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Summary Rapid advances in adaptive signal processing are making active noise control products a practicable proposition. One unit available in the industry was chosen in this study to evaluate its application and effectiveness in controlling noise in ventilation systems. Its performance with aerodynamic and static noises was evaluated against the conventional duct lining approach. Experimental results indicate that the system is applicable in air-conditioning systems despite the existence of some constraints that have to be taken care of. It is proposed to use the equipment to supplement the traditional dissipative silencer for noise level control and for the achievement of spectrum balance.

Active noise control: Evaluation in ventilation systems

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1 Introduction

1.1 *Low-frequency noise in ventilation systems*

Reduction of low frequency noise in ventilation systems has been a problem area for engineers. The traditional passive type silencers, making use of dissipative or reactive effects, are usually effective in the mid to high frequency range which will lead to a distorted frequency spectrum resulting in an undesirable humming noise. Low frequency noise reduction is either ineffective or made marginally possible through the employment of large and bulky absorptive devices which in many occasions results in accommodation difficulties.

To tackle the problem, attention is diverted to some other non-traditional means of noise reduction amongst which is active noise control. The fundamental principle is simple: superimposition of an antiphase copy of the unwanted noise on the unwanted noise itself to cause destructive interference. Four basic components have to be included: a noise sensing primary microphone to pick up the unwanted signal; a device to produce an antiphase noise of equal magnitude for superimposition; a loudspeaker for generating a cancellation source; and an error microphone for monitoring and feedback. However, there are numerous technical problems and it is not until recent years that commercial products have become available.

1.2 *Active duct noise control*

Superimposition is one of the physical phenomena that makes active noise control feasible. To achieve global attenuation, an acoustic impedance coupling between the active and unwanted noise sources is another important criterion. Coupling is considered effective when the two sources are separated by less than about one-third of a wavelength or are connected by a waveguide(1).

A brief review of waveguide acoustics reveals that impedance coupling is also the mechanism involved in sound power reduction. Ventilation ductwork can be considered acoustically as a plane waveguide for frequencies with wavelengths at least twice the longest dimension of the cross-sectional area of the duct. Higher frequency noise tends to bounce off the walls resulting in a non-uniform acoustic pressure distribution associated with these higher-order propagating modes(2). Active sound attenuation in ducts is generally limited to low frequencies with the higher order dispersive cross-modes taken care of by passive means.

The development of active noise control from the original idea to the practical applications in industry has been presented by Swanson(1) and Eriksson(3). Efforts have been made to solve the technical problems encountered:

— feedback of the cancelling noise to the primary microphone resulting in contamination and instability;

— changes in transfer functions between the active loudspeaker and the microphones resulting from changes in acoustics of duct systems, which include, for example, a change in temperature, air flow rate and loudspeaker response;

— and lack of of signal coherence at the primary and error microphones due to turbulent flow pressure fluctuations adding uncorrelated noise to the acoustic pressures.

Extension of the effective frequency range beyond the cut off frequency of duct plane waves is also receiving attention(4,5).

1.3 *Objective of the studies*

The success of an active noise control system depends greatly on its ability to carry out on-line system modelling effectively and efficiently to adapt to the continuously changing environment without frequent manual recalibration(6). Earlier work using analogue electronics for signal processing was only effective for highly controlled and steady conditions. Following recent advances in adaptive signal processing theory and hardware, a few commercial products making use of the basic principles are available. One of them with application in fan/duct systems is identified in this paper and is evaluated for suitability of local use.

The study aims at evaluating the effectiveness and applications of this active sound cancellation system in tackling the low frequency noise problem in ventilation systems. It will be evaluated against the traditional duct lining approach. Its ability to deal with constant air and variable air volume air conditioning systems will be studied. Its success will be measured against initial and running costs. Particular attention will be focussed on situations commonly encountered in Hong Kong.

2 The commercial product

The digital sound cancellation system consists of four basic elements: a primary noise sensing microphone, an error microphone, a controller/amplifier and a speaker. No technical detail about its operating principles is included in the catalogue or manuals supplied. However, it is claimed to be fully automatic, requiring no pre-programming, training or manual calibration. It can be used simply by connecting up the system components and pressing a switch. Guidelines are supplied for the setting up of components and their relative positions. The fully automatic start up procedure will adjust for duct sound levels and system characteristics after which the controller will compensate for any changes in temperature, airflow, fan speed, microphone sensitivity or speaker output. The proper functioning of components is revealed through indicating lights[7]. Auto-calibration noise is audible on system start-up.

The hardware configuration of the system[8] will not be discussed in this paper; a brief description of its operating principles is provided. As shown in the simplified circuit in Figure 1, an infinite impulse response (IIR) adaptive filter is used to model the acoustical system while compensating for acoustic feedback. As the acoustic feedback introduces resonance or poles into the response of the system, a single continuous adaptive model of a pole-zero form can be used to remove the effects of the feedback poles and to model the direct plant simultaneously using a single error signal. The recursive least mean square (RLMS) algorithm of Feintuch[9] is adopted for this purpose. The effects of transducers and filters are also included.

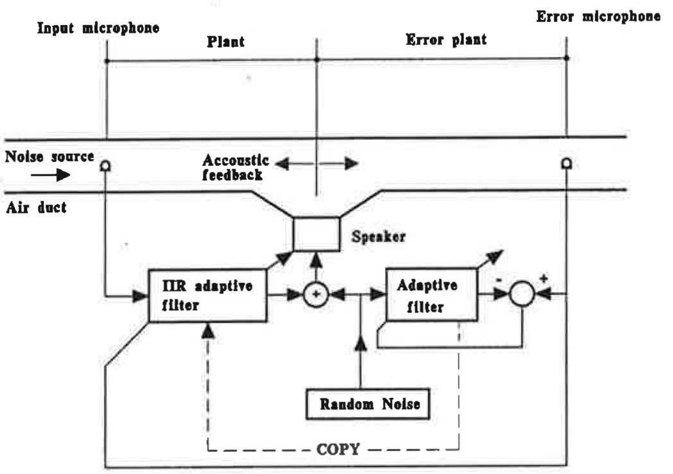

Figure 1 Active noise control using iir adaptive filter for feedback compensation and random noise source with adaptive filter for on-line auxiliary and error path modelling

In addition to the time-varying characteristics of the direct and feedback paths, the acoustical characteristics of the loudspeaker, error path and the error microphone have to be compensated to ensure convergence of the RLMS algorithm for a minimum error signal. An independent modelling process is used to model these transfer functions continuously on line. This is achieved with the aid of an independent random noise source and a second adaptive filter based on the LMS algorithm to provide the required input correlators for the RLMS algorithm. Variation in temperature, flow and other characteristics of the acoustic system such as ageing effects can then be handled.

Fluid flow in a duct generates undesired turbulent flow pressure fluctuations which will superimpose on the acoustic pressure fluctuations. As the two types of pressure fluctuations are measured simultaneously by the two in-duct microphones, the former will add uncorrelated noise to the signals resulting in the reduction in signal coherence at the primary and error microphones. To tackle this problem, anti-turbulence microphone probes are used[10]. Spatial averaging of the turbulent flow pressure fluctuations is effectively carried out at the probes and signal coherence at the two microphones is greatly improved.

As described by the manufacturer, this sound cancellation system is effective in the frequency range of 20 through 400 Hz for fan applications. Objectionable pure tones in a broadband sound spectrum will be eliminated first followed by the broadband noise. Reduction of 20 to 30 dB for pure tones and 10 dB for random broadband noise is reported. For spectra with no pure tone, broadband sound reduction of up to 20 dB can be achieved. Several successful practical installations have been accomplished[11–16].

3 Experimental studies

An experimental set up was devised to study the performance of the cancellation system and to evaluate its effectiveness against the traditional duct lining approach. As shown in Figures 2 and 3, two pieces of 1mm thick aluminium circular duct were used. One of them was bare and with the sound cancellation system connected while the other was internally lined with 25 mm thick 48 kg m^{-3} fibreglass insulation. A net internal diameter of 300 mm was maintained for both ducts so as to keep approximately the same in-duct air velocity.

A centrifugal fan equipped with a frequency inverter for speed/flow variation was used to provide an aerodynamic broadband noise while a noise/signal generator was employed to supply a static broadband noise as well as pure tones to the ductwork. The sound measuring system consisted of a wind screened $^1/_2$" (13 mm) microphone fitted with a preamplifier to a measuring amplifier and bandpass filter for measurement in $^1/_3$ octave bands at the exit end of the duct sections. For better suppression of turbulence due to high air flow velocity, a turbulence screen should be used and this is preferred to a noise cone. The microphone was inserted inwards to the duct ends to minimize the effect of background noise. Average values across the duct cross sections were obtained. The in-duct air volume flow rate was measured with the aid of a pitot-static tube and a manometer. Similarly, mean values were recorded.

The active noise control system was installed and connected following the manufacturer's recommendations as far as possible[17]. A buffer distance varying from a minimum of 0 m to a maximum of 2.7 m has to be maintained in front of the primary input microphone probe depending on the turbulence condition. A straight duct section is preferred for less turbulence. A cancellation distance varying from 1.2 m to 6 m between the primary microphone and the speaker is suggested, based on the delay required in signal processing. An error distance varying from 0.5 m to 3 m between the speaker and the error microphone is recommended. Under normal situations, the suggested straight duct section required for the system is around 4 m and it can take up a maximum length of 6 m in conditions of heavy turbulence. In the study, 0.6 m, 1.6m, and 0.8 m were chosen to be the three respective distances.

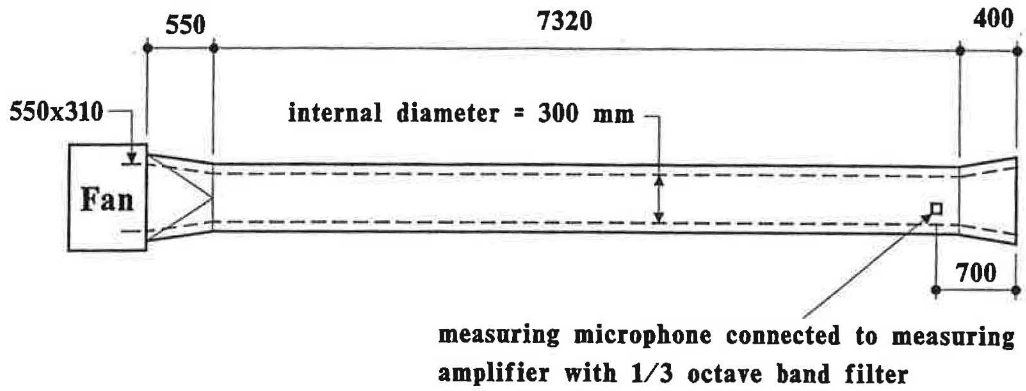

Figure 2 Test duct with 25 mm thick internal lining

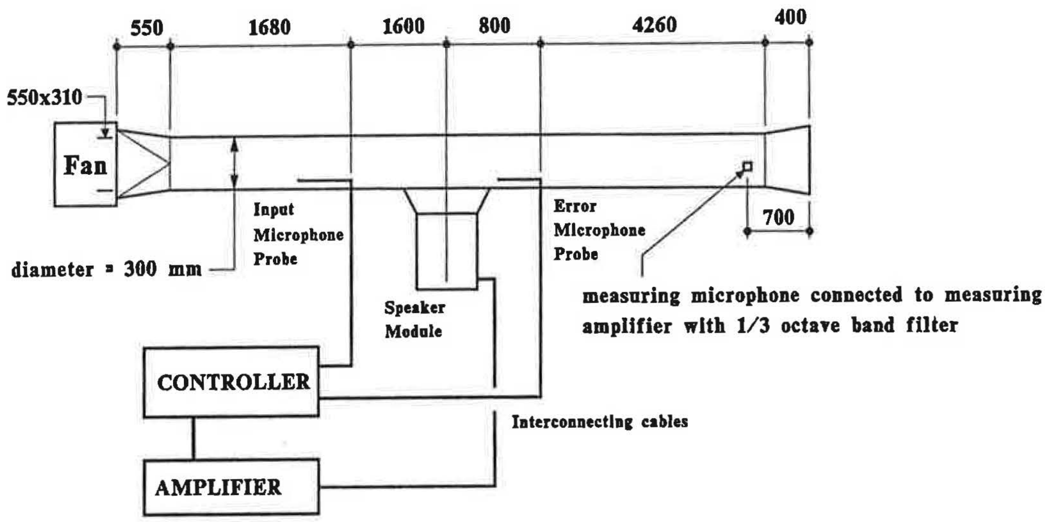

Figure 3 Test duct with digitial sound cancellation system

The study was divided into two stages: with and without a fan connected to the two test pieces. In the first stage, a fan was used to produce aerodynamic noises to the duct sections at three different fan speeds/volume flow rates. In-duct sound spectra from 16 Hz to 4000 Hz in $^1/_3$ octave band intervals were obtained with/without operation of the fan, and with/without the sound cancellation system functioning . Aerodynamic sound spectra were observed during the processes of increasing and decreasing volume flow rates to simulate the operation of a variable air volume system in which the supply air flow rate to an occupied space is varied according to its cooling/heating load requirements in partial load conditions.

In the second stage, the effects of static noise were studied. Pink noise with an overall sound pressure level comparable to that of the fan at full speed was generated to test the performance under no-flow conditions. The low frequency enriched pink noise was chosen to resemble the actual sound spectrum encountered in air conditioning systems. System behaviour under selected pure tones was also analysed.

All the equipment was calibrated before taking measurement and the sound cancellation system was tested for its proper operation. It was recalibrated at the end of the study to verify its reliability.

4 Results

4.1 Effects of background noise

The relative values of background noise, aerodynamic noise at full fan speed, and pink noise injection inside the bare duct are shown in Figure 4. The effect of background noise on the useful data was negligible except at very low frequency below 31.5 Hz for the static noise investigation and this has to be taken into consideration in later analysis. This also reflects the inefficiency of the sound production system to generate low-frequency noise.

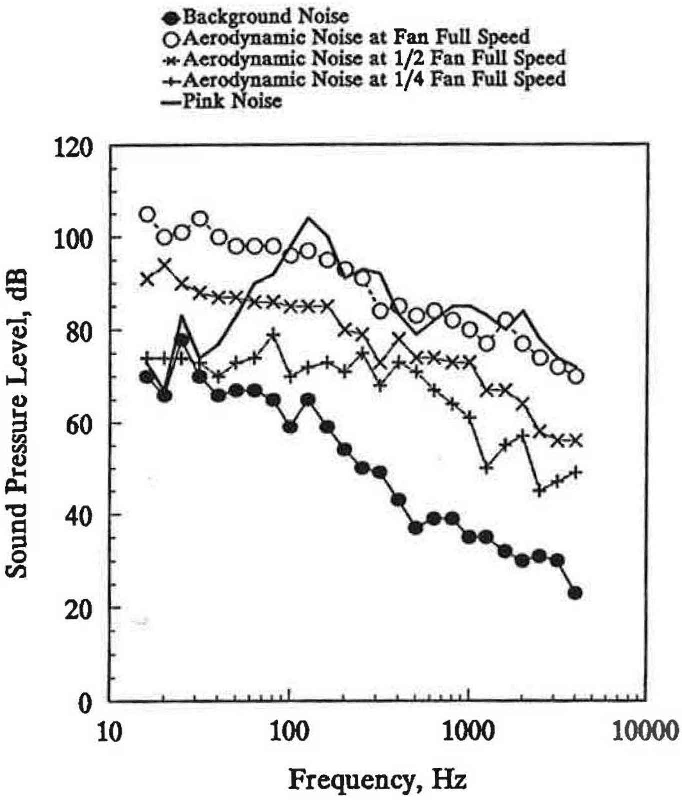

Figure 4 In-duct sound spectra without treatment

4.2 Effect on broadband aerodynamic noise

With the fan at full speed, the in-duct air flow velocity was 25 m s^{-1} and the noise reduction performances for duct liner and the sound cancellation system are shown in Figure 5. The noise spectrum produced by the fan was typical and was low frequency dominated with a few strong tones at 31.5 Hz and 1600 Hz. Duct lining was more effective nearly throughout the whole frequency range of interest. It also agrees with the long established theory of better attenuation for mid to high frequency sound. The relatively lower attenuation in the presence of air flow may be due to regeneration at the surfaces of the liner. On the other hand, it seems that the digitial sound cancellation system was not so effective in cancelling the broadband background noise. Instead, there was relatively more notable sound reduction at 31.5 Hz, 80 Hz and 1600 Hz which coincided with the peaks that existed in the fan sound spectrum. The corresponding maximum noise reduction was around 5 dB.

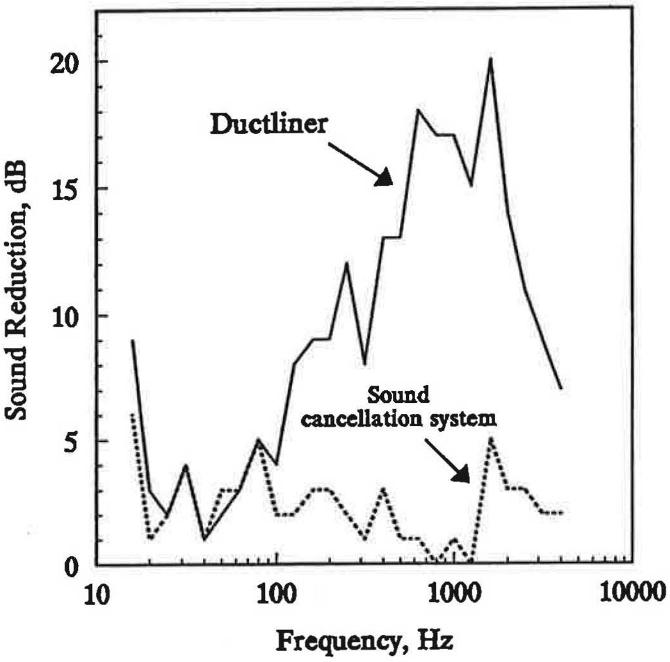

Figure 5 Sound attenuation at full fan speed

Similar results were obtained for 50% fan speed as shown in Figure 6 and the average noise reduction throughout the whole frequency range was only a few dBs. The in-duct air velocity was 14 m s^{-1}. As the fan speed was changed, the system underwent a re-calibration process and a louder note was noticed momentarily. It took around half a minute for the system to stabilise.

With the fan at 25% speed for an in-duct air velocity of 7 m s^{-1}, a completely different picture was obtained. As revealed in Figure 7, there was a drastic increase in sound cancellation with a peak reduction of 21 dB at 400 Hz. Another effective region was at 160 Hz with 20 dB reduction. At 2000 Hz, there was another relatively large cancellation of 13 dB. These generally fell in line with the peaks that existed in the fan spectrum. At around 80 Hz where there was a maximum untreated noise level, a rapid increase in sound cancellation was observed. An average achievement of 10 dB was obtained for the broadband noise excluding the peaks. The performance of the sound cancellation system was more effective than the ductliner up to 250 Hz. Its overall performance also dropped with frequencies greater than 630 Hz.

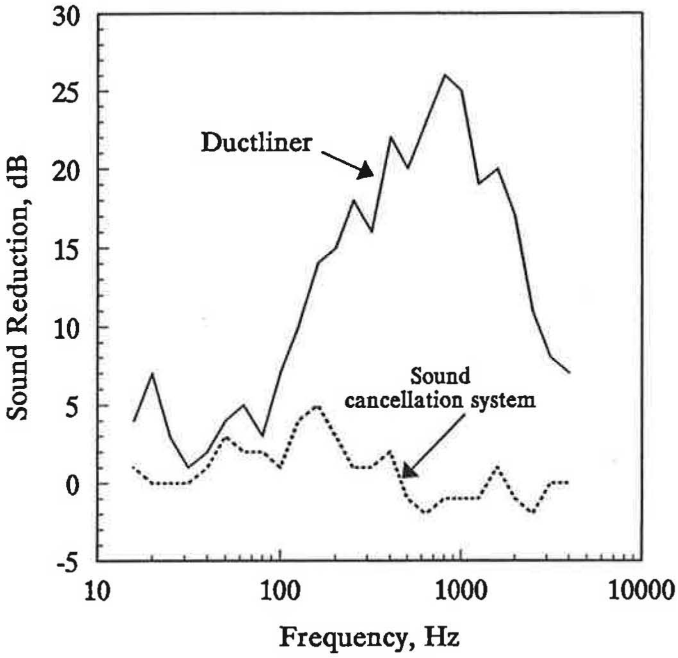

Figure 6 Sound reduction at 50% fan speed

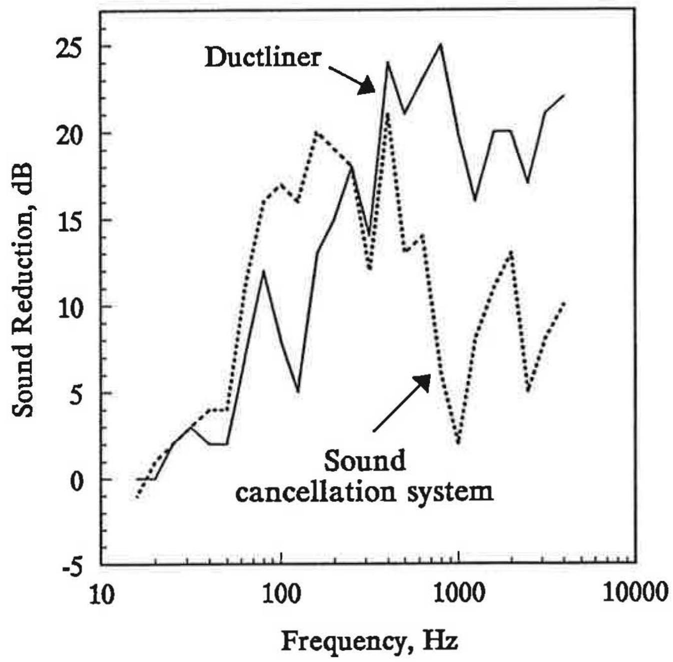

Figure 7 Sound reduction at 25% fan speed

Comparison of the results with different air flow velocities indicated that turbulence might be the factor that affect the attenuation. Lack of coherence of signals at the microphones resulted in degradation of performance. Nevertheless, under normal situations, the maximum recommended velocity to be used for this set-up is 7.5 m s^{-1}.

4.3 Effect on pink noise

The performance of the noise treatments are shown in Figure 8. Both the ductliner and the sound cancellation system exhibited insignificant achievement for low frequency noise up to 63 Hz after which there was a rapid increase for the ductliner until a maximum of around 50 dB reduction was reached at 1250 Hz. There was only a slight increase for the sound cancellation system, approaching a peak of 8 dB at 125 Hz and falling to 0 dB at 250 Hz and upward. The average noise reduction was only 3 dB.

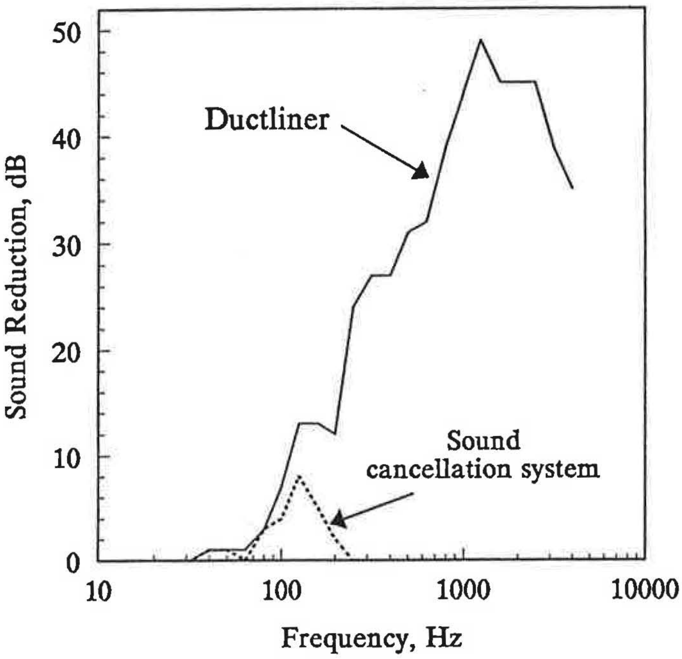

Figure 8 Sound reduction for pink noise injection

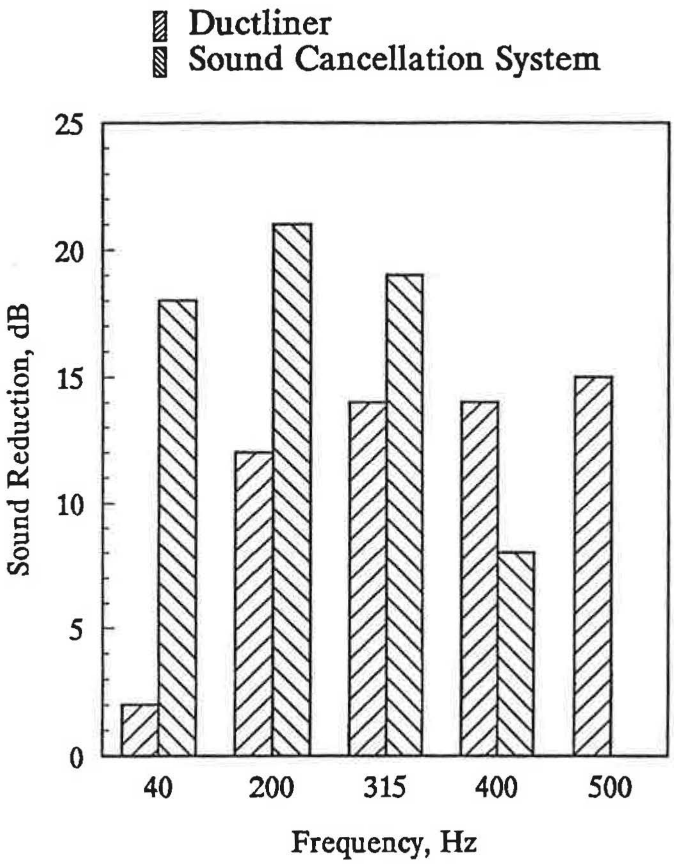

Figure 9 Sound reduction for pure tones

4.4 Effect on pure tones

The sound cancellation system was more effective in tackling pure tones. As revealed in Figure 9, a maximum cancellation of 21 dB was obtained at 200 Hz. It performed better than the ductliner up to 315 Hz. The lower the frequency, the better was the relative performance. Yet, at frequencies greater than 500 Hz, attenuation was insignificant. An average sound reduction of 17 dB was achieved for the four pure tones when compared with the 11 dB in relation to the ductliner.

4.5 Relative performance

The test results indicated that the sound cancellation system was effective for low frequencies and this agreed with the suggested range of less than 400 Hz. For better visualization, the relative performance under different test conditions for frequencies up to 630 Hz is summarised in Figure 10. The system was weak in handling high air flow and broadband noise situations. On the other hand, it demonstrated its ability in treating pure tones and low air flow broadband noise.

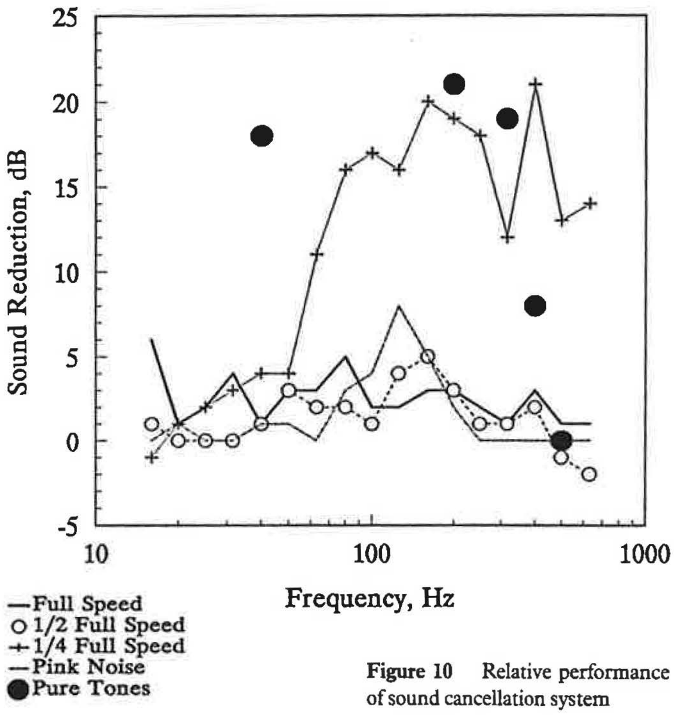

Figure 10 Relative performance of sound cancellation system

5 Discussion

5.1 Technical feasibility of digital sound cancellation system

Experimental results reveal that the performance of the digital sound cancellation system is not entirely satisfactory for relatively large air flow conditions. At 25 m s^{-1} and 14 m s^{-1}, there was very limited success in the low frequency range. Nevertheless, the anticipated preferential treatment of tonal noise to broadband background noise was revealed to a certain extent. The inefficiency may probably be due to the lack of coherence of the primary and error microphone signals resulting from the high speed/turbulence conditions, despite the employment of anti-turbulence microphone probes. The buffer distance adopted in the study may not be adequate and the suggested 2.7 m in highly turbulent flow conditions may be needed.

This would result in the requirement of at least 5 m duct run immediately downstream of the supply fan outlet. The problem is further complicated by the fact that the active noise control system should preferably be installed in a plant room to reduce the high in-duct sound level before the ductwork enters an occupied space; otherwise,there could be duct breakout noise problems. However, this type of plant room and ductwork arrangement cannot easily be achieved, particularly in confined spaces. It is not unusual in Hong Kong to find that several ductwork modificationsions have to be installed in highly congested plant rooms to avoid clashes with other services.

Another possible reason may be due to the over-simplified version of the setup in highly turbulent conditions. According to the recommendations of the manufacturer, the system will operate up to 30 m s^{-1} on tones because, despite the turbulence, tonal coherence remains high. This cannot be achieved for broad band noise. Situations with higher flow velocities can be used provided special precautions are taken to remove turbulence. Configurations with larger number of microphones may have to be employed and this would result in a more complicated and expensive arrangement. In cases where the in-duct sound level far exceeded the turbulence noise, the cancellation system would be capable of handling higher flow velocities. This is, however, more likely to be encountered in industrial rather than commercial systems.

On the other hand, the system performs well at the lower velocity of 7 m s^{-1} and the recorded tonal and broadband noise reduction agrees reasonably with the reported data. The effective frequency range of less than the first cross-modal frequency, calculated to be 573 Hz[18], is also verified. This signifies that under low turbulence conditions, a two-microphone configuration is adequate and this is in line with the manufacturer's recommended flow range of up to about 7.5 m s^{-1}.

In most constant air volume ventilation systems, the in-duct velocity seldom exceeds 10 m s^{-1} while in variable air volume systems, 20 m s^{-1} may sometimes be encountered. It is not common to have too high an air speed as flow generated noise may then become a problem. The two-microphone configuration may therefore be more suitable for low velocity systems while higher order systems have to be considered for high velocity variable air volume systems.

At first sight, the result obtained from the pink noise source was discouraging when compared with others as a relatively significant reduction was only obtained in the 125 Hz band; a closer look reveals a probable explanation. Because of the poor response of the sound reproduction system at low frequencies, the untreated pink noise spectrum obtained is clearly distorted at frequencies lower than 125 Hz which otherwise should be continuously increasing. The turning point at 125 Hz acts as a peak in a broadband spectrum at which the system carries out reshaping in priority to trim down the tone. Yet, the magnitude of the reduction is on the low side. As no air flow is involved, signal coherence will not be a factor. It may be due to the random nature of the pink noise source when compared with the more or less repetitive fan noise. The system may be more capable of cancelling periodic signals as experienced in the real fan spectum and this may lead to poor performance with random noise.

The reduction of pure tones is reasonably effective and is anticipated because of the lack of turbulence so the system can devote all its computing power to tonal reduction. Nevertheless, pure tones seldom exist in practice, particularly in ventilation systems, so this aspect is of less importance.

Another point of concern is the positioning the microphones at the strongest anti-nodes of the acoustic field inside the ductwork in order to obtain maximum cancellation. Due to the wide frequency range encountered, this criterion cannot be satisfied easily without sacrificing most of the frequencies. The more appropriate way is to identify the strong tonal content present in the spectrum and to install the microphones in the corresponding anti-nodes for tonal cancellation. The spectrum can then be reshaped to fit the pattern most favourable to human ears. This approach, however, is more applicable to small ducts as large ducts with flexible walls tend to have breakout at lower frequencies and standing waves will not be formed so prominently.

Although automatic adaptation to changing conditions is a strong point of the system, the extra noise generated from the re-calibration process taking place between the changes in system parameters may cause disturbance to occupants. This would be most notable for small variable air volume systems in which the supply air volume flow rate changes frequently with the space loading and there is insufficient downstream ductwork for subsequent attenuation. Larger systems can help to mitigate the problem while constant air volume systems with less frequent change in system parameters will be ideal for its application.

In addition, infrasound can lead to a lot of problems causing low frequency rumbling sound/vibration and affecting health. The adaptive sound cancellation approach should be an appropriate treatment. However, experimental results indicate the deficiency in this frequency range and it may be due to the limitation of the speaker module to produce efficiently the required cancelling infrasound. The limited length of the plant may also be a factor.

5.2 Hybrid ductliner and sound cancellation system

Conventional ductliners and silencers are generally effective in the mid to high frequency range. If they are to be used on their own for the whole frequency spectrum, they need to be very long in order to absorb the low frequency component. The immediate consequence will be a higher initial cost, and a greater pressure loss. Besides, the resultant sound spectrum will be distorted with an extremely small component of mid and high frequency sound. Even though the overall sound level is small enough to satisfy the widely adopted criteria such as dB(A) and NC level, the quality of the noise will not be pleasant and the principles behind the development of the NC curves or the NCB curves will not be satisfied[19].

To mitigate excessive noise and to satisfy the spectrum balance requirement simultaneously, a combination of ductliner or silencer with the digital sound cancellation system can complement each other for the desired result. The argument is best illustrated by an example of a common real-life situation.

Consider an air conditioning system which consists of an air handling unit serving an office floor. A centrifugal fan rated at 7 m^3s^{-1} at 750 Pa moves conditioned air throughout the distribution ductwork. Included in Table 1 are all the essential calculation data. Although the actual data will vary according to situations, the relative magnitude in various frequency bands tends to follow the pattern listed. As is typical with centrifugal fans, the sound power spectrum emitted is richer in low and mid frequencies. On the other hand, the natural attenuation afforded by the ductwork is mid and high frequency biased. To satisfy the criterion of NC35, the dynamic insertion loss of the silencer has to be chosen specifically for the low frequency end and a 3 m long unit is required. This results in an extremely distorted spectrum with almost no high frequency sound.

Table 1 Calculation data of the hypothetical air conditioning system

	Frequency (Hz)							
	63	125	250	500	1000	2000	4000	8000
Fan sound power level (dB)	90	89	86	84	83	81	77	73
Sound pressure level for NC 35 (dB)	60	52	45	40	36	34	33	32
Required attenuation for NC35 (dB)	14	18	25	17	21	23	19	16
Dynamic insertion loss for 900 mm silencer (dB)	7	12	16	28	35	35	28	17
Dynamic insertion loss for 3 m silencer (dB)	13	23	42	52	55	53	51	45
Resultant sound pressure level with 900 mm silencer (dB)	67	58	54	29	22	22	24	31
Resultant sound pressure level with 3 m silencer (dB)	61	47	28	5	2	4	1	3
Sound cancellation system attenuation (dB)	10	17	18	16	0	0	0	0
Attenuation due to sound cancellation system and 900 mm silencer (dB)	17	29	34	44	35	35	28	17
Resultant sound pressure level with the hybrid system (dB)	57	41	36	13	22	22	24	31

The situation is much improved through the incorporation of the active sound cancellation system. Based on the data measured and assuming that the system is ineffective above 500 Hz as indicated in the manufacturer's literature, the combined effect of it and a 900 mm long silencer is adequate in reducing the noise to comply with NC 35. The quality of the resultant spectrum is also enhanced as it follows more closely the shape of the NC or NCB curves.

5.3 *Economic viability*

Economic viability is another important aspect not to be ignored. Based on the example discussed above, a relatively high initial investment in the cancellation system of around HK$90 000 will be involved. This expenditure can be offset partly through the substitution of the 3 m long silencer with a 900 mm long unit which amounts to around HK$5000.

Another saving will be from the running cost of the whole system. Using a shorter silencer results in a reduction in pressure drop of around 45 Pa. With a volume flow rate of 7 m^3s^{-1} and an overall efficiency of 65%, around 500 W of fan power is saved. On the other hand, the cancellation system draws 14 W. Based on 52 working weeks per year and 44 working hours per week, the annual saving in power consumption is around 1112 kWh which corresponds to HK$778 at a tariff rate of HK$0.7 kWh^{-1}. The exact figures will vary slightly depending on the configuration of the air conditioning system. It seems that the saving throughout the operating life of the equipment is not adequate to cover the initial expense. Maintenance cost has not been included in the calculation as this type of electronic system generally requires very little maintenance and the service life of the system, particularly the loudspeakers, can be prolonged if sufficient capacity is incorporated in the initial selection.

Another consideration is the initial cost of the equipment when compared with that of the air conditioning and refrigeration system. Consider an open plan office with a floor area of 1500 m^2 in which an all-air system is employed. To simplify the situation, assume that two air handling units, each with an air flow of 7 m^3s^{-1} are to be used and both CAV and VAV systems are analysed. The estimated figures are given in Table 2. The digital sound cancellation equipment, if installed only in the supply ductwork, will account for less than 7% and 4% of the air side and the overall air conditioning/refrigeration systems respectively. The figures will be doubled if both the supply and return ductworks are fitted with the equipment.

On average, the initial cost of the cancellation equipment will account for less than 10% of the total investment in the air conditioning and refrigeration system. The annual saving in the electricity charge can be used to offset the routine maintenance cost involved. It is then up to the developer to evaluate the cost effectiveness against the benefits gained.

6 Conclusion

The active noise control concept is applicable in air conditioning systems despite the existence of inherent constraints: limited effective range governed by the cut-off frequency of the ductwork, space and system complexity associated with flow turbulence, positioning of microphones for system performance optimisation, leakage of self-calibration signal upon changes in system parameters, and limitation in the sound reproduction system towards the low frequency end. It can be used to supplement the conventional passive type silencer for cancelling low frequency noise. The resultant spectrum will be more balanced and acceptable to human ears. However, these advantages have to be balanced against the initial cost which cannot be recovered in its servicable life, despite the saving in running cost arising from the reduction in system pressure loss through the use of a shorter length silencer.

The development of microprocessors, transducers and algorithms has been advancing rapidly in recent years leading to the evolution of more sophisticated products in the market. Reportedly, many problems previously encountered have been overcome. The stabilisation effect associated with system re-calibration is no longer an issue. The level of the calibration noise can be adjusted so that it is not so prominent. It is also possible to have systems which store the last mathematical model even after switching off, and upon re-starting, give immediate attentuation. The noise reduction range can also be extended to cover the first mode of propagation in ducts. Products incorporating active and passive means of noise reduction are also available. In addition, the hardware prices have dropped and this may render the active noise control method more viable economically.

Table 2 Cost estimation and analysis

System	Average air side system cost per floor (HK$)	Percentage of cancellation equipment in air side system (%)		Percentage of cancellation equipment in the overall air conditioning and refrigeration system (%)	
		Supply only	Supply and return	Supply	Supply and return
CAV	1 300 000	7	14	4	8
VAV	1 700 000	5	10	3	6

Further investigation in these new digital sound cancellation systems should be carried out under different ductwork arrangements. The effects of different duct endings including open, anechoic or reflecting environments would be areas of interest. The effect of high turbulence but low in-duct noise conditions commonly encountered in non-industrial applications deserves more detailed study. The seemingly lower performance in handling random noise should also be investigated.

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