A Systems Approach to Appliance Compressor Quieting Using Active Noise Control Techniques

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Introduction

The purpose of this project was to quiet a standard household refrigerator/freezer using active noise reduction techniques. The model chosen for demonstration was US manufactured, had exposed coils and an open compressor compartment. The goal was to achieve the quietest refrigerator of its class that included a Japanese active cancellation model. The activity quickly focused on the single noisiest component - the compressor; however, the overall goal of a feasible, quiet refrigerator was the overriding constraint.

Active noise attenuation, a concept first described over fifty years ago, is artificially produced sound to counteract unwanted sound. The fundamental problem in active noise control in acoustics is to engineer the mixing of sound waves so that their superposition produces destructive interference throughout the whole sound field. This mixing of in-phase and anti-phase sound at the same acoustic power level as the original noise field attenuates the resultant noise field.

There are two main approaches to active noise attenuation. The first is to process the original sound and inject it back into the sound field in anti-phase. The second approach is to synthesize the canceling waveform with or without prior knowledge of the original sound. Any active noise control application will consist of the following, see Figure 1 : primary noise source (compressor, fan, etc.), secondary noise source (speaker, shaker, etc.), detector(s) for primary noise and residual error (accelerometer, microphone, etc.) and electronic controller to minimize the electrical signal at the residual error sensor.

It has been demonstrated that global attenuation of noise sources can be achieved only if the primary and secondary sources are located in close proximity (on the order of one-tenth of the wavelength of interest). If the secondary source is several wavelengths distant from the primary source, the room interference pattern will be adjusted by changing the residual error microphone location. But, a minimum sound pressure level will still be achieved at the error microphone. However, the size of the quiet zone is only one-tenth of the wavelength of interest and not global.

In this paper, application of an active and passive noise control system for a household refrigerator is evaluated. Experimental results showed that significant global attenuation can be achieved using this control system.

Experimental Methods

The method of solution to the problem was to take data on the noise sources, investigate transmission paths and to design a solution based on the physical constraints of the system. The solution could not increase the footprint of the appliance or cause operating problems with the compressor due to heat transfer.

Measurements were taken on the compressor itself in a reverberant room. Both A-weighted and linear plots were made. The compressor was operated on a load stand at the nominal suction and discharge pressures of 19.2 and 175 psia at 115 V.

The initial data (plotted in the results section) showed both low frequency, below 1000 Hz, and higher frequency noise from the compressor. The solution was therefore a combination of active and passive noise reduction. Active was designed for lower frequencies and passive designed for the higher frequencies. Thus, an enclosure was built that incorporated passive and active noise cancellation.

In order to control the refrigerator noise actively, NCT used the synthesizing technique to generate the anti-noise waveform from the primary compressor noise, see Figure 2. The synthesizing waveform was generated using an accelerometer that detected the fundamental motor turn rate on the compressor shell, about 58.7 Hz. A residual microphone was placed outside the enclosure and the speaker inside the enclosure in proximity to the compressor. The NCT controller was placed outside the enclosure. Passive measures consisted of 1/2 inch open cell foam inside the enclosure.

The enclosure was designed both from an acoustic and heat transfer point of view. The volume was tuned to 60 Hz to increase the acoustic coupling of the anti-noise signal. Therefore, this increased noise with the passive enclosure was in turn was canceled with the active system. For heat transfer, the enclosure had an inlet slit 1/2 by 9 inches near the compressor mounting feet and an outlet port in the back of the enclosure facing the wall.

Next, measurements were made on two appliances with factory installed compressors. One appliance was to be the control unit and was not further modified. The other was the experimental unit with the active enclosure installed. The measurements were made in a semi-reverberant room, about 1 meter from front and rear, with a room volume similar to that expected in a small kitchen.

In parallel with the above efforts, NCT also began looking into ways to quiet the compressor itself. Both passive and active means were investigated.

Experimental Results

The results are seen in the following Figures.

Figure 3 shows the A-weighted spectrum up to 200 Hz of the compressor in the reverberant room. Some peaks are noted with a low frequency broadband hump. The relationships of the peaks to physical phenomena in the compressor were not yet clear.

The next plot, Figure 4, shows compressor data linear weighted up to 2000 Hz. Clearly, all the peaks are harmonics of the turn rate of the compressor, about 58.7 Hz.

This result led to the approach of synchronizing to the motor turn rate and canceling that frequency to take the energy out of higher harmonics.

The next plots, Figures 5 and 6, were taken in the small semi-reverberant room from rear and front respectively. Figure 5a shows the narrowband spectrum of the unmodified refrigerator. Figure 5b is the modified with control off and 5c is modified with control on. Figure 5b shows how the higher frequencies are attenuated and the 58.7 Hz line is increased with the addition of the enclosure while Figure 5c then shows how that low frequency tone is reduced with the active control. Figures 6 a, b and c show similar results for measurements made in front of the unit.

Overall 1/3 Octave measurements are seen in Figures 7 a and b. The overall reduction is 5 dBA in the front and 10 dBA in the rear with the measurements taken in the small semi-reverberant room.

Table's 1-3 are data taken in controlled environments at the manufacturer. Table 1 is the sound pressure data showing the increase in 60 Hz tone but overall decrease with the enclosure while turning the active on reduced both 60 Hz and reduced the overall a total of 4 dBA. Table 2 shows similar sound power data with an overall reduction of 4.5 dBA. Finally, the environmental test at 110 degrees F was run to determine the effect of the enclosure on compressor temperature and efficiency. Table 3 shows the shell temperature stayed below the goal of 250 degrees F. Further heat transfer design would lower this maximum shell temperature.

Conclusions

The problem of refrigerator quieting has been approached as a system. Overall system noise was reduced by a combination of active and passive techniques using existing technology. The compressor itself was analyzed and the application of quieting methods was researched with promising results. This feasibility demonstration has shown that a quieter system can be obtained without modifying the compressor itself. Future work in designing a quieter compressor combined with the active enclosure will produce a very quiet appliance.

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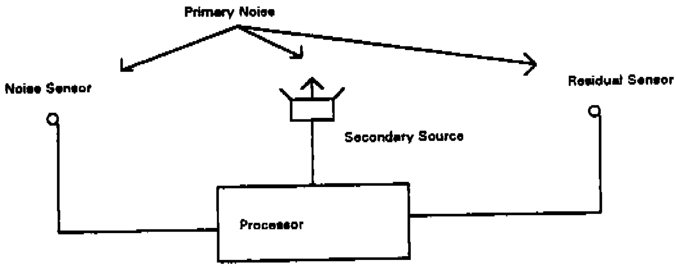

Figure 1. Active Noise Control Basic System

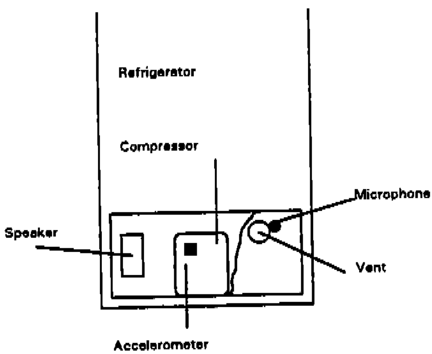

Figure 2. Active Refrigerator Enclosure

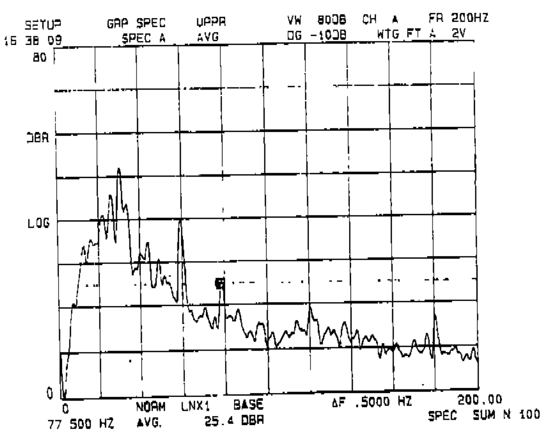

Figure 3. A-Weighted Compressor Noise Data

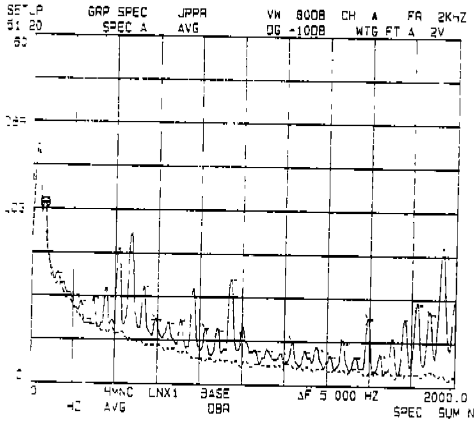

Figure 4. Linear Weighted Compressor Noise Data

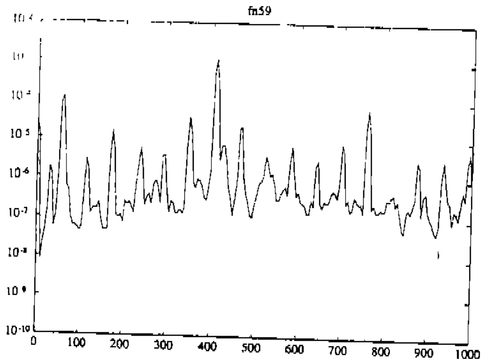

Figure 5 a. Narrowband Rear Measurement of Unmodified Refrigerator

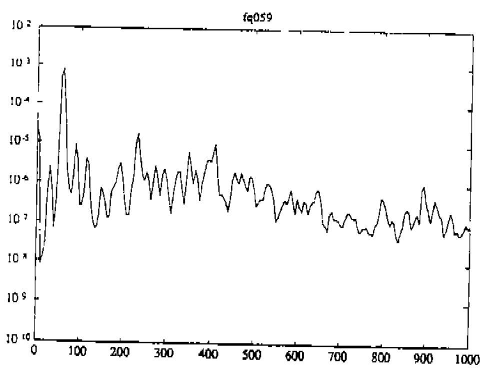

Figure 5 b. Narrowband Rear Measurement, Modified Refirgerator Control OFF

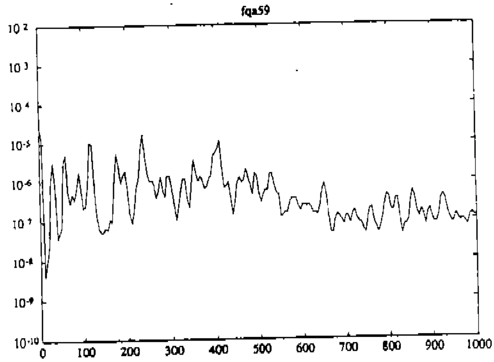

Figure 5 c. Narrowband Rear Measurement, Modified Refrigerator Control ON

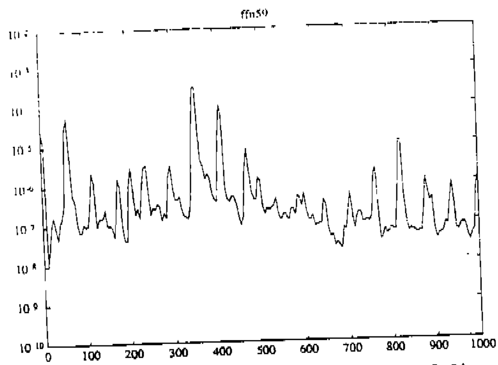

Figure 6 a. Narrowband Front Measurement of Unmodified Refrigerator

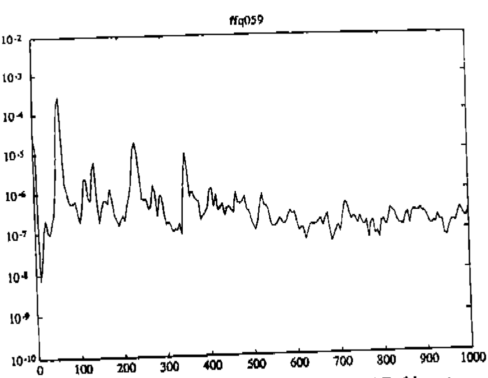

Figure 6 b. Narrowband Front Measurement, Modified Refrigerator Control OFF

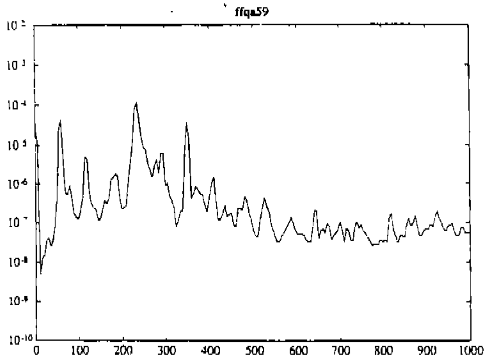

Figure 6 c. Narrowband Front Measurement, Modified Refrigerator Control ON

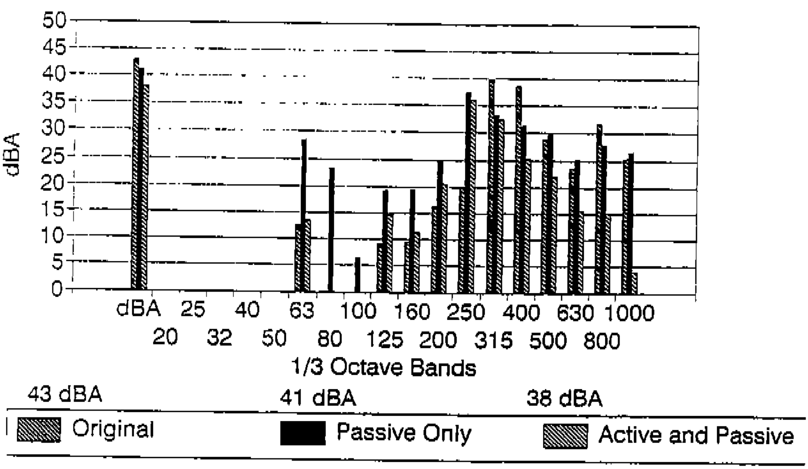

Figure 7 a. Refrigerator Comparisons, Front

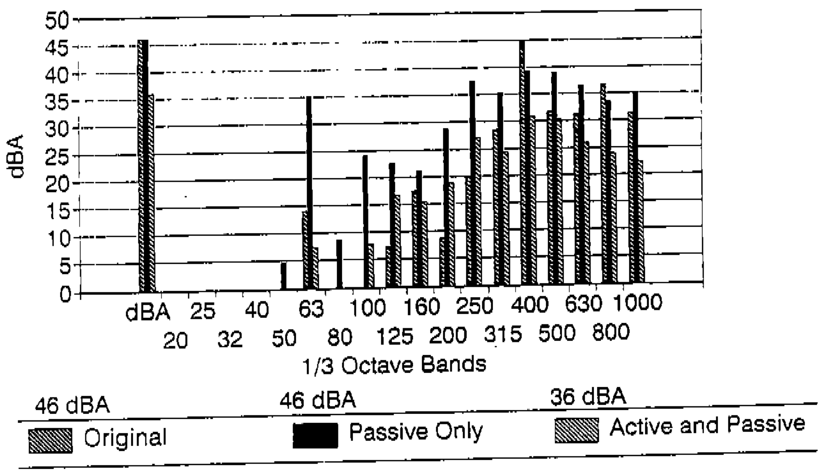

Figure 7 b. Refrigerator Comparisons, Rear

	60 Hz dBA	Total dBA
Baseline	32.7	39.3
With Enclosure Only	33.2	37.5
Active on	29.9	35.6

Table 1. Sound Pressure Data

	60 Hz dBA	Total dBA
Baseline	35.4	47.6
With Enclosure Only	36.2	44.9
Active on	30.4	44.1

Table 2. Sound Power Data

	Shell Temperature Degrees F
Baseline	221.8
With Enclosure	249.7

Table 3. Shell Temperature Data