POST CONSTRUCTION HVAC NOISE CONTROL

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1. INTRODUCTION

Newly installed HVAC systems that fail to meet acoustic design parameters share the common feature that the fault is detected just prior to occupancy. There are many configurations that give rise to the additional noise and for most there may be no effective and in-expensive mitigation. There is also some friction between the various parties: owner, architect, mechanical designer, mechanical contractor, and the acoustic consultant. The latter is expected to provide an instant solution.

2. SOURCE IDENTIFICATION

HVAC systems tend to evolve in time: differing from the lay-out set in the design phase to the as built configuration. Issues such as lack of clearance or cost constraints result 'creative' on site solutions that fit but add noise. Before recommending noise mitigation, it is necessary to identify the source of the additional noise. This is important, as a trial and error approach is really not appropriate. Experience certainly is an asset. Listening and selected sound measurements provide additional information, but may not suffice.

A two point space-time correlation is an effective tool for localizing sound sources. The concept is straight-forward in that the time of travel from the source to the listeners is measured. For low frequencies sound propagation in HVAC ducts is effectively one-dimensional. It is in this frequency range that a large percentage of the excess HVAC noise is generated. Consider a noise source at Y in a duct and transducers at X1 and X2 (Figure 1).

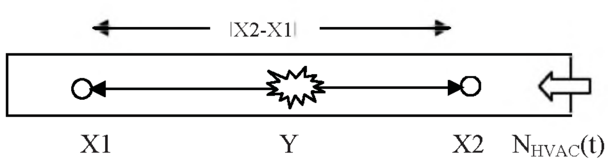

Figure 1 Schematic of HVAC system

transducer locations (X1, X2) and source location (Y)

The pressure has several components: $N_{HVAC}(t)+N(t)+S(t)$. $N_{HVAC}(t)$ is the HVAC system noise, $N(t)$ uncorrelated noise (some of which due to the microphone-flow interaction), and $S(t)$ the sound generated by the noise source. Referring to figure 1 it is evident that the HVAC noise detected at X2 will be measured $|X2-X1|/c$ seconds later at X1, and $R_{HVAC}(t)$ has a maximum at $t=|X2-X1|/c$. Similarly, $R_{SS}(T)$ has a maximum at $t=|X2-Y|/c-|X1-Y|/c$.

It follows that the correlations is the sum of two distinct contributions: $R_{12}(\tau)=R_{HVAC}(\tau)+R_{SS}(\tau)$. The magnitude of the correlation is a measure of the similitude of the two signals and the coherent energy in relation to the overall signal energy. This then permits one to assess source location and the relative source strengths [1].

The measured NC rating of an occupied space exceeded the specification by almost 10 points. There was a difference of opinion as to the cause of the additional noise. One party claimed that fan noise was the issue, while others suspected a change in duct configuration near the supply air (SA) and return air (RA) openings of the occupied spaces.

Cross-correlations were measured to determine the source location and relative strength. The transducer locations were selected to detect fan noise, as well as noise generated near the SA and RA openings (Figure 2). The measured cross-correlations are shown in figures 3 and 4 with time delay replaced by an (equivalent) length. The figures support the conclusion that the sound is generated near the duct opening. An up-grade to the fittings resolved the noise problem.

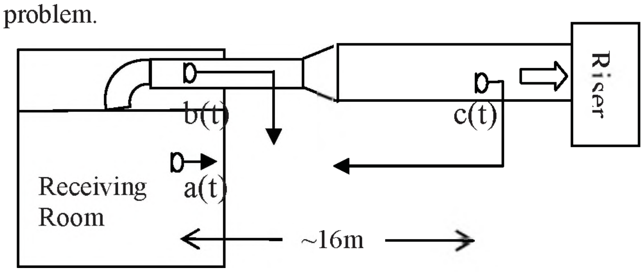

Figure 2. Cross-correlations of a(t),b(t),c(t) determine source location

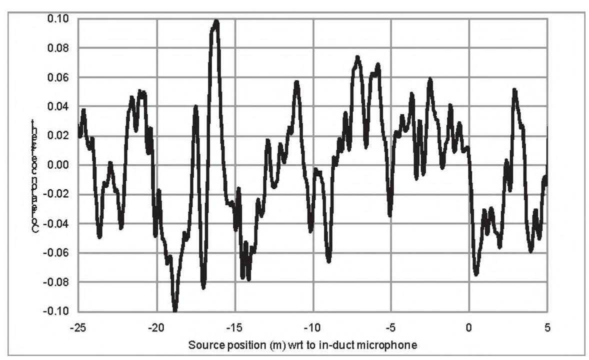

Figure 3, Cross-correlation Rac(t), low correlation indicates sound from riser is not a major source

3. LOW FREQUENCY RUMBLE

The most common acoustic deficiency in occupied spaces is an excess of low frequency noise (125Hz or 250 Hz octave band). Typically a reduction of the order of 10 dB is required. Review of narrow band data, measured at a large number of "problem sites" suggests that attenuation over at most 1/3 octave would suffice. Conventional dissipative silencers may provide the noise reduction needed. However, cost and space constraints are always of concern. Modern active noise control systems are possible alternatives. Cost, the need to provide electrical power, and maintenance are factors that make this solution less attractive.

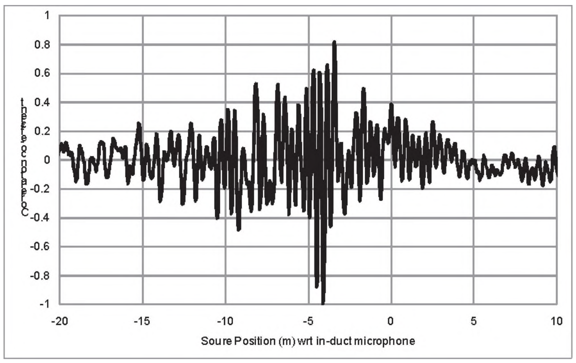

Figure 4. Cross-correlation Rab(t) shows high degree of correlation: source is located near microphone b.

4. HELMHOLTZ RESONATORS

A Helmholtz resonator may provide the relatively modest bandwidth and level reduction that is required. Helmholtz resonators require somewhat more careful design, but have been shown to provide good attenuation. Also manufacturers offer `tuned` sound absorbers that augment the dynamic insertion loss of fan system silencers. The absorber is tuned to the blade passage frequency.

It should be possible to install similar devices near SA and RA openings. A series of scale model tests was conducted by Aercoustics Engineering to obtain information on the performance of a Helmholtz resonator and an engineering design procedure.

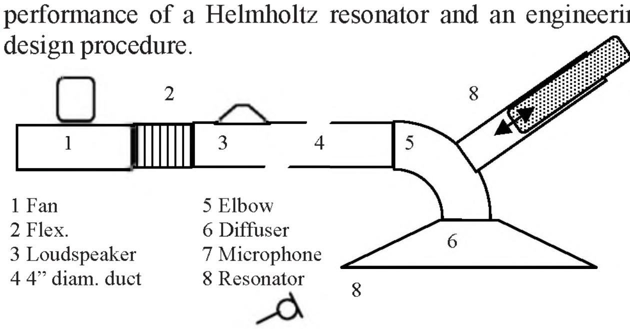

Figure 5. Schematic of measurement facility.

The experimental facility (Figure 4) was 10 cm diameter sheet metal duct. A small centrifugal fan delivered air to the duct, and a loudspeaker was the noise source. The duct was terminated with an elbow and a rectangular diffuser. Sound was measured 30 cm from the diffuser and processed by a dual channel FFT analyzer. A tunable Helmholtz resonator, 5 cm diameter ABS pipe, was installed on the elbow. A tight-fitting piston controlled the volume. The design procedure was based on the method outlined in reference [2]. The throat diameter (2.54 cm) and throat length (0.046cm) were kept constant. The change in sound pressure level afforded by the installation of resonator is interpreted as the effective transmission loss.

Measurements were conducted with and without air flow. The resonator performance was not affected by the flow, which had a nominal speed of 1000 FPM, which is representative of flows near RA and SA openings. The key parameters and performance of each configuration tested is summarized in the table below. The wavenumber ($ka=2\pi fa/c$) and diameter-wavelength ratio ($2a/\lambda$) are also listed so that the potential performance at full scale can be assessed.

Results Table

$f_{meas.}$ (Hz)	L (m)	L_{design} (m)	TL (dB)	ka	$2a/\beta$	$4L/\beta$
843	0.24	0.04	16.5	0.79	0.50	2.38
898	0.21	0.04	10.5	0.84	0.54	2.22
990	0.19	0.03	6.8	0.93	0.59	2.21
358	0.17	0.25	3.6	0.34	0.21	0.72
383	0.15	0.22	4.0	0.36	0.23	0.68
438	0.13	0.17	4.3	0.41	0.26	0.67
463	0.11	0.15	9.1	0.43	0.28	0.60
518	0.09	0.13	7.7	0.49	0.31	0.55
605	0.07	0.09	5.5	0.57	0.36	0.50
713	0.05	0.06	12.2	0.67	0.43	0.42
843	0.03	0.04	15.1	0.79	0.50	0.30
898	0.01	0.04	6.1	0.84	0.54	0.11

When the length of the resonator tube (L) is increased, the resonance frequency diminishes, until wave-effects become important ($4L/\lambda > 1$) Note that the resonator still functions, albeit at a totally different frequency. There is a discrepancy between the actual resonator volume (determined by L) and the design value. Further testing is required, with different throat diameters and throat lengths to establish a definitive design procedure.

5. SUMMARY

Cross-correlations are a useful diagnostic tools to determine source location and source strengths. Helmholtz resonators can provide substantial noise reductions. However, the standard design procedures require some empirical corrections.

6. REFERENCES

[1] Bendat, J.S., Piersol, A.G. *Engineering Applications of Correlation and Spectral Analysis*, Wiley Interscience, New York, 1980.

[2] Beranek, L. (ed.) *Noise and Vibration Control*, McGraw Hill, New York, 1971.