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Control of noise – systems for compact HVAC units

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Summary
This paper discusses noise control systems for implementation in compact HVAC units. The control of low-frequency noise presents different problems than at higher frequencies. This is mainly related to the long wavelength, which means that passive solutions require a significant volume of space, often not available in compact HVAC units. Active control can provide attenuation over a significant frequency range, including low frequencies, while requiring a more limited space. While the concept of active noise control is simple, a number of limitations in the acoustical, electrical and control systems affect the performance of implementations. The source pressure and the impedance of a centrifugal fan were measured, and a number of configurations for noise control were investigated. The performance of a simple analogue feedback control was tested in a physical prototype. An adaptive digital controller (feedback and feedforward using FXLMS with cancellation of feedback to reference) was simulated and implemented on a low-cost TI C55XX DSP platform.

1 Introduction
The concept of active noise control (ANC), the cancellation of sound by superimposing it with an electro-acoustically generated sound field of opposite phase, dates as far back as the early nineteen thirties with the independent works of Lueg and Coanda (Guicking, 1990), (Lueg, 1936) and (de Heering, 1993). Only much later has ANC been adopted in a wider range of applications.

This paper discusses the on-going work of exploring active noise control solutions for compact heating ventilation and air conditioning (HVAC) units. Both low-cost and more complicated solutions are investigated. In compact systems, only a limited space is available to apply noise control measures. Particularly at low frequencies, the long wavelength causes passive solutions to require a significant volume of space. Active control can provide attenuation over a significant frequency range, also at low frequencies, while requiring comparatively small volume of space.

For the present application, the space is of particular concern. The HVAC's that are the target systems contain "fresh air" and "return" flows and heat exchanging in one single (tightly packed) enclosure. This means that there is only room for one single loudspeaker driver, and special attention has to be paid to sound radiated from the back of the driver, which must either settle with a very small volume or be ducted to a location, where an appropriate volume can be offered. For active control, both an analogue feedback approach and an adaptive digital approach are explored. The systems outlet chambers may need a redesign to get the full benefit of the cancellation measures, or allow space for feedforward control. The noise radiated from these HVAC units is predominantly present in the flow. The noise is generated by the ventilation fans, but at low frequencies, turbulence on the inlet side of a heat exchanger is conjectured to be a significant contributor as well.

It is desired to be able to model the low-frequency response of the cancelling loudspeaker when positioned in various locations in the HVAC systems. This allows analysis of the volume flow required from the loudspeaker to cancel the noise from the source. The low-frequency performance of loudspeakers can be modelled well by simple lumped elements. It is considered that the coupling to a HVAC system that does not contain ducts, but rather small chambers connected by flow resistances, might be modelled correspondingly. This may facilitate design choices regarding the construction of new HVAC systems as well as optimizing the parameters of the loudspeaker.

The main difficulties with this are considered to lie in obtaining correct values of flow resistances, leaks and radiation impedances of complicated outlets. In addition, inherent noise due to turbulence and flow, not directly attributed to the acoustic properties of the source, cannot easily be accounted for. It is however necessary to know the properties of the primary noise source.

2 Method

Source properties

The fan that is currently used is preferred for its noise performance, rather than its efficiency. It is a centrifugal fan of 0.2 m diameter in its own housing and with forward-curved blades and a single inlet. It is not described by its manufacturer in terms of its acoustic source properties, other than likely being a pressure source, which is consistent with the claims in various literatures, though no references to measurements were found to substantiate the claim for a centrifugal fan. Though the fan is a two-port device, its one-port properties should be valid, when the impedance in the inlet is kept constant. Thus, the acoustical one-port properties of the source were measured, using both a direct and an indirect two-load method. The properties can be applied in a Thévenin equivalent circuit.

The source impedance and pressure should be measured at the operating point, at which the fan runs in a HVAC. Thus, measurements must occur in the presence of flow. There are numerous methods available; either a direct approach using an external source, or indirect methods based on applying multiple known loads to the source. Initially the indirect "Four-load" method (Prasad, 1987) seemed ideal, as it would allow measurements to be taken of the absolute pressure radiated from a duct end in anechoic conditions, which is outside of the direct flow in the duct. Also using four loads, only a single channel would be necessary, as no complex pressure measurements were required. It was, however, discovered that several authors have reported erroneous results from the four-load method. It was shown (Bodén, 1995) that this results from an ill-conditioned set of equations due to too little variation in the real part of the load impedance when only using different length ducts as loads. This means that, probably, only options that require complex pressure measurements directly in the flow, with all their associated problems, are applicable.

Instead, a direct approach using an external sound source and a two-microphone transfer function technique (Seybert, 1977) (Chung, 1980), while accounting for the different wave number of the incident and reflected waves due to the flow (Munjal, 1990) was applied. With this approach, the sound field in a duct connected to the source can be separated into incident and reflected waves, and accordingly the reflection coefficient and impedance of the source can be calculated. However only the source impedance can be found directly, while the source pressure must be obtained with a different procedure. Furthermore, the procedure is only applicable when the setup can accommodate an additional sound source that can produce higher levels than the source to be measured.

Measurements were made in a duct with 0.16 m diameter (cut-on at 628 Hz) with a total length of 3.67 m.To accommodate the frequency range from around 20 Hz to cut-on measurements had to be taken with three different distances between the two microphones. Distances of 0.12 m, 0.4 m and 1.62 m, giving a working frequency range of around 22 Hz to 2.4 kHz, were chosen. This is much higher than the cut-on frequency of the duct, but ensures sufficient overlap to cover the frequencies for which the method has non-unique solutions.

A simple two-load procedure, e.g. (Pande, 1982), can accommodate both the source impedance and pressure, yielding a simple set of two equations with two unknowns. However, due to the required operating point of the fan, it only allowed to use open-ended ducts as acoustic loads. Open-ended ducts exhibit complicated impedance, due to standing wave patterns resulting from a reflection at the open end. As a result, using two open-ended ducts as loads, the solution to the set of equations may be ill conditioned numerically. Add to this, the problems related to measuring in the presence of flow, and it is clear that the results should be considered carefully. As results could be compared to the direct method, over-determination was not used. Measurements were made using two ducts as loads (diameter 0.16 m, lengths 2.06 m and 3.67 m), while measurements were taken in three locations downstream (same distances as with the direct method measurements) to simultaneously evaluate the load impedance (for comparison to calculated load values).

Simultaneous measurements were taken in three channels using a 01dB HARMONIE system and Brüel & Kjær 4134 and 4133 microphones fitted to G.R.A.S. 26 AK preamplifiers. To reduce the influence of turbulence the microphones were fitted with half sphere windscreens (10 cm diameter) cut to fit the curvature of the duct. Further turbulence reduction was achieved by fitting the edge of the inlet duct with a curved foam lining. Time signals were recorded and transferred to MATLAB for analyses. As the two-load method requires the measurement of complex pressure, the fourth channel was used to capture the signal from the fans built-in tachometer.

The fan was kept at a rotational speed of approx. 1045 revolutions per minute, corresponding to a flow rate of 800 m^3/h (calculated based on pressure difference obtained from a Halton-cross inside the pipe). The external pressure difference was measured to be 150-156 Pa. Pressure differences were measured with a Testo 512 manometer. Results of measured source characteristics are shown in section 3.1.

Active control

For noise control to be effective at low frequencies in small spaces, an active approach is generally needed to obtain significant attenuation. A number of excellent textbooks are available on the subject e.g. (Nelson, 1992) and (Kuo, 1996). Both feedback and feedforward systems were explored, as well as a hybrid combining the two.

Initially the possibility to construct ANC based on an analogue feedback controller is explored, as it may offer systems that are considerably cheaper for volume products than DSP based solutions. Also, it may be fitted into systems where there is no space to obtain the appropriate a priori measure of the disturbance. Its performance (frequency range and peak attenuation) is however limited by the physical properties of the loudspeaker and microphone setup, as the system must abide by the Nyquist theorem to achieve stable operation. Also it seems feasible that an analog feedback control ANC may be extended by a digital controller as reported in e.g. (Carme, 1999). Thus an initial experiment was carried out, in order to evaluate the performance limitations incurred by the stability criterion. The transfer function from the disturbance to the error in a feedback controller, also denoted the sensitivity function, is given below, see e.g. (Elliott, 2001).

$$S(s) = \frac{E(s)}{D(s)} = \frac{1}{1 + C(s)G(s)}$$

E(s) denotes the error, D(s) the disturbance, C(s) the compensation filter, and G(s) the plant transfer function.

In our lab facilities a HVAC prototype was available. Due to the “return” and “fresh” air not being acoustically isolated from each other, measurements were taken inside the outlet chamber while dampers were fitted to the inlet. It was seen that the low-frequency part of the noise consisted not of tones, but there were significant contributions in the 40 Hz and 50 Hz third octave bands. Based on an impedance model of the loudspeaker in a closed box, it was found that the highest attenuation would be around the speaker’s system resonance when a low-order analogue compensation filter would be applied to ensure that the closed-loop gain is less than 1 at the Nyquist frequency. A closed box loudspeaker with a resonance at 54 Hz was available in our facilities. It consisted of a SEAS 33 F-WK in a 54 litre closed box. A Panasonic WM-61A electret condenser microphone cartridge was mounted as close to the speakers diaphragm as possible, while allowing maximum excursion. The system was positioned in the outlet chamber of the HVAC prototype and the transfer function G(z) measured. The delay incurred by the electronic signal path and the acoustic centre offset of the loudspeaker was evaluated by evaluating the phase at the resonance frequency, which was found to be 74 degrees. Under the assumption that the second order transfer characteristic of the closed box speaker system would have a phase of 90 degrees at resonance the delay was found to be 0.82 ms. This is quite high, and with the sound speed *c* equal 343 m/s, the distance to the acoustic centre of the loudspeaker is 0.28 m, which is far behind the diaphragm of the loudspeaker.

A simple compensation filter C(z) was implemented based on first order low-pass filter to obtain a stable system, with a phase margin of 45 degrees. The rather large phase shift of G(z) results in the frequency, at which the openloop gain should not exceed one was as low as 215 Hz. The physical data for the driver suggests that the large phase shift must be caused by the drivers voicecoil inductance which is rather high, measured to 3.4 mH. An RC compensation network was constructed to yield a perfectly flat impedance as function of frequency, but with virtually no change to the measured acoustic phase.

This limits the maximum open-loop gain that can be obtained with a simple first order compensation filter, and directly affects the obtainable attenuation. The stability criterion below the resonance frequency was also investigated, but not specifically accounted for in the filter design for this experiment, and did not appear to cause any stability issue. Analysis of the open-loop gain shows that a maximum attenuation of 13.9 dB is obtained at the 54 Hz resonance frequency. Attenuation decreases with a second order slope below resonance and a first order slope above resonance with the current filter. Positioning the system directly against another loudspeaker used as a sound source, and the microphone between the speakers, the attenuation matched the theoretical value at the resonance frequency closely. The results of positioning the system in the outlet chamber of the HVAC prototype are reported in section 3.2.

Digital adaptive controller

As HVAC systems are constructed to transport a fluid medium, they often allow the sound in a propagating wave field to be measured sufficiently upstream of a secondary source (cancelling loudspeaker) that a cancelling sound can be produced. This is approach is known as feedforward, where the generated sound is independent of the output of the cancelling loudspeaker. In practice, however, the sound produced with the cancelling loudspeaker will propagate both upstream and downstream. As a result, if the signal is measured upstream with a microphone (the case in most applications) it consists of the sound produced by the primary source, superimposed by sound from the cancelling loudspeaker. The cancelling sound is essentially being fed back. The severity of the problem depends on the physical setup, and for compact systems or short ducts it can limit overall performance. Numerous solutions to the problem exists. Some involve using several microphones or several loudspeakers, and thus with appropriate signal processing, creating a directivity that limits the upstream propagation as measured or the upstream propagation itself. Other approaches involve subtracting the cancellation sound from the measured signal in the signal processing chain. This requires knowledge of the transferfunction from the cancelling loudspeaker to the reference microphone. Also the choice of algorithm for the controller may help reduce the problem, as those based on an IIR filter structure may cope better with the feedback than those based on an FIR filter (Romeu, 2001). The IIR structure is however not inherently stable, and additional checks for stability may be necessary. For compact systems a purely signal processing based approach may be prefered or even necessary.

Experiments were carried out with a single channel digital controller, where both an adaptive feedforward and adaptive feedback can be used seperately or together (a hybrid), as desribed in (Kuo, 1996). The controller is based on the FXLMS algorithm for both the feedforward and feedback structures. As the acoustic feedback to the reference microphone is a relevant issue in short ducts, It is attempted to account for this by subtracting the produced cancellation noise convolved with the transfer function from the loudspeaker to the reference microphone. Also, the shape of the spectrum of the target systems is dominated by low-frequency noise, that does not necessarily dominate the A-weighted noise, thus it is desired to implement some degree of control of the residual noise spectrum. This can be obtained in a number of ways that are well described, e.g. FELMS (Kuo, 1996), the variants Adjoint LMS and the Secondary Path Equalization method and HFELMS (DeBrunner, 2006). Due to impracticalities with inverse modelling of the error path on a DSP or long FIR filters, a standard FELMS approch was used, where the noise shaping filters on the feedforward structure employ an IIR (Direct-Form I) structure. A block diagram of the simulated controller, (without error filtering on each side of the LMS algorithm that updates A(z)) is shown in Figure 1.

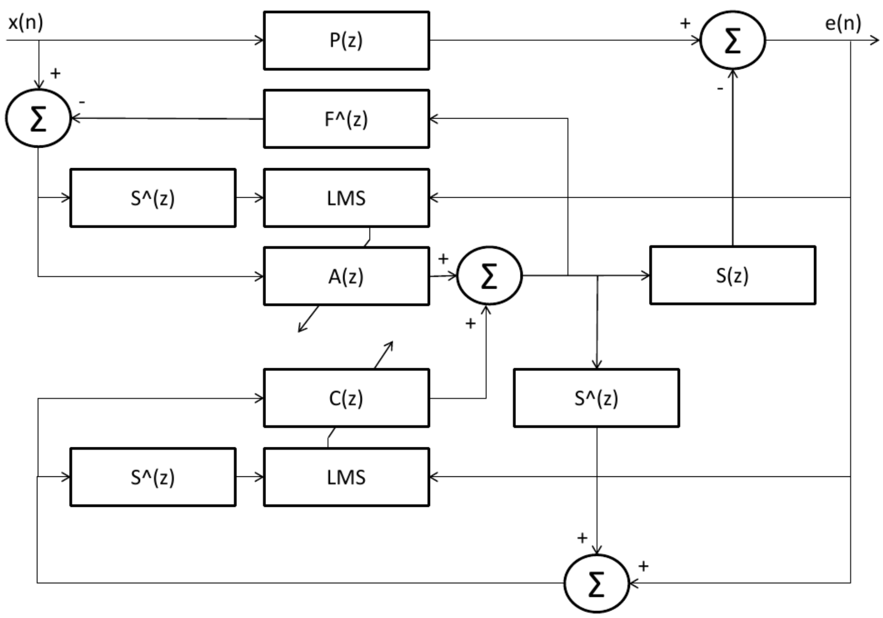

Figure 1: Block diagram of hybrid controller based on the FXLMS algorithm. The acoustic transfer path of the system is, denoted "plant", is modelled by P(z). The electroacoustic transfer function of the cancellation loudspeaker and associated electronic equipment and acoustic transfer path to the error microphone, denoted "error path" is modelled by S(z). The remaining blocks are run in signal processing on a DSP. S^(z) is a model of S(z). F^(z) is a model of the acoustic feedback to the reference microphone. A(z) is the feedforward control filter. C(z) is the feedback control filter.

The system should be implemented on DSP hardware that was preferably not too costly. The performance of a Texas Instruments C55XX DSP was evaluated, and it was found that the system should be implementable on that platform using a 4.8 kHz sampling frequency and filter lengths of 128 taps. The delays in the codecs (with associated linear phase filters) and resampling filters amount to 2.3 ms, thus it is required to capture the primary noise at least approx 80 cm upstream of the cancelling loudspeaker for feedforward control of random noise. It is possible to reduce the delay, at the cost of filter lengths. The controller was simulated in MATLAB using Q15 number format, including a leaking mechanism on the controller's tap weights. The physical transfer characteristics of a test setup are measured as impulse responses and included in the model.

The performance of the adaptive digital controller has been evaluated using primary noises consisting either of tones or of shaped random noises. Acoustic transfer functions were measured in a test setup. It consisted of a metal duct (undamped), connected to a fan and a source loudspeaker at one end, and a cancelling loudspeaker was connected through a T-junction near the radiating end of the duct. The reference microphone to measure the primary noise was positioned near the source with 1 m distance to the cancelling loudspeaker, while an error microphone was positioned 0.2 m equidistant to the cancelling loudspeaker and the duct end. The transfer functions of the plant, P(z), error path, S(z), and the feedback path F(z) were measured. Due to the undamped duct the impulse responses exhibit a long ringing at higher frequencies. In the simulation, the models S^(z) and F^(z) are obtained using system identification of the measured S(z) and F(z) applying a half Hanning window before truncating to the available 128 filter taps. Thus, the transfer characteristics of the test setup represent a difficult setting for the controller.

The primary noise is a white noise signal with a low-pass filter applied to attenuate signals above the duct cut-on. Simulations are shown without the use of the residual noise shaping function or coefficient leakage. The performance of the simulated controller is given in section 3.3.

3 Preliminary results and discussion

3.1 Source Characteristics

The one-port source characteristics were evaluated using three different length duct (diameter of 0.2 m): 0.7 m, 1.3 m and 2 m. It was found that the length of the inlet duct would influence the source characteristics, but the general observations were the same. Results for a 0.7 m duct on the inlet are given in Figure 2 and Figure 3, using respectively the direct method and the indirect two-load method. For comparison, the results for a 2 m duct on the inlet measured with the indirect two-load method are given in Figure 4.

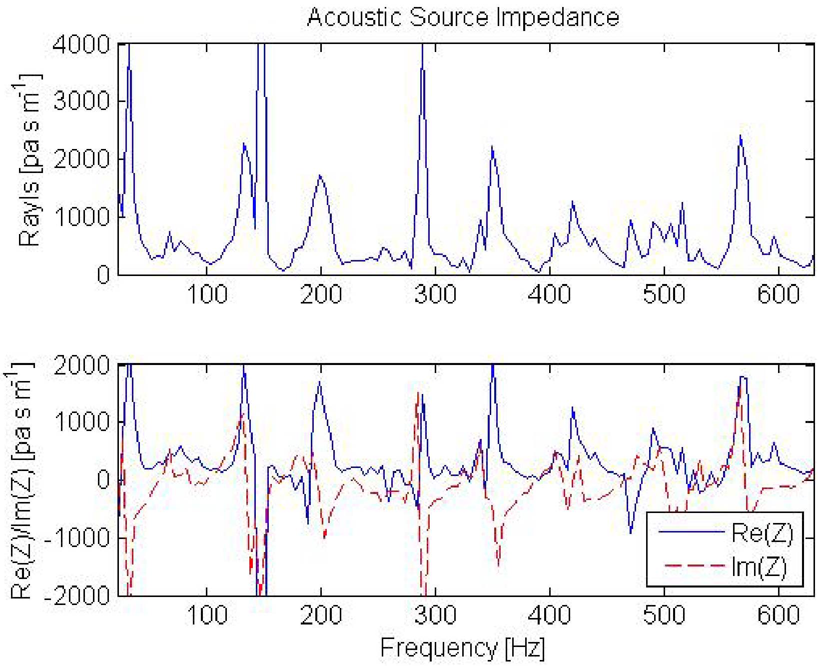

Figure 2: Measured one-port source impedance of centrifugal flow fan. Direct measurement. 0.7 m inlet duct. Top: Absolute impedance. Bottom: Real and imaginary parts of impedance.

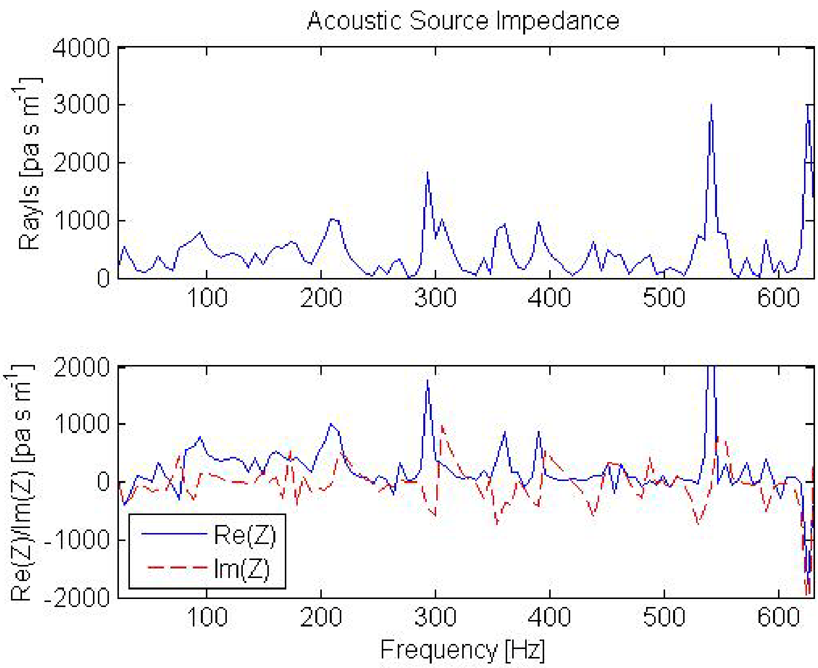

Figure 3: Measured one-port source impedance of centrifugal flow fan. Indirect two-load measurement. 0.7 m inlet duct. Top: Absolute impedance. Bottom: Real and imaginary parts of impedance.

Generally, the source impedance is found to be low, around a few times the specific acoustic impedance of air. However, the impedance is quite irregular and large spikes appear with a seemingly fixed frequency interval. This looks suspiciously like the impedance of a finite length duct. However, similar structures are seen on source impedance measured in the exhaust system on internal combustion engines e.g. (Peat, 2002). Some of the spikes are quite large while others appear to average over a larger range. Thus it is difficult to estimate the exact frequency interval at which the spikes occur. The interval appears to be approx 90 Hz for the 0.7 m inlet duct, at which half a wavelength does not coincide *exactly* with any of the duct lenghts or sum of duct lengths in the setups.

The direct measurement approach appears to result in the spikes being of higher value than which the indirect two-load method, in which the structure of the impedance is less pronounced, particularly at low frequencies. At the lowest frequencies, a large spike is seen, which if correct, is rather underestimated by the indirect two-load method.

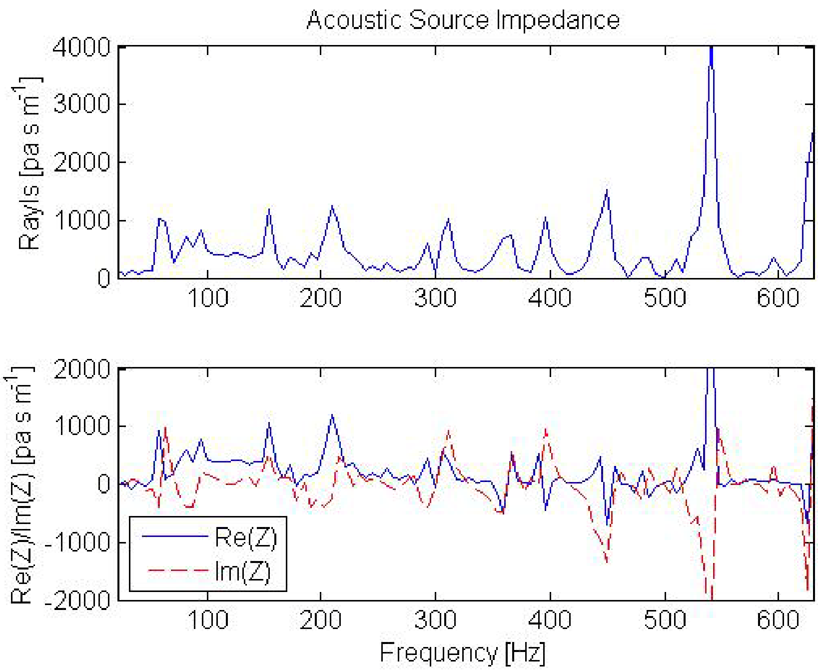

Figure 4: Measured one-port source impedance of centrifugal flow fan. Indirect two-load measurement. 2 m inlet duct. Top: Absolute impedance. Bottom: Real and imaginary parts of impedance.

For the 2 m inlet duct the same general result is seen. The frequency difference between the spikes is difficult to read precisely, but appears to be similar to the 0.7 m inlet duct. An interesting result is the impedance at the lowest frequencies, which does not show a pronounced peak. This result is confirmed by a direct measurement using a 2 m inlet duct (not shown). This difference at the lowest frequencies is also apparent in the resulting source pressure shown in Figure 5, where the pressure is significantly lower below about 40 Hz for the 2 m inlet duct.

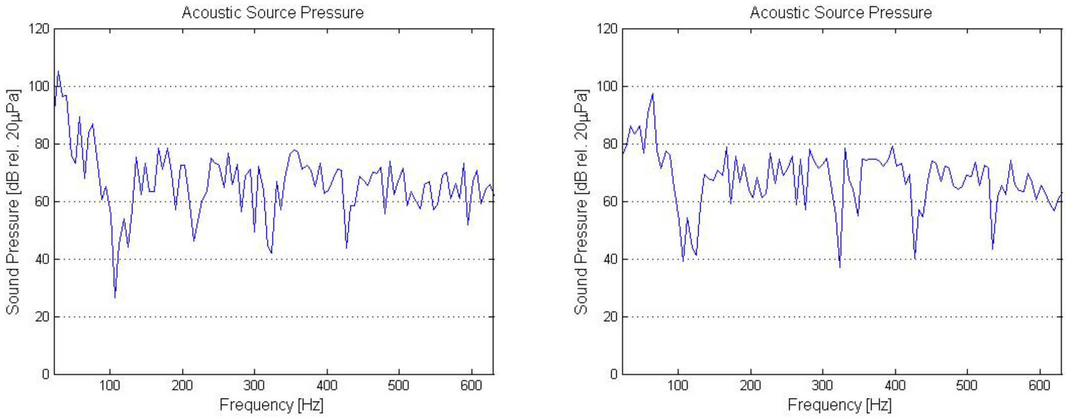

Figure 5: Measured one-port source pressure of centrifugal flow fan. Indirect two-load measurement. Left: 0.7 m inlet duct. Right: 2 m inlet duct.

3.2 Analogue feedback ANC

An analogue feedback ANC was setup in a HVAC prototype to evaluate if the theoretical attenuation could also be observed in practice. The measured attenuation is shown in Figure 6. The HVAC prototype was located in a reverberation chamber, and the inlets and outlets were in the same room, making meaningful measures of the radiated noise difficult. Thus, dampers were fitted on the inlet while the outlet was shielded of and measurements taken inside the outlet chamber. The theoretical peak performance at resonance (54 Hz) of 13.9 dB is not quite observed, as the disturbance is not tonal. At the Nyquist frequencies below and above the resonance, amplification is seen. This is described as the “waterbed effect” (Elliott, 2001), which occurs for non-minimum phase systems.

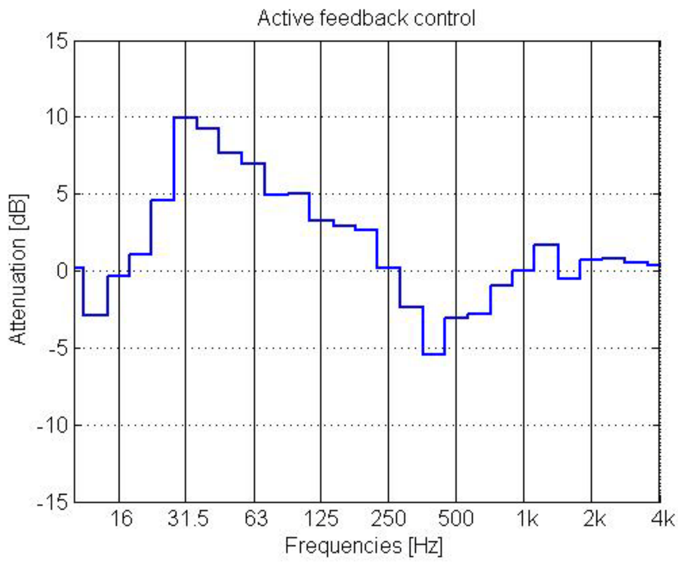

Figure 6: Measured attenuation of analgue feedback control. HVAC system located in a reverberation chamber, thus the measurements were taken inside the outlet chamber adjacent to the cancelling loudspeaker. Theoretical attenuation is 13.9 dB at 54 Hz.

The attenuation is rather low, and a reduction in the acoustic delay is required to allow increase of the open-loop gain and thereby the theorecital attenuation. The acoustic centre concept is described further by (Vanderkooy, 2006) and

(Vanderkooy, 2007) in terms of flow-lines. Interestingly it is found to be dependent on baffle dimensions, and it is found that the sound wave emanates from a point in front of the driver. Thus the large centre offset found in the test setup is likely to be related mainly to the phase response of the speaker, dominated by either the moving mass or voicecoil inductance (in this case voicecoil inductance).

A number of drivers were checked to evaluate if an analogue feedback could be feasible to pursue further. It was found that particularly drivers with a copper sleeve in the magnet system, to short eddy currents, may have very low voicecoil inductance and the associated phase shift becomes small. The acoustic centre was evaluated for a non-standard 7 inch driver (made by Scan-Speak) with a voice coil inductance of 0.4 mH. In a 22 litre closed cabinet the resonance frequency was found to be 61 Hz (factory supplied Thiele/Small parameters predicts 51 Hz) and the acoustic phase 88 degrees, measured 1 cm from the centre of the (inverted) dust cap. Given the mentioned assumptions, the distance from the microphone to the acoustic centre at resonance is found to be just 0.963 ms, or 3.3 cm from the microphone. The point where the acoustic phase is -45 degrees in G(z) is at 896 Hz. Thus the peak attenuanuation obtained with the same filter topology on this driver, at resonance, would be 23.9 dB.

The performance of the analogue feedback controller may be improved further by a more sophisticated compensation filter. Particularly, a second-order biquad filter is reported to have desirable characteristics when the poles and zeros are arranged in complex conjugate pairs (Nelson, 1992), which allows to boost the open-loop gain further, while avoiding excessive phase shifts at high frequencies.

3.3 Adaptive digital ANC

Simulations of the adaptive digital hybrid controller are made using measured transfer functions of the physical test setup. The convergence and filter coefficients of the feedforward (A(z)) and feedback (C(z)) structures are shown for a 10 second run in Figure 7.

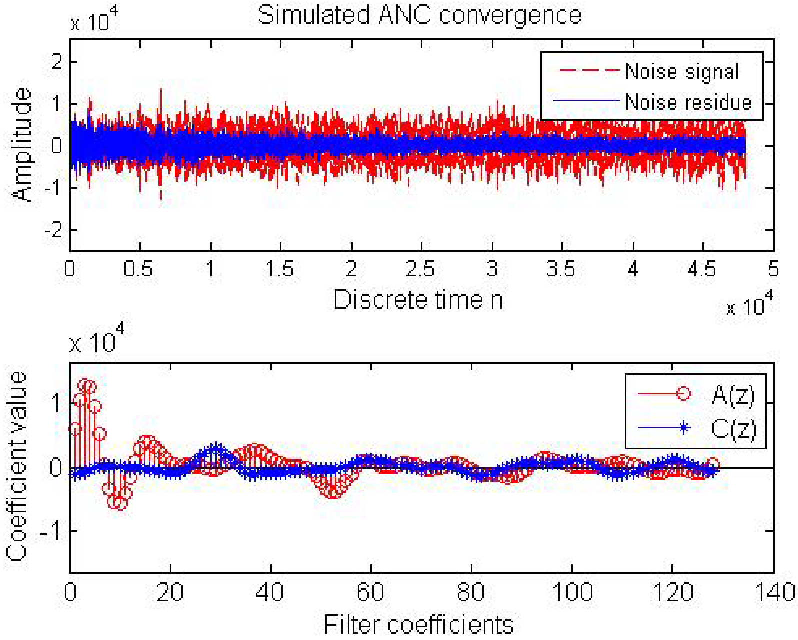

Figure 7: Simulated digital hybrid controller. Top: Signal amplitude of primary noise at error microphone for 10 s duration (blue: without ANC, red: with ANC). Bottom: Filter coefficients after convergence (feedforward structure A(z) and feedback structure C(z)).

The spectrum of primary noise at the error microphone with and without ANC after a 100 second run is shown in Figure 8. The attenuation (difference between the summation of third octave bands with and without ANC) amounts to 16 dB.

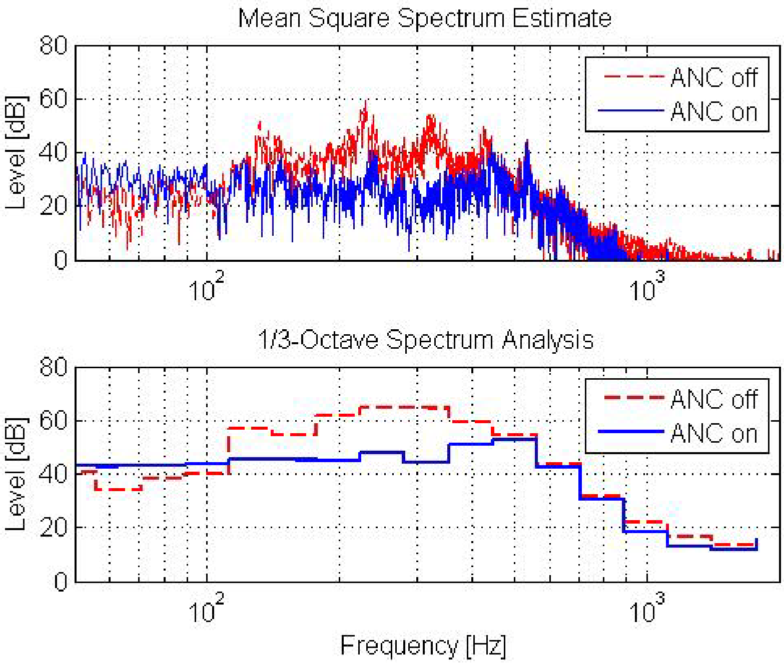

Figure 8: Simulated digital hybrid controller. Top: Mean square spectrum estimate of primary noise at error microphone (blue: without ANC, red: with ANC). Bottom: Third octave analysis of primary noise at error microphone (blue: without ANC, red: with ANC).

The hybrid control algorithm is implemented on a TI C5505 DSP platform. In the physical test setup however the performance is poorer than simulated, and generally attenuation of broadband random noise appears to be limited to around 6 dB. Deterministic signals however fare much better, as single pure tones can be attenuated by as much as 60 dB in the setup. The performance of the feedforward control is entirely dependent on the coherence (γ^2) between the reference signal and the noise measured at the error microphone. A measure of the cross coherence is shown in Figure 9.

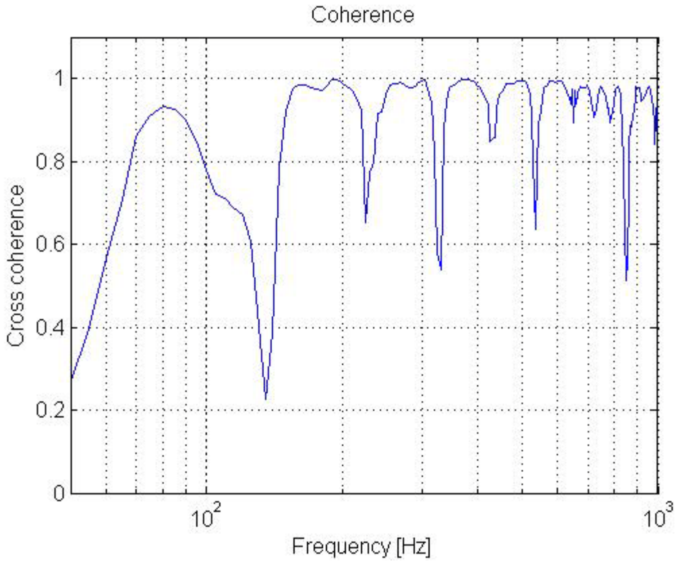

Figure 9: Measured cross coherence between reference and error signals.

The coherence function exhibits narrow dips at a fixed frequency interval of 90 Hz. As the maximum attenuation that may be obtained is given by $10\log_{10}(1-\gamma^2)$, the dips in the coherence function reveal that broadband attenuation is limited by the coherence. At 90 Hz, half a wavelength coincides exactly with the distance from the reference microphone to the radiating duct end. Thus, a reflection from the termination is the likely culprit. Indeed moving a microphone inside the duct while playing a tone at either of the frequencies at which the dips occur shows that up to 10 dB lower levels occur at the position of the reference microphone at those frequencies.

4 Conclusions and further development

In the preliminary work on implementing ANC in compact HVAC systems the investigation of the source impedance of a low-noise centrifugal fan has shown it to be a pressure source with rather low impedance, though with high spikes at certain frequencies. At the lowest frequencies, the spikes appear to be underestimated when using the indirect two-load method compared to the direct two-microphone transfer function method. Also, at the lowest frequencies (below 40 Hz), the source pressure is found to drop significantly when a 2 m inlet duct is used instead of a 0.7 m inlet duct. This is also reflected by a drop in the source impedance. Fans of different construction that have higher efficiency, but are noisier, are subject for further investigation.

For active control the tested analogue feedback using a very simple compensation filter was found to behave close to the expectations in the outlet chamber. An analogue feedback design should have many of the attributes to make it a good option in a compact system as no propagation distance is required to obtain an a priori measure of the disturbance. In the test setup the attenuation obtained with a simple compensation filter was rather low, and a reduction in the acoustic delay required to be able to increase the open-loop gain. However, it is only after finding woofers with very low phase shift that it has become interesting to pursue the option further. It is found that a Nyquist frequency above 1 kHz can be achieved. Further developments will comprise a compensator based on an optimized biquad filter.

An adaptive digital controller using both feedforward and feedback was implemented on a TI C55XX DSP platform. In the test setup using metal ducts of diameter 0.16 m pure tones can be attenuated up to 60 dB. The attenuation of random noise is however much more limited as only 6 dB is obtained on wideband noise, while 10 dB is obtained when band limited to 100-500 Hz. Measurements indicate that standing waves are at least part of the problem, and that these influence the cross coherence measured between the reference and error locations. Though measures are taken to solve the issue of acoustic feedback to the reference microphone, further developments will involve improving the reference signal by measuring in more locations near the source, as well as reducing flow induced noise by cutting a slit in the duct wall, and moving the microphone out of the flow.

Finally, a hybrid comprising both an analogue feedback and an adaptive digital controller is to be tested, e.g. similar to the configuration reported in (Carme, 1999).

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