Proceedings of The Institute of Acoustics

THE SOUND EMISSION FROM AIRCONDITIONING SYSTEMS

D.J. CROOME

SCHOOL OF ARCHITECTURE AND BUILDING ENGINEERING, UNIVERSITY OF BATH AND BURO HAPPOLD (UK) CONSULTING ENGINEERS

Ordinarily there is not a close connection between the flow of air in a room and its acoustical properties, although it has been frequently suggested that thus the sound may be carried effectively to different parts. On the other hand, while the motion of the air is of minor importance, the distribution of temperature is of more importance, and it is on reliable record that serious acoustical difficulty has arisen from abrupt differences of temperature in an auditorium. Finally, transmission of disturbing noises through the ventilation ducts, perhaps theoretically a side issue, is practically a legitimate and necessary part of the subject.

(W.C. SABINE, Collected Papers on Acoustics, Dover, 1964)

BACKGROUND SOUND LEVEL IN BUILDINGS

Noise within buildings may originate from the building services but there is only a limited amount of acoustical data available. Doig (1976) and Croome (1977) have carried out surveys which establish that noise can cause serious problems in building design. The acoustical environment in the occupied space is the resultant of the noise arriving to the space from the engineering services, from adjacent areas by direct or indirect transmission and cross-talk, from the external environment and from noise generation within the space. The building services technologist has a clear responsibility to consider noise problems.

NOISE FROM AIRCONDITIONING SYSTEMS

Airconditioning noise is, or course, only part of the background noise spectrum for a space, but it has become increasingly significant as more high velocity systems are being installed and as people have become more sensitive to internal noise sources in buildings that are well insulated from the external environment.

A series of case studies and some laboratory measurements suggested a simpler approach for estimating the sound emission from airflow systems than the existing method in the CIBS Guide. It also has the advantage of taking into account the interaction between each element comprising the system. This research shows that where there is a strong interaction between any elements of the ventilation systems; any natural attenuation may be cancelled out by generation of sound. Doig (1977) has confirmed this important aspect of noise patterns occurring in air flow systems and proposes the use of coupling factor theory to analyse the problem.

THE SOUND EMISSION FROM AIRCONDITIONING SYSTEMS

In order to ascertain the importance of the problem, a questionnaire was circulated to 49 universities and colleges throughout Great Britain (Croome 1977). The questionnaire consisted of ten questions requesting details of any ventilation or airconditioning services, the acoustic acceptibility of the lecture rooms provided with ventilation, steps taken at the design stage to ensure satisfactory noise levels, specific cases worthy of investigation and details of any amplification systems used. Replies from 16 universities were received from architects and building and planning officers concerning some 120 lecture rooms. In about a third of these the acoustics were considered unsatisfactory from some point of view by lecturers, building officers or students. Although such rooms are built for the sole purpose of allowing communication between lecturer and student, the inclusion of background noise level limit in lecture room specifications appears to be rare. In only one instance was noise level specified: NC25 was chosen. Nearly half the cases made mention in a vague manner that the noise level should be kept to a minimum.

Often the ventilation systems were designed to provide a greater air-change rate when the lecture rooms were fully occupied and this meant that the fan speed increased in proportion to the extra air supplied. Acoustic treatment was not usually recommended at the design stage. Frequently the system was installed, noise complaints were then received and further money had to be spent to provide acoustic treatment.

As a result of the questionnaire survey, it was decided to carry out a site appraisal of the acoustic conditions in 74 of the lecture rooms. In 13 per cent of the cases for single speed systems the acoustic conditions were intolerable and in another 21 per cent, unsatisfactory. (See Table 1).

TABLE 1

ACOUSTIC ACCEPTABILITY OF LECTURE ROOMS (PERCENTAGE AVERAGED OVER THE FREQUENCY RANGE 125 - 4000 Hz)

Subjective acoustic Rating degree of satisfaction	*Single Speed Systems*	*Two speed sytems (17 lecture rooms)*	
		Slow Speed	*Fast Speed*
Satisfactory (< NR 28)	35 %	81 %	15 %
Moderate (NR 28-NR32)	31	13	17
Unsatisfactory (> NR 32)	21	1	19
Intolerable (< NR 37)	13	5	49

It is difficult to say which of the rooms classified as moderate would be unsatisfactory, but according to the standard recommendations, those which exceed NR30 by more than 2dB should be classed as unsatisfactory, the 2dB being the accuracy of the measurements. This means that about 50 per cent of the rooms are acoustically inadequate. The field studies exposed two broad categories of noise source; those originating from the basic nature of the airflow system and those arising from bad planning or installation.

THE SOUND EMISSION FROM AIRCONDITIONING SYSTEMS

Further research has been carried out on a variable speed ventilation system serving a 74 seat lecture room in a university. Air is distributed from a unit comprising a 444 mm diameter centrifugal fan with ten backward-curved blades. The drive shaft is mounted on ball-bearings and is driven by a variable speed motor with four fixed speeds *via* a double vee-belt and pulleys having a 1·6:1 diameter ratio. Table 2 gives the essential performance specification for the multi-speed centrifugal fan unit.

TABLE 2

PERFORMANCE SPECIFICATION OF MULTI-SPEED CENTRIFUGAL FAN UNIT

	Speed 1	*Speed 2*	*Speed 3*	*Speed 4*
Motor:				
No. of poles	4	6	8	12
hp	1.2	1.3	2.5	5.5
speed (rpm)	480	720	960	1140
Fan:				
speed (rpm)	760	1145	1530	2290
volume flowrate (m^3/s)	0.73	1.17	1.55	2.33

Ductwork is of rectangular cross-section and constructed of 18 gauge galvanised sheet steel; the interior and exterior surfaces are unlined. The heater battery comprises a parallel bank of ten 9.5 mm finned hot water pipes arranged evenly across the duct in the air stream. The outlet grilles have two rows of aerofoil vanes at right angles; the five of the inner row are fixed, whereas the six of the outer row are hinged and adjustable. The complete system is suspended from a void ceiling by 9.5 mm threaded rods and supported on brackets. The only contact with the void floor is at the grille outlets.

Air turbulence was evident at all bends in the systems at all fan speeds. Attenuation of sound occurred at all frequencies but was offset almost entirely by generation of sound which is evident at each bend and at each fan speed. The generation was most marked in the 500-1000 Hz frequency range. Sound was generated as the air travels through the attentuator at the three higher fan speeds. One significant conclusion is that sound generation is just as likely to occur in a low speed ventilation system (average air velocity in this example of about 4 m/s) as well as in high speed system (average air velocity in this example of about 13 m/s); this contradicts the traditional belief that sound generation is insignificant at low air speeds.

It is well established that ventilation system noise has a large low frequency content. This work reaches a similar conclusion. At the lowest fan speed, sound in the 31·5-25 Hz range had sound levels along the system of 63 to 88dB (linear) whereas sound in the frequency range 500-1000 Hz had levels in the range

39-75dB (linear). In either case the peak levels were due to sound generation. This pattern of behaviour was similar at the highest fan speed, although the frequencies in the range 500-1000 Hz gradually increased their share of the sound energy as the fan speed was increased.

The effect of sound generation began at some distance before the bend: at low speeds this distance was about 1 m but extended to 2 m or more as the air velocity increased. The generation at the first bend showed some interaction with that which had preceded it at the dampers, the breeches piece and through the attenuator; such interaction also happens to some extent between the generation effects at the bends themselves. Duct resonances do not appear to be significant in this system as no fundamental or harmonic frequencies were observed in the spectra.

A given rise in sound level due to generation occurred over a much smaller distance than a similar decrease in sound level. This work confirmed the point that airflow systems should be designed to have the most simple configuration wherever possible. Good aerodynamic design will assist acoustic design, although a precise relationship between turbulence and sound generation was not established for this system.

The results showed that where there is a strong interaction between any elements of the ventilation system, any natural attenuation can be cancelled out by sound generation. The varying attenuation rates - some are even zero - are also indicative of interaction effects. The attentuation rates were variable and very different to those used in practice (*see* IHVE Guide Book B, 1970). Sound generation magnitudes at bends were of a much higher order - from 4dB at low frequencies to 18 dB at high frequencies - than attenuation magnitudes in straight metal ducts which range from 0.5 to 3 dB/m; no attenuation occured at bends. These results dispute much traditional design data and further research is needed to resolve the conflicts.

A satisfactory design criterion for the lecture room would be an L_{10} level of 36dB(A) or NR30. The lowest speed ventilation produces a sound level of 48dB(A) in the room. The attenuator made no effective contribution.

Relationships between room sound pressure level and fan speed, and between room sound pressure level and main duct air velocity were derived. It may be possible to predict room sound pressure level from this type of relationship instead of the more complicated procedures given in the 1970 IHVE Guide mentioned above, but more research is needed in order to establish the influence of systems configuration, duct material and various makes of equipment on this function. These preliminary studies show that the sound level in the room, L, can be predicted from the fan speed N (rpm) or the main duct air velocity v (m/s) by:

$$L = 40\left(\frac{v}{17} + 1\right) \quad \text{in dB(A)} \qquad (1)$$

$$L = 37.4\left(\frac{N}{2110} + 1\right) \quad \text{in dB(A)} \qquad (2)$$

If N and v are fixed by other circumstances than acoustic ones the order of attenuation required can be estimated. Alternatively, if N can be chosen to match the selected criterion for L then this value of L can also be chosen to determine v, and hence the size of the main distribution duct can be calculated. Using Darcy's formula the room sound level may be related to pressure drop per unit length of main duct, Δp, by:

$$L = 40 \left[\frac{1}{17} \left(\frac{2gm\,\Delta p}{f} \right)^{\frac{1}{2}} + 1 \right] \quad \text{in dB(A)}$$

for a duct having a mean hydraulic depth m, friction factor f, and where $g = 9.81\text{m/s}^2$.

Further work has established the following conclusions: -

(a) Formula (2) is more reliable than Formula (1) and only applies to centrifugal fans.

(b) Natural attenuation and airflow effects are included in the empirical derivation of Formula (2), i.e. the system attenuation and generation which occurs between the fan and the measuring point in the room; additional attenuation, α, may be required and Formula (2) will be modified to:

$$L = 37.4 \left(\frac{N}{2110} + 1 \right) - \alpha \qquad (3)$$

(c) An overall natural attenuation rate of 0.5dB/m is a conservative estimate which may be used in planning the distance between an airconditioning plant and occupied spaces but any straight sections in the network which measure less than 7 m in length between fittings such as bends and these should be ignored when making this estimate. If L_p is the sound level in the plant room, L in the occupied space and D is the length of airflow ducts between the plantroom and the occupied spaces (less any straight runs which are less than 7m), then the sound level in the room is:

$$L = L_p - 0.5D - \alpha \qquad (4)$$

Thus plant selection, choice of fan speed, distance planning can be decided for a specified value of L. These factors may be combined in such a way that $\alpha = 0$ and no cost for sound control equipment is necessary, but in practice it is more likely than $\alpha > 0$. Good maintenance is also necessary to maintain the design operating conditions.

(d) The results give reliable predictions ($\pm$ 5dB) for centrifugal fan speeds below 900 rpm. Higher fan speeds give less reliable predictions.

(e) German research reported by Eck (1973) shows that the loudness level of an axial fan is dependent on speed according to:

$$L = 54\left(\frac{N}{2160} + 1\right) \text{ phons} \quad \text{or} \quad 42\left(\frac{N}{1680} + 1\right) \text{ dB(A)}$$

(f) Adjustment of louvres in ceiling grilles can have a significant effect on the sound level measured in a room.

REFERENCES

Croome, D.J., - 1977 - Noise, Buildings and People (Pergamon Press).

Doig, R., - 1977 - PhD Thesis, The Transmission of Noise through Ducted Air Sytems, University of Liverpool.

Eck, B., - 1973 - Fans (Pergamon Press).