97	20.62
<b>Right Ear</b>	34.95	22.24	30.48	30.60
<b>Left Ear</b>	35.02	19.41	23.68	26.72
<b>Error Sensor</b>	34.49	22.58	32.92	32.05

Table 1. Engine tone attenuation for constant tractor engine speeds (values are in dB).

	Dynamic	idle	1800 RPM	2200 RPM	Full throttle forward
<b>12-Mic Avg</b>	0.99	4.11	0.39	6.12	2.77
<b>Right Ear</b>	1.57	3.84	0.60	8.71	4.75
<b>Left Ear</b>	1.62	4.27	0.57	8.70	4.69
<b>Error Sensor</b>	1.55	2.60	0.36	7.68	4.48
<b>15-Mic Avg</b>	<b>1.1144</b>	<b>4.0085</b>	<b>0.41731</b>	<b>6.6516</b>	<b>3.1794</b>

Table 2. Overall sound level attenuation (dBA).



Figure 1. Two-dimensional energy density sensor.



Figure 2. Mock cab and noise control system electronics.

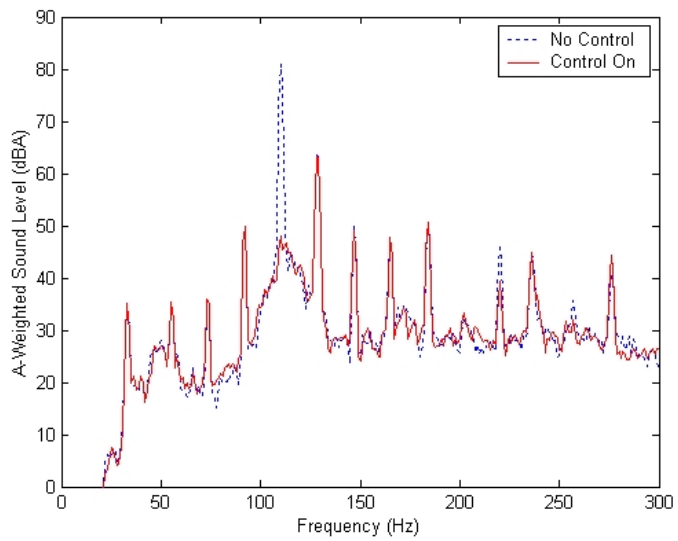


Figure 3. A-weighted spectrum for the tractor engine speed of 2200 RPM (110 Hz engine tone) measured at the error sensor.

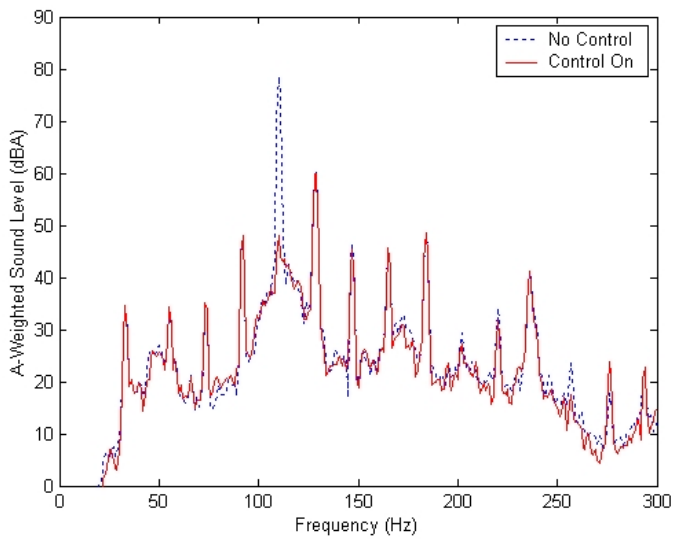


Figure 4. A-weighted spectrum for the tractor engine speed of 2200 RPM (110 Hz engine tone) measured at the operator's right ear.

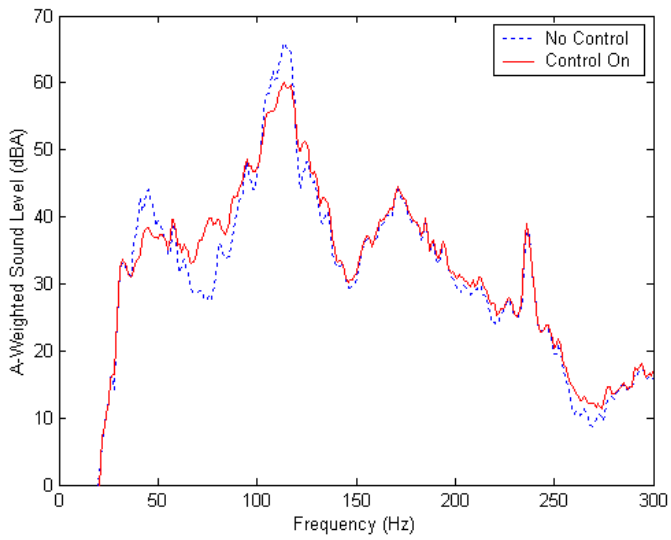


Figure 5. A-weighted, time-averaged spectrum for tractor dig and dump cycle, measured at the operator's right ear.