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'Blue Pages'

Articles on the state of the art of developments in building services engineering

Welcome to the 'Blue Pages'. This is where guest editors from the Editorial Advisory Panel for *Building Services Engineering Research and Technology* address current developments in building services engineering practice and research. The articles are very short, on one theme of current interest, and do not go through the longer refereeing process for conventional research papers. This is to encourage consultants and contractors to discuss their latest developments in a non-commercial manner. Academics will also be asked to outline current research areas in universities and colleges. Without the constraint of full refereeing there is scope in the articles to provoke interest and to raise issues for discussion. Consequently the Editor of *BSER&T* is happy to receive and publish letters furthering the discussion of 'Blue Pages' articles. If you have a suggestion for an article or topic you would like to see included in the 'Blue Pages' please inform the Editor, Barry Copping. Comments on the 'Blue Pages' are also welcome. 'Blue Pages' will only succeed if readers of the journal wish the feature to succeed and contribute to its success.

Acoustics: Part 2

Editorial

Prof. David Oldham (School of Architecture and Building Engineering, University of Liverpool)

Part 2 of the Acoustics 'Blue Pages' opens with C M Mak's description of possible future developments in the treatment of flow—generated noise. John Shelton then discusses the relatively new technique of sound intensity measurement made possible by advances in digital electronics, which enables the direct determination of the acoustic power flow from a noise source. Building services engineers are increasingly becoming involved with the installation of public address systems, and Ken Dibble explains the problems likely to be encountered with such installations. Finally, Mario Rossi discusses the effect of masking noise on the perception of speech and how this can be employed in a positive manner to enhance acoustic privacy.

A

Towards generalised prediction techniques for regenerated noise in ventilation systems

C M Mak (Acoustics Research Unit, School of Architecture and Building Engineering, University of Liverpool)

1 Introduction

Regenerated noise is the name frequently given to noise which is generated on the quiet side of the primary attenuators of a ventilation system, resulting from the interaction between turbulent air flow and duct discontinuities or flow spoilers. At long distances from the fans in a flow duct, regenerated noise from in–duct elements can cause serious noise problems. However, if regenerated noise could be predicted at the design stage then suitable remedial treatment could be prescribed to reduce the problem. If the noise levels are not correctly predicted and if a problem arises after commissioning, it is often impossible to apply remedial treatment since usually no space is available. Hence the accurate prediction and control of the flow-regenerated noise of duct fittings is of considerable engineering importance.

Existing design guidance methods are based upon a limited number of measurements made on a few representative components. Application of the limited amount of available experimental data to components differing in size or shape from those on which measurements have been made is fraught with danger, and may be one reason why the current techniques for the prediction of regenerated noise levels are recognised as being inaccurate.

The lack of regenerated noise data reflects the difficulty in obtaining such information by conventional measurement techniques. The conventional measurement approach requires the use of an expensive special combined acoustic and aerodynamic facility consisting of (*a*) a powerful fan system; (*b*) a high-performance silencer system to reduce the noise from the fan entering the section of test ductwork; (*c*) a long test duct to ensure that flow conditions are stable at the test element; (*d*) a large reverberant room, characterised by a low level of background noise, in which the sound power generated is measured (see Figure 1).

Over the years a number of researchers have attempted to overcome the need for measurements on all possible compo-

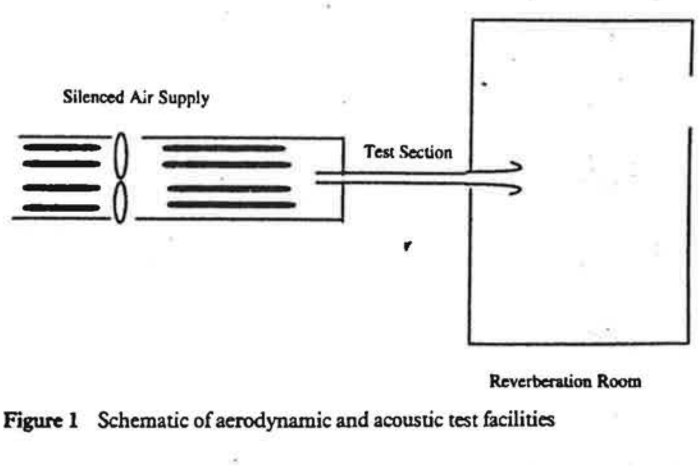

Figure 1 Schematic of aerodynamic and acoustic test facilities

nents by devising generalised prediction techniques which could be applied to air duct elements. In this article we examine past attempts based upon pressure loss and discuss new possiblilities arising from recent developments in computational fluid dynamics.

2 Pressure-based techniques

As well as causing regenerated noise, an in-duct flow spoiler causes a loss of static pressure. The more obtrusive the spoiler is, the greater is the pressure loss and the greater is the sound power generated. A number of investigators have tried to devise a generalised technique for the prediction of regenerated noise in duct elements by looking for correlation between the two effects.

The most significant early work was that carried out by Gordon[1,2] who investigated the sound power generated by a variety of spoilers close to the end of a pipe carrying high-velocity air flow. Gordon was primarily interested in noise problems related to aircraft engines and spoilers situated close to the end of a pipe; hence the generalised technique that he developed was not suitable for predicting regenerated noise in ventilation systems. However, his work did lend encouragement to the concept of pressure-based techniques for ventilation systems.

Nelson and Morfey[3] devised a pressure-based prediction technique for simple strip spoilers in low-speed flow ducts. In developing their technique they took into account the effect of the duct environment on the generation of noise. The basis of their theory is that the sound power radiated by an in-duct spoiler is related to the total fluctuating drag force acting on the spoiler which is a function of the turbulence intensity in the vicinity of the spoiler. To arrive at a predictive technique, since they were unable to determine the actual spectrum of the turbulence in the vicinity of the spoiler, they assumed that fluctuating drag force is in direct proportion to the steady drag force. This is the same assumption that Gordon made in devising his theory, and its validity has been confirmed by the experiments of Heller and Widnall[4]. Their experimental data were collapsed into a single generalised spectrum through empirical evaluation of the constant of proportionality K(St) between the fluctuating and steady drag forces as a function of Strouhal number St.

Nelson and Morfey obtained two expressions for determining the sound power generated by an in-duct spoiler, one corresponding to frequencies below the duct cut-on frequency fo and one corresponding to frequencies above the duct cut-on frequency:

For $f_c < f_o$:

$$120 + 20\log_{10}K(\mathrm{St}) = \mathrm{SWL_D} - 10\log_{10}\{\rho_0 A[\sigma^2(1-\sigma)]^2 C_D^2 \times U_c^4/16c_0\} \qquad (1)$$

For $f_c > f_o$:

$$120 + 20\log_{10}K(\mathrm{St}) = \mathrm{SWL_D} - 10\log_{10}\{\rho_0 \pi A^2 \mathrm{St}^2 [\sigma^2(1-\sigma)]^2 C_D^2\ U_c^6/24c_0 d^2\} - 10\log_{10}[1 + (3\pi c_0/4\omega_c)(a+b)/A] \qquad (2)$$

where $\mathrm{SWL_D}$ is the in-duct radiated sound power level; K(St) is the single Strouhal number dependent constant; ρ is the density of air; A is the cross-sectional area of the duct; $\sigma = A_c/A$ is the duct unobstructed area/duct area, i.e. the open air ratio; St is the Strouhal number $= f_c d/U_c$; $U_c = q/A_c$ is the volume flow rate/duct unobstructed area i.e. the velocity in the restriction; c_o is the ambient speed of sound; d is the characteristic dimension of the spoiler; ω_c is the angular centre frequency of the band of frequencies under consideration; a is the duct width; b is the duct height; C_D is the drag coefficient $= F_3/[U^2A(1-\sigma)/2]$ where F_3 is the steady-state force on the spoiler.

Although the Nelson and Morfey equations appear complex, all terms in their proposed predictive equations are either constants or measurable variables plus the single Strouhal number dependent constant. The prediction method is essentially based upon an empirical relationship between the loss of static pressure caused by spoilers and the sound power generated. With the value of the Strouhal number dependent constant established from a series of experiments, it is possible in principle to employ the Nelson-Morfey theory for predictive purposes.

Recently, Oldham and Ukpoho[5] have extended their work to the case of circular ductwork and more complex flow spoilers. They rewrote the Nelson-Morfey equations by determining the appropriate values of the open air ratio and the characteristic dimension from consideration of the pressure loss, so that their work could be applied to more complex flow spoilers in circular or square ducts.

The Nelson-Morfey equations as modified by Oldham and Ukpoho for determining the sound power generated by an in-duct spoiler in circular ductwork are as follows:

For $f_c < f_o$:

$$120 + 20\log_{10}K(\mathrm{St}) = \mathrm{SWL_D} - 10\log_{10}\{\rho_0 A\sigma^4 \times C_L^2 U_c^4/16c_0\} \qquad (3)$$

For $f_c > f_o$:

$$120 + 20\log_{10}K(\mathrm{St}) = \mathrm{SWL_D} - 10\log_{10}\{\rho_0 \pi A^2 \mathrm{St}^2 \sigma^4 \times C_L^2 U_c^4/24c_0 d^2\} - 10\log_{10}[1 + 3c_0/8rf_c] \qquad (4)$$

where

$$d = \pi r(1-\sigma)/2 \qquad (5)$$

and

$$\mathrm{St} = f_c \pi r(1-\sigma)/2U_c \qquad (6)$$

Oldham and Ukpoho carried out experiments on dampers and orifice plates in a circular duct and obtained a generalised spectrum by the collapse of experimental data on the basis of the modified Nelson-Morfey equations. The spectrum obtained was similar to that observed by Nelson and Morfey with simple strip spoilers. Figure 2 shows the spectrum obtained by Oldham and Ukpoho. The value of K(St) can be obtained from this curve and used in equations 3 and 4 to predict the level of regenerated noise for a spoiler. All other terms in these equations are either known or derived from the

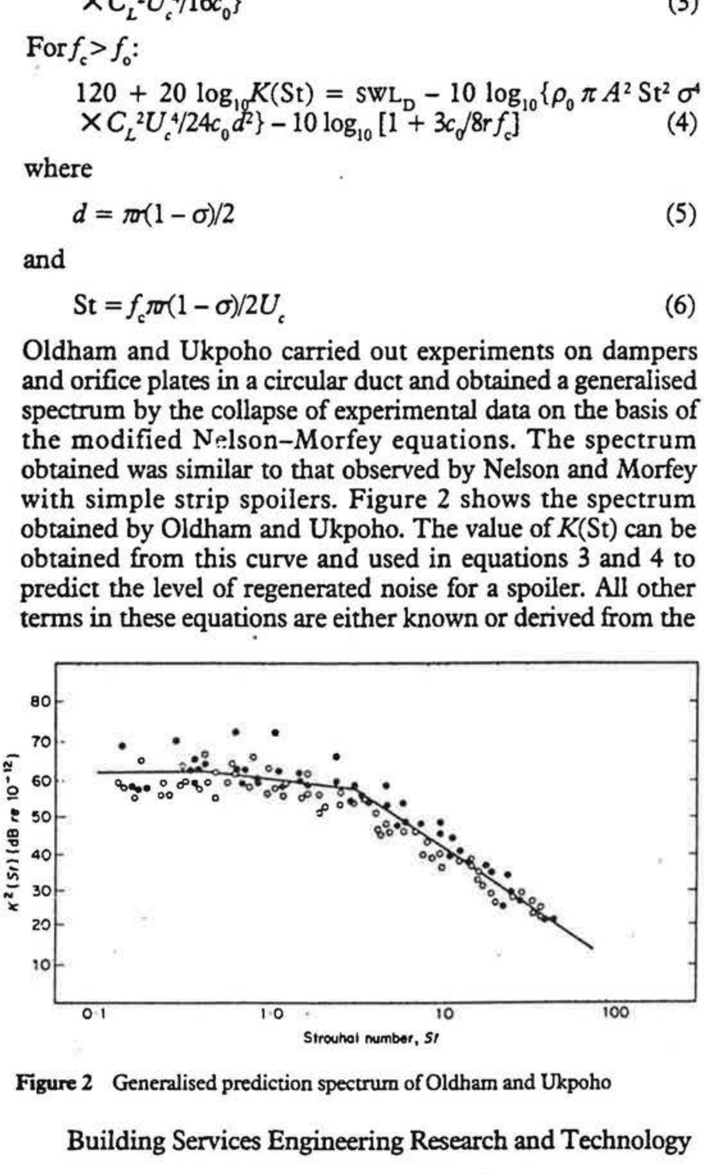

Figure 2 Generalised prediction spectrum of Oldham and Ukpoho

static pressure loss across the spoiler. Figure 3 shows a comparison between the regenerated noise level predicted using Figure 2 and equations 3 and 4 and measured values for a spoiler configuration not employed in obtaining Figure 2. The agreement between the two sets of data is good.

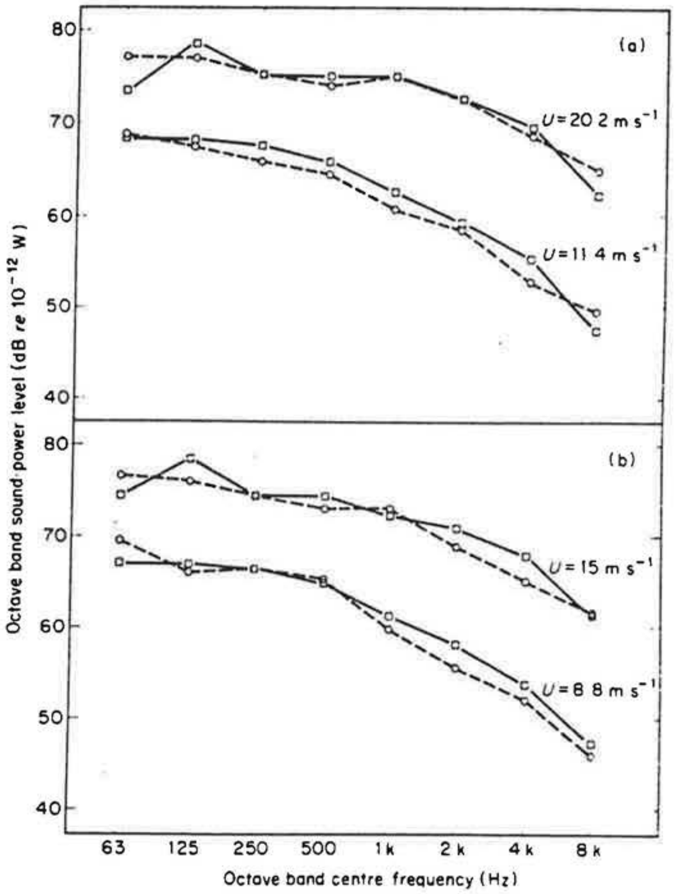

Figure 3 Comparison of measured ——— and predicted - - - - - - data for flow spoiler

Although the results of Oldham and Ukpoho lend further support to the concept of a generalised prediction technique based upon pressure loss, their experiments were carried out using ductwork of size similar to that of Nelson and Morfey, and the generalised prediction curve which they derived may not be applied to systems having very dissimilar duct dimensions. The technique was also developed from a study of sound generated by simple in–duct elements such as orifice plates and dampers. Further work is necessary to investigate the technique when applied to those flow spoiling elements typically found in ventilation systems, such as bends, transition pieces etc.

3 Towards a generalised prediction technique using CFD

In recent years, considerable advances have been made in computational fluid dynamics (CFD), so that there are now available a number of commercial codes which can be used to predict the behaviour of moving air. In principle, these codes could be used to determine the fine details of airflow turbulence and this could then be used to predict the regenerated noise. In practice, in order to solve the relevant turbulent flow equations with sufficient accuracy, it is necessary to have a three—dimensional computational mesh large enough to cover the region of interest (which may be of dimensions of several metres) but with spacings smaller than the smallest turbulent motion (eddy) present in the flow (in air, this can be as small as 0.1 mm). In addition, the computations must be made for unsteady flow, utilising a time—step smaller than that associated with the fastest eddies. This would mean working with elements so small as to make the computing time required extend over many centuries, and hence, the direct determination of regenerated noise is impossible. In this article we have suggested an alternative approach in which computational fluid dynamics is used in such a way as to permit regenerated noise levels to be determined indirectly.

Although direct numerical simulation of turbulent flows is not practical, CFD codes use 'turbulence models' to represent the small-scale effects of turbulence. The turbulence model consists of a set of differential equations and/or algebraic formulae which allow determination of Reynolds stresses (these are introduced by the averaging processes of the instantaneous Navier–Stokes equations) and hence close the time-averaged equations of fluid motion. CFD models can thus be used to provide an estimate of the turbulence kinetic energy in the vicinity of a flow spoiler.

The first stage in the search for an indirect method of predicting regenerated noise using CFD is to determine the relationship between the turbulence in the vicinity of a flow spoiler and the sound power generated. This may be done indirectly by determing the relationship between turbulence and the steady drag on the spoiler, since the work of Nelson and Morfey and Oldham and Ukpoho has established a relationship between the latter and the sound power generated.

In generalised prediction techniques there is a need to collapse all data onto a single curve and this is normally done by plotting data against a Strouhal number. Calculation of the Strouhal number requires values of frequency, velocity and a 'characteristic' dimension. The first two parameters present little problem but the third requires further thought. For the strip spoilers employed in the Nelson and Morfey experiments this representative dimension was assumed to be the width of the spoiler strip employed. For more complex (and more realistic) flow spoilers the choice of the characteristic dimension is less obvious. It may be possible to establish this value by means of CFD. Two techniques could be employed. The first is to employ more complex turbulence models which may be able to reveal information regarding the frequency of the largest eddies, which may determine the peak frequency of the turbulence spectrum. Figure 4 shows the time history of streamlines in the wake of a cylinder computed using the renormalisation group turbulence model of the FLUENT package. The value of the Strouhal number relating to the frequency of vortex shedding determined from the CFD simulation is 0.185, which compares very well to the experimental value of 0.18–0.19.

An alternative approach would be to relate the characteristic dimension to the spatial separation of regions of maximum turbulence (hot spots) in the vicinity of the spoiler. Figure 5 shows an idealised representation of the flow of air around a cylindrical spoiler. Maximum turbulence acting upon the spoiler will be experienced in the two flow separation regions, and the distance between these regions will determine the peak frequency of the resulting spectrum. For a simple cylindrical spoiler in a free airstream the spectrum is essentially a pure tone the frequency of which is given by:

$$St = 0.19 = fd/v$$

where v is the air velocity and d is the diameter of the spoiler.

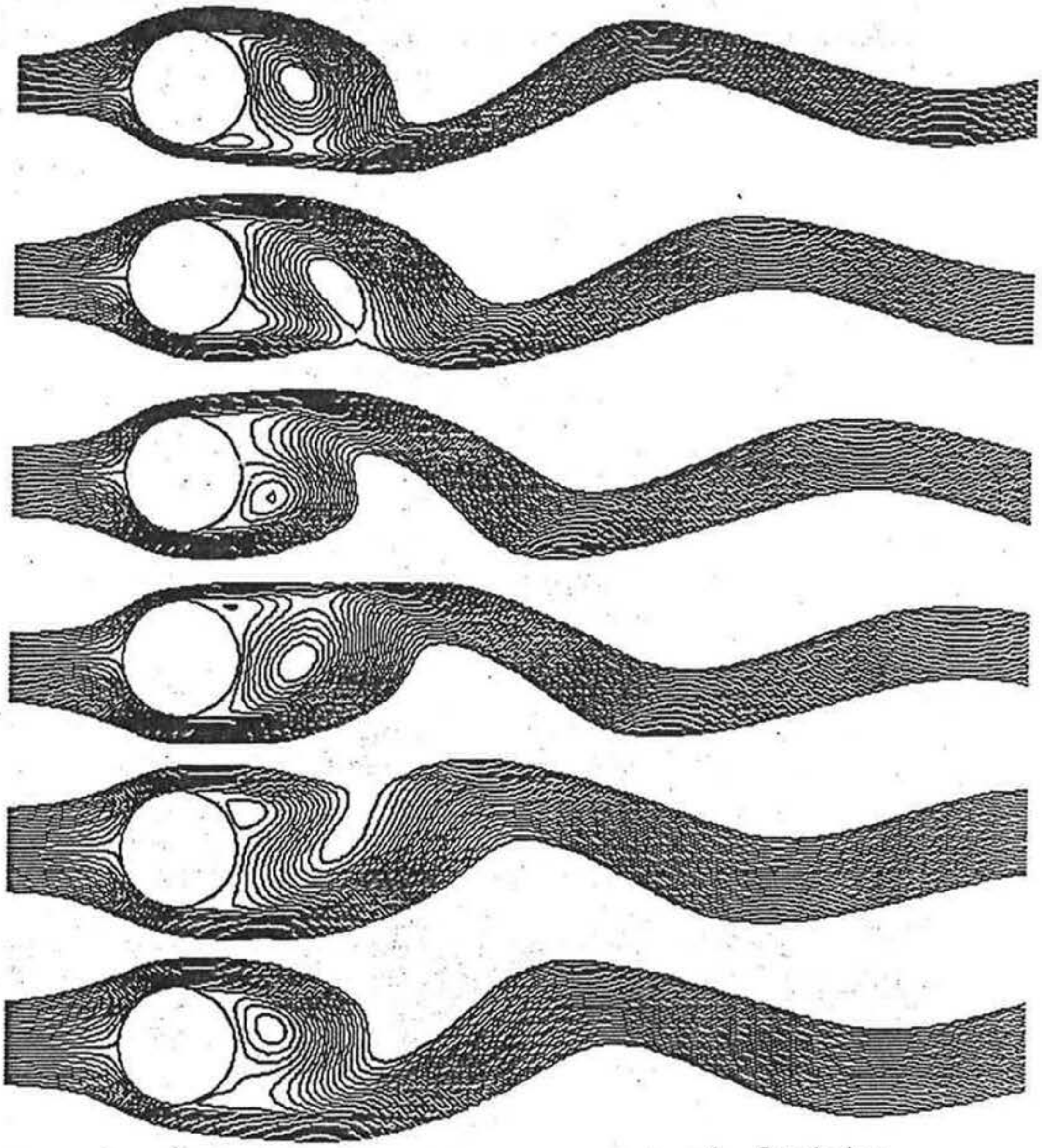

Figure 4 Streamlines due to cylindrical spoiler obtained using FLUENT CFD package

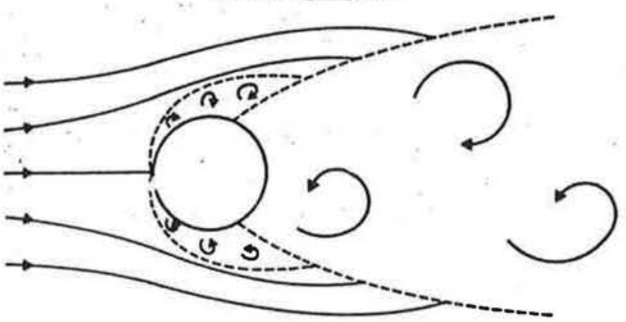

Figure 5 Vorticity in the boundary layer and wake of a cylindrical spoiler

Note that although the Strouhal number is calculated using the spoiler diameter, the value of the 'hot spot' separation distance could equally well have been used since, for this spoiler configuration, it is proportional to spoiler diameter (at least over the range of Reynold's numbers likely to be encountered). An examination of the flow characteristics of a number of in-duct flow spoilers reveals pairs of turbulence 'hot spots' to be a common phenomenon, and suggests the possibility that their spatial separation might be a suitable dimension for determining the Strouhal number.

4 Conclusions

Pressure-based techniques for predicting regenerated noise have been shown to be very promising in laboratory tests. However, because of the difficulties associated with the need for large facilities these tests have been on a limited range of spoilers. Further work is needed on a range of components in order that the true potential of this method can be evaluated. Work on the application of CFD to the prediction of regenerated noise is still in its early stages but holds considerable promise for the future.

5 References

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The measurement of sound intensity and its practical applications in building services

J Shelton (AcSoft Ltd)

1 Introduction

The measurement of sound intensity, one of the fundamental parameters in acoustics, is relatively new, and has been made possible by advances in instrumentation. Sound intensity measurement brings many new possibilities for the noise control engineer, both in standardised measurements and troubleshooting, although the additional complexities involved can trap the unwary.

This article reviews the basic theory behind sound intensity and its measurement, and discusses its applications and advantages over traditional measurement techniques.

2 Basic theory

Any acoustic source has a *sound power* in watts (or expressed in decibels re: 10^{-12} watts), and this quantity describes that source's ability to generate sound energy.

This energy is transmitted into the surrounding environment as *sound intensity* which is expressed as power flowing through a given area in watts per square metre (or decibels re: 10^{-12} W m^{-2}). Sound intensity is therefore a vector quantity with both magnitude and direction, and the amount of energy emitted by the source in a given direction is described by its directivity.

The result of this energy radiation will be sound pressure at a particular point, expressed in pascals (or decibels re: 20 μPa). The sound pressure will depend not only on the sound power of the source, but also the acoustic environment.

The sound power W is given by

$$W = \int_S \hat{I}.dS \qquad (1)$$

where $\hat{I}$ is the sound intensity and S is the surface area of an imaginary surface enclosing the source.

The intensity $\hat{I}$ is described as the acoustic flux density, and is given by

$$\hat{I} = \lim_{T\to\infty}(1/T)\int_0^T p(t)\,.\,u(t)\mathrm{d}t \qquad (2)$$

where p(t) and u(t) are the instantaneous sound pressure and particle velocity respectively. In a given direction n, the sound intensity is given by

$$I_n = \overline{p(t).u_n(t)} \qquad (3)$$

where the bar indicates time averaging.

The particle velocity is a complex vector quantity, which in a free field (with no reflections) is in phase with the sound pressure. In this rather theoretical situation, the particle velocity can be calculated from the sound pressure as follows:

$$u = p/\rho c \qquad (4)$$

where ρ is the density of air and c is the speed of sound in air. The product ρc is known as the characteristic impedance of air, and an analogy in terms of electrical theory can be drawn with the relationship between voltage and current:

$$I = V/R \qquad (5)$$

where R is a pure resistance. Therefore, in a free field, we can calculate the sound intensity from the sound pressure directly:

$$|\hat{I}| = \overline{p.u} = \overline{p^2/\rho c} = p^2_{rms}/\rho_c \qquad (6)$$

It is then an easy matter to calculate the sound power. Also, by choosing suitable reference values for the decibel representation of sound pressure and sound intensity level, the two levels are numerically the same. This is why special acoustic facilties are built to provide a free field (anechoic chamber) so that the sound power can be calculated from sound pressure measurements with, for example, a simple sound level meter.

This is all very well if we can move the source to be measured into a special room. However, this is often impossible, and in the case of building services equipment, the sound power of a source may depend òn other ancillary equipment, which in turn will interfere with the measurement. The simple relationship between sound pressure and intensity breaks down in normal sound fields, and therefore there is no simple way to calculate the sound power from sound pressure measurements.

As more reflections are introduced into the sound field, energy is stored as reverberation, and the particle velocity starts to become reactive, and out of phase with the pressure. In the limit, where the acoustic energy is re—circulating in the environment, as in a standing wave or reverberation room, the particle velocity is completely out of phase with the pressure, and the time—averaged product of the two (equation 3) becomes zero, i.e. there is no net flow of energy in any direction.

Real acoustical environments always fall somewhere between the two extremes described, and therefore the sound intensity level is generally below the sound pressure level (in dB). In simple terms, the intensity measures the active part of the sound field, whereas the pressure indicates both the active and reactive parts of the field. Therefore, to calculate the sound power of a source in a real environment requires the measurement of sound intensity directly.

3 Measuring sound intensity — the theory

To measure sound intensity, we need to sense both the pressure and particle velocity components simultaneously, and calculate the time—averaged product of the two.

Measuring sound pressure presents no problem, and the use of high—quality condenser microphones has been at the heart of acoustics for many years. Many attempts have been made to measure particle velocity directly, however, with limited success, and this is the reason that the measurement of intensity is a relatively new phenomenon.

The solution actually uses two condenser microphones, and by detecting the pressure gradient between the two, the intensity can be approximated. The accuracy of this approximation depends on the configuration of the probe, the quality of the instrumentation and the sound field itself.

The principle involved is based upon Euler's equation, which is essentially the application on Newton's Second Law to a fluid. Newton's Law states that if we know the force applied to a mass, we can then find the acceleration of that mass. In Euler's equation, it is the pressure gradient which accelerates a fluid of density ρ, and if we integrate the acceleration, we can deduce the particle velocity.

The particle acceleration is given by:

$$a = (1/\rho)\nabla p \tag{7}$$

or in one direction r:

$$a = \delta u/\delta t = (-1/\rho)\,\delta p/\delta r \tag{8}$$

and so the particle velocity is given by

$$u = (-1/\rho)\int \delta p/\delta r \, dt \tag{9}$$

The pressure gradient can be approximated by taking two closely spaced microphones, measuring the instantaneous pressure between the two, and dividing by the spacing:

$$\delta p/\delta r \approx \Delta p/\Delta r = (p_1 - p_2)/\Delta r \tag{10}$$

where Δr is the microphone separation.

Equation 9 now becomes:

$$u = (1/\rho)\int (p_1 - p_2)/\Delta r \, dt \tag{11}$$

This gives the expression for the particle velocity, which can then be multiplied by the mean pressure between the microphones to give the intensity as

$$I = \overline{p.u} = (p_1 - p_2)/2\{(-1/\rho)\int (p_1 - p_2)/\Delta r \, dt] \tag{12}$$

or

$$I = -[(p_1 + p_2)/2\rho\Delta r]\int (p_1 - p_2) \tag{13}$$

A typical sound intensity probe therefore consists of a mechanical arrangement holding two condenser microphones and their preamplifiers with a fixed spacing between the two (Figure 6). The actual spacing depends on the frequency range to be covered by the measurement, caused by the approximation used in equation 10 above, which is a finite—difference approximation.

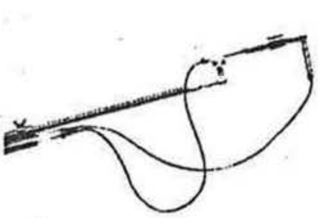

Figure 6 A two-microphone sound intensity probe

At high frequencies, the wavelength of the sound becomes the same order of magnitude as the microphone spacing, and the pressure gradient is underestimated, and in fact becomes zero when the wavelength is twice the spacing. For a 12 mm spacing, this zero crossing occurs at 14 kHz, well within the audio bandwidth. Therefore, we must apply a limit for which this underestimation is less than, say, 1dB in intensity, which for 12 mm is 5 kHz. To measure up to higher frequencies requires a smaller microphone spacer.

At the other end of the scale, at low frequencies, the spacing becomes small compared with a wavelength. The resulting pressure gradient becomes extremely difficult to detect, and will depend on the inherent matching of the microphones (the uncertainty principle). Typically, if the microphone phase mismatch is around 0.2°, then for a maximum error of 1 dB with a spacing of 12 mm, the lower limit is 80 Hz.

Clearly, for high-performance measurements, a good phase match between the two measurement channels is essential, and ingenious techniques have been used to reduce this error to a minimum, with a resulting complexity in the instrumentation. In order to achieve the full frequency range required it may be necessary to take two measurements with two probe configurations.

All of the above discussion applies to a free field. However, when reactivity creeps in, with the real part of the particle velocity becoming smaller, as the reactive (reverberant) part of the sound field builds up, the phase difference becomes even more difficult to detect at low frequencies, and this increases the lower limiting frequency of the measurement.

The reactivity of the sound field is defined as

$$L_k = L_I - L_P \tag{14}$$

where L_I and L_P are the sound intensity and sound pressure levels respectively. This is close to 0 dB in a free field, and tends to $-\infty$ dB in a purely reverberant field.

In the example above, if the reactivity increases by 3 dB, then the lower limit increases by an octave (i.e. to 160 Hz). Therefore, the accuracy of the measurement depends not only on the probe configuration, but also on the quality of the microphones and on the sound field itself.

5 Measuring sound intensity — the practice

All of this can be a minefield for the uninitiated, but thankfully, along with the development of the technique, safeguards have been developed which ensure that accurate measurements are taken.

The first is calibration. Calibrating condenser microphones is now routine, and as the probe consists of two special condenser microphones, pressure calibration presents no problems. However, although the probe will leave the manufacturer in top condition, it is essential to be able to measure the residual phase matching of the microphones in the field, to determine the lower limiting frequency of the measurements for a given microphone spacing and field reactivity. This is done by using a special coupler which exposes the microphones to an identical sound pressure, in other words simulating a reverberant field. The theory tells us that the intensity should be zero ($-\infty$dB) in this situation, so that any measured intensity will be due to mismatch between the probe and analyser. This is termed the residual intensity level. The difference between this and the measured pressure in the coupler is the *residual P–I index*, and defines the amount of true reactivity in the sound field which can be tolerated by the system without introducing errors. More details on this can be found in the sound intensity standards described later.

The second safeguard which can be taken is to use digital signal processing (DSP) to calculate the sound intensity. The recent advent of DSP has allowed more accuracy to be achieved in acoustic measurements, by replacing analogue components by digital calculations. As the calculations for sound intensity are between two channels of data, then if the signal can be digitised immediately after the microphone preamplifiers, inaccuracies due to phase mismatch can be minimised. Also, assuming that the programmer has done his job correctly, no errors will creep into the calculation with time.

Additionally, the residual phase mismatch can be removed from the calculation digitally to improve the dynamic capability of the system.

Two ways of calculating sound intensity digitally have arisen, one in the time domain, and one in the frequency domain. Which to choose will probably depend on the budget, and whether other types of measurements will be needed.

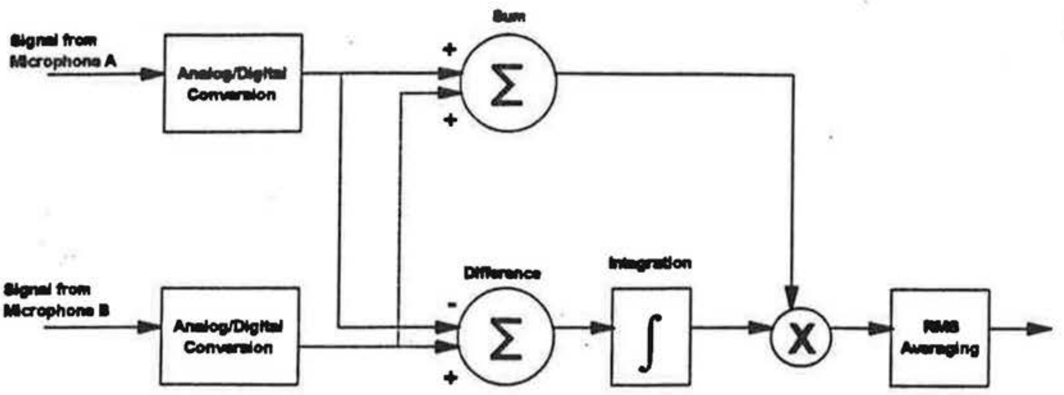

Figure 7 Direct digital realisation of an intensity analyser in the time domain

6 Time or frequency domain?

The time domain analyser basically realises equation 13 in digital form, as a sample—by—sample calculation, using sum—and—difference and digital integration/averaging (Figure 7). Frequency analysis is achieved by digital filtering, giving results in octaves, third octaves and sometimes 1/12 octave (6%) bandwidths.

These analysers are true real—time analyzers, and will find their application mainly in acoustic measurements, where fractional octave results are commonplace. However, frequency domain realisation of equation 13 involves fast Fourier transformation (FFT). The advantage of this is that FFT analysers are becoming commonplace, and offer excellent price/performance ratios. The FFT gives

$$I = (-1/\omega\rho\Delta r)\,\mathrm{Im}\,G_{AB} \qquad (15)$$

where G_{AB} is the cross spectrum between the two microphone signals, and ω is the angular frequency.

Calculating sound intensity is therefore a case of scaling what is a common function in a two-channel FFT analyser, and many analysers now incorporate this calculation as standard. Spectra are calculated down to very narrow resolution, excellent for troubleshooting, and fractional octaves are achieved by synthesis from narrow bands.

The additional benefit of FFT analysis is that it will find many other uses in vibration measurement and analysis, and may go some way to justify the investment in a system at all.

It is no longer necessary to purchase dedicated stand—alone analysers, as systems can now be built up from acquisition cards installed in a personal computer, to give fully integrated systems, making full use of industry standard graphics and storage.

Again, the standards provide extensive guidance on performance criteria of analysis systems, and key phrases are real—time speed and accuracy of octave synthesis.

7 Applications

The two main applications of the sound intensity technique are

(*a*) Sound power determination

(*b*) Source location and source ranking

7.1 Sound power determination

As discussed at the beginning of this article, the main reason to measure sound intensity at all stems from a need to determine sound power in normal environments, without using specialised acoustic facilities such as anechoic chambers.

In order to determine the sound power of a device, we simply enclose the source in a control surface, and determine the average sound intensity flowing through that surface. The sound power is then given by equation 1:

$$W = \int_S \hat{I}.\mathbf{d}S \qquad (16)$$

In practice, we are only measuring the intensity normal to the control surface, in the direction of the probe, and we approximate the surface integral by taking discrete samples on the surface, so the equation becomes:

$$W = \sum_{i=0}^{n} I_i.\mathbf{d}S_i \qquad (17)$$

where I_i is the intensity measured normal to the surface and $\mathbf{d}S_i$ is the incremental area associated with the measurement.

This is illustrated graphically in Figure 8.

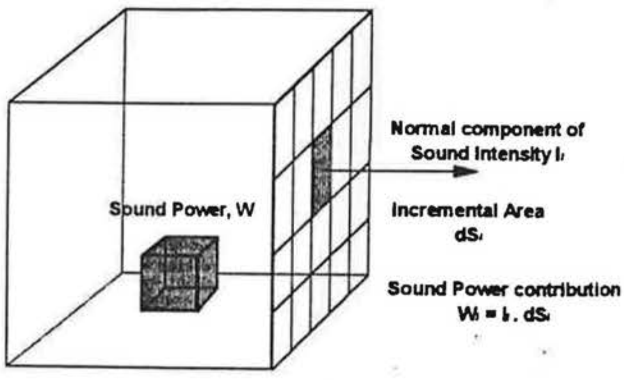

Figure 8 Calculation of sound intensity

Because the surface completely encloses the source, any sound intensity entering the surface, from another source close by for example, will be cancelled through the summation process by the energy leaving the surface elsewhere (Gauss's theorem). This assumes no absorption within the surface, but allows the determination of the sound power, even though other sources may be present.

For building services this is an attractive benefit, as it allows determination of the sound power of, say, a fan, even though ductwork or terminal units are radiating noise nearby. Clearly, measurements in specialised rooms would be impossible in this situation. Similarly, it is possible to subdivide complex machines into smaller sources, and to establish the relative strength of these sources (Figure 9). Using this method, external noise sources up to 10 dB higher in level can be excluded successfully. Apart from the absorption proviso, the noise must also be constant in character, so that the averaging process is valid.

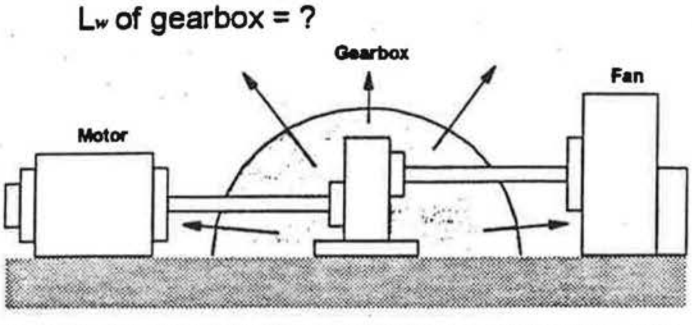

Figure 9 Sound intensity allows the sound power of this gearbox to be determined, even though the motor and fan are generating noise.

The shape and size of the enclosing surface are not important, but guidance is available in the standards to ensure that the sound field is not too complex, or influenced by general background noise.

7.2 *Source location*

There are two forms of source location using sound intensity, one using 'live' measurements and one using post—processing.

On-line measurements take advantage of the directional characteristics of the probe. As the two microphones are measuring the component of intensity in line with the microphone axes, the sensitivity of the probe drops according to a cosine characteristic with angle off–axis, reaching a null point at 90° off–axis. Intensity from the front of the probe is termed positive and from the rear, negative intensity. These will be displayed on the intensity analyser in different colours or directions (Figure 10).

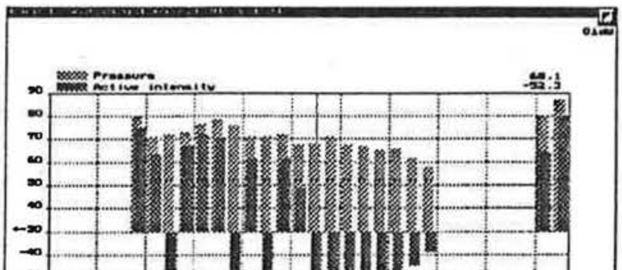

Figure 10 Example of sound intensity display showing positive and negative components

The characteristic of the probe has the appearance of two squashed tennis balls, and the null point in decibels is extremely sharp (Figure 11). We can therefore move the probe around in the field, while watching the analyser screen, using the null point to track the intensity flow from a source, and hence locate the source itself. Although this is a very

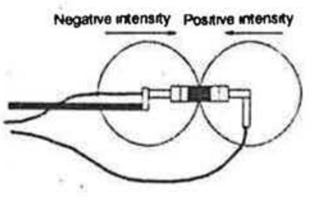

Figure 11 Directional characteristics of the intensity probe

qualitative method, it can give great insight into, for example, acoustic leaks in duct work, or into poor insulation treatment in panelling/ceilings.

Post-processed measurements involve measuring the intensity at discrete points on a surface, and mapping the results using a computer, which can generate contour, colour scale or three-dimensional maps, with interpolation between the grid points.

This can reveal sources, and indicate where further investigation is required. Additionally, the data can be represented as vectors, showing the relative strengths of the sources (Figure 12).

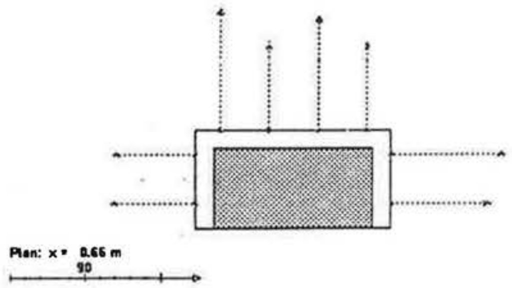

Figure 12 Vector representation of sound intensity

The downside of these techniques is that there is no background noise cancellation as in the sound power determination, as the surface does not enclose the source. However, used with care, these techniques can be very enlightening.

7 Standardisation

Fortunately, much measurement experience over the last ten years has been incorporated into national standards, to ensure that new users can quickly and easily take valid intensity measurements, and evaluate the accuracy of their results.

The first standard, *BS EN 61043:1993*, identical to *IEC 1043*, covers the performance of the analysis system itself, addressing issues such as phase matching of the probe and analyser, calibration and calculations. The system is split into 'probe' and 'processor' and two classes of instrument are defined with measurement tolerances. An important part of this standard covers the measurement of the residual pressure–intensity index, and hence the dynamic capability of the system. Manufacturers will in future quote the accuracy of their systems according to this standard, as they do with sound level meters. For most measurements it is advisable to choose a Class 1 probe and processor.

Currently FFT analysis systems have a special classification system according to their real-time capability and synthesis accuracy, and this reflects the wide spread in performance in the FFT analysis market. Good-quality FFT analysis systems will achieve excellent performance for intensity measurements, as good as if not better than that of digital filter analysers.

The second standard, *BS7703:1993*, identical to *ISO 9614* Part 1, is a procedural standard for the determination of sound power using sound intensity measurements. It provides a framework for accurate measurements, and guides the user through the procedures for ensuring that the control surface is adequate, that background noise is controlled and that 'hot spots' on the source are measured adequately. This standard has been criticised by many for its complexity, but it ensures that all the compromises and dangers, described earlier in this article, are kept to a minimum.

The latest analysis systems for sound intensity now reflect this standard, and PC-based software gives the user immediate access to the 'indicators' calculated in the document (Figure 13).

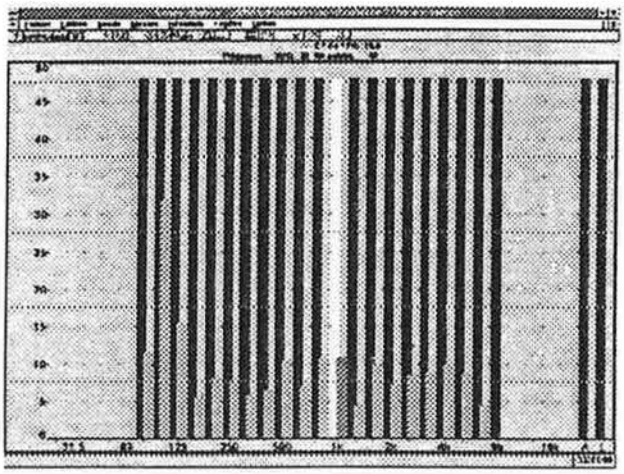

Figure 13 Display of test criterion F2 from the BS7703 sound power standard

All serious users of sound intensity data should acquire these two standards, and familiarise themselves with the basic theory of sound intensity. Further advice can also be obtained from responsible equipment manufacturers.

8 Summary

In this article we have examined the basic theory of sound intensity measurement and discussed its application, in particular to sound power determination. It is fair to say that because two channels of measurement are involved, the subject is probably more than twice as complex as simple sound level measurements, with, say, a sound level meter. Unfortunately, this also means that the equipment is relatively expensive, and short cuts can rarely be taken. However, standardisation and good-quality measuring systems now make it possible for the technique to be used more widely, yielding results, where before it would have been impossible or uneconomic to take any measurements.

A new generation of machines is appearing on the market, designed with the help of sound intensity measurements, giving an edge in performance and reduced noise radiation. This trend will only increase with the issue of further European Union directives covering the sound power of machinery in the building services arena and elsewhere.

To be heard but not seen: Public address as an integral element in the building services package

K Dibble (Ken Dibble Associates)

1 Introduction

It is unfortunate that the very mention of the term 'public address'conjures up in most peoples' minds recollections of the ubiquitous garbled and distorted 'railway station tannoy'.

What is amazing is that British Rail has never even recognised as a problem the abysmally poor quality of the speech communications facilities on most of its stations. This situation is unfortunately compounded by the fact that in the great majority of building projects, the provision of a public address system is at the top of the 'sacrifice list' when the money runs out. It is rarely accorded more than cursory attention by the mechanical and electrical services consultants. The architect is concerned only that the loudspeaker grilles do not clutter up the geometric pattern of HVAC nozzles and luminaires across the designer ceiling. Yet it is likely that neither the Hillsborough nor Bradford disasters would have resulted in anywhere near the numbers of fatalities they did had adequate public address facilities been used.

2 That word...

The word 'Tannoy' is in fact a proprietary trade name — like 'Hoover' or 'Biro' etc. It does not appear in the Concise Oxford Dictionary and is not recognised by any word processing thesaurus or spellcheck software known to the author. It appears to originate in the early warning systems used on Royal Air Force World War 2 airfields, and to relay the BBC Home Service into wartime factories, with the manufacturer's name emblazoned around the horn rim or across the baffle in oversized letters. Its adoption into common parlance is probably due to its onomatopoeic association with the old sailor's hail shouted through a hand-held megaphone: 'Ahoy!'. 'Tannoy!' has very similar connotations, and of course the horn flare of the Tannoy public address loudspeaker has obvious physical similarities to its historical forebear. This misuse of English grammar even extends to the use of the word as a noun, as in 'a tannoy', or as a verb as in 'can you tannoy the security guard for me?'

Be that as it may, 'public address' or 'paging' are the proper terms according to the application. Public address is just that — a means by which an address or an announcement may be heard at a large gathering; a system intended as a means of locating personnel or the giving of instructions or information is clearly a 'paging system'.

Either type of system can also be used to relay music, but this is usually a secondary consideration and unless the system is specifically designed with music in mind, the music quality invariably reflects this status. *BS6259 Planning and Installation of Sound Systems* recognises at subsection 7.3 three distinct categories of system:

(*a*) Type 1 systems aim at the highest possible quality of reproduction. Use of Type 1 is not usually justified unless the audience is expected to be critically interested in the programme for its own sake and not merely as a background. It is essential that the listening conditions be very good acoustically. It is also essential that the inputs to this system be of adequately high quality, because any defects will sound more objectionable. Poor signal quality from an amplitude modulated broadcast receiver, for example, may cause the reproduction from this system to sound worse than that obtained from a Type 2 system.

(*b*) Type 2 systems are those most commonly used. The effect desired is that of intelligible and natural sounding speech, and of musical quality that the average listener considers pleasant. It is desirable that the listening conditions be either fairly good acoustically already, or that they can be made so by acoustic treatment of an acceptable kind.

(c) Type 3 systems aim mainly at the reproduction of speech of good intelligibility but not necessarily a high degree of naturalness. Music, if reproduced at all, would be of an acceptable quality but not necessarily of artistic merit. Where acoustic conditions are bad, e.g. because noise levels are high or reverberation excessive, this system should provide the best that can be achieved in the conditions available.

3 Intelligibilitv

There is no possible technical excuse for the abysmally poor speech intelligibility which we have to endure from many public address systems in use today. I use the word 'intelligibility' rather than 'quality' as that is invariably the object of the exercise — as now recognised by *BS7443 Sound systems for emergency purposes*, in which a speech transmission index (STI) of 0.5 or greater is called for. Speech intelligibility can be measured in several ways but STI, or its simplified variant RASTI (Rapid STI) is the method most widely used and is the one recognised by the standard. The method uses a scale of 0.1 to 1 as shown in Figure 14, thus providing an objective measure of the performance of a public address system. *BS7443* calls for a minimum RASTI of 0.5. This should be achievable and exceeded in most areas by correct speaker placement. RASTI is measured at two specific frequencies, whereas a full STI is measured at nine. There is a groundswell of opinion in the audio/acoustics profession that RASTI is insufficiently reliable to be enshrined in a standards document which is frequently called up in tender specifications for very large—scale emergency systems. However, this highly technical argument is at present wending its weary way through the international standards—making process, and is outside the scope of this article.

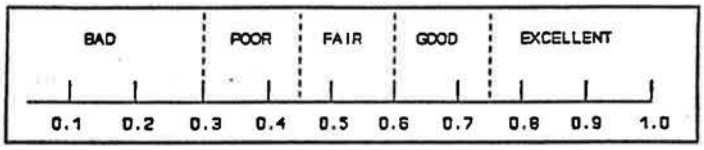

Figure 14 RASTI (Rapid Assessment of Speech Transmission Index) for sound system equipment

The term 'quality' as used in the 'hi-fi' context implies a natural and realistic reproduction of the original sound — which is indeed necessary for music or drama, but can actually detract from good intelligibility in a speech context. This situation is clearly recognised by subsection 7.3 of *BS6259* as earlier cited, and is quantified by Figure 4 from that same standard, which is reproduced here as Figure 15. It is a golden rule in public address is that a PA-type loudspeaker which has an intentionally restricted bandwidth is not connected to a full-range audio amplification system, since the 'out-of-band' energy will overload the loudspeaker, cause overheating, and result in acute distortion into the bargain! For this reason any properly engineered public address distribution system will incorporate filtering to limit the system response to something along the lines of the *BS6259* recommendations shown in Figure 15, or with a still more rigorous high-pass characteristic, depending on the type of loudspeaker being used and the acoustics of the environment. In onerous conditions, filters of 12 dB/octave with turnover frequencies as high

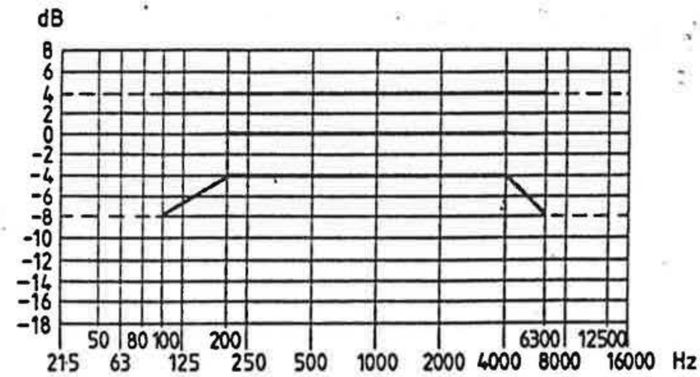

Figure 15 Recommended axial frequency response limits for loudspeakers for Type 2 systems (0 dB is the mean characteristic sound pressure level measured over the range 200 Hz to 4kHz.)

as 400 Hz have been used to overcome particular acoustic difficulties.

4 The influence of acoustics

It is this factor, probably above all else, that will determine the choice of system to be provided and how effective it will be in a given environment. The choice of loudspeaker device, the power needed to drive it, the bandwidth requirements, placement and aiming, the maximum length of throw achievable, will all be dictated by acoustic considerations — yet most systems are installed without regard to such criteria, usually by installers with no knowledge of the science of acoustics, and often without a decibel meter in sight! Is it any wonder that most systems fail to deliver?

In essence, the attainment of a given STI goal is dependent upon the transfer function between the source and the receiver. In this case the source is a loudspeaker device, and the receiver is the ears of the listener. The transfer function concept is shown at Figure 16 and the principal parameters affecting the result are as follows:

(a) the reverberation time (RT60) of the space
(b) the presence of discrete echoes or reflections
(c) the background noise level
(d) the Q of the source
(e) the loudness of the source
(f) aiming of the source
(g) the source-to-receiver distance
(h) the number of sources present
(i) the physical distances between sources

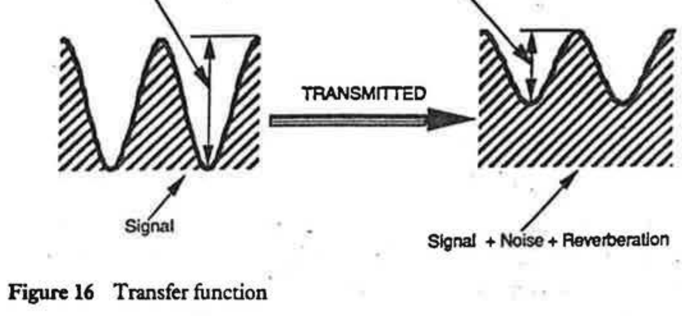

Figure 16 Transfer function

(j) the system bandwidth

(k) the system distortion

(l) the diction of the person using the microphone.

The prediction and measurement of speech intelligibility is a complex matter which would take this entire article to explain. For this reason it is usually carried out using computers and specialised measurement systems. Instead I propose to illustrate good design practice in the context of the factors which affect intelligibility using some practical examples as follows.

5 A typical distributed ceiling system

Consider a typical open—plan office or large hotel lobby environment. We usually find a suspended acoustic ceiling, wall-to-wall carpeting, soft furnishings, etc. Because most of the room surfaces are acoustically absorptive the reverberation time (RT60) is low — typically 0.3–0.5 seconds depending on the room volume. Also there are likely to be few significant discrete reflections or echoes and the background noise level will also be of a low order — typically 65/70dB(A). So half the problems are immediately removed from the equation. Also, a ceiling system has a significant advantage over other loudspeaker options in that listeners, even when moving around, are normally at a fixed distance from the sources, thus eliminating any problems associated with variations in the source-to-receiver distance. So all we need to do to ensure uniform coverage — architect permitting — is to space the loudspeaker devices on a grid so that the dispersion angles intersect at a typical standing or seated ear height according to whether we are designing for an office or lobby environment. Figure 17 shows the geometry of this arrangement based on a 3 m ceiling height and using commonplace ceiling baffles fitted with cone drive units of 200 mm diameter. Most such devices will exhibit a nominal 90° dispersion angle, which produces a spacing formula as follows:

$$S = 2(H - 1.5)$$

for standing listeners, or

$$S = 2(H - 1.3)$$

for seated listeners, where all units are metres.

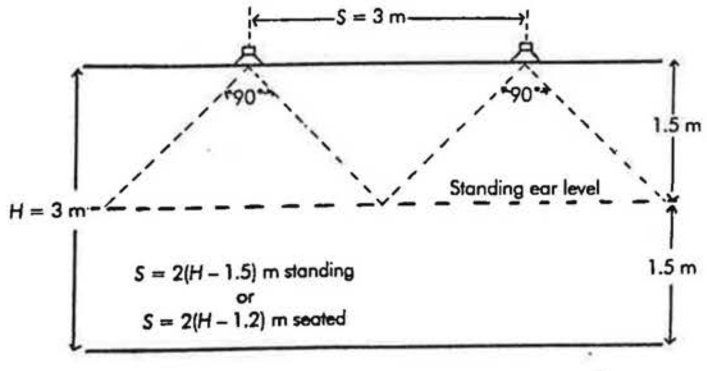

Figure 17

Note that in this example, the formula shows that the loudspeakers require to be just 3 m apart. This is about half the spacing which would be 'guesstimated' in a typical intuitive solution, and would thus quadruple the number of loudspeakers required, with a consequent increase in cost. So we immediately run into the problem of competitive bidding where the tender specification fails to qualify properly the required performance criteria.

Given that dispersion angle is a function of wavelength and source diameter, one way of increasing the spacing is to use smaller loudspeaker devices, as these will have a larger dispersion angle. Figure 18 shows the effect of using a 100 mm diameter ceiling baffle with a nominal dispersion angle of 120° in the same situation. It can be seen that the spacing is increased to 5 m and the spacing formula becomes:

$$S = 3.3(H - 1.5)$$

for standing listeners, or

$$S = 3.3(H - 1.3)$$

for seated listeners, where all units are in metres.

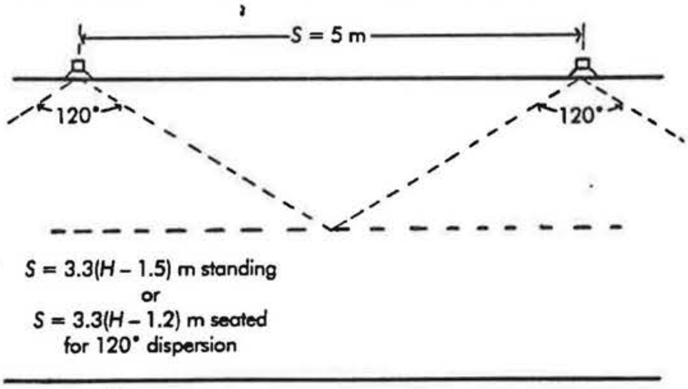

Figure 18

Clearly then, by using a smaller loudspeaker drive unit, the spacing grid dimension is increased and the number of drive units is reduced, with a consequent reduction in amplifier power and labour costs for installation.

The next part of the design process is to determine the power input required by the loudspeaker. This is a function of the sensitivity of the particular loudspeaker device to be used, the source—to—receiver distance and the background noise level. As a rule of thumb we would normally design for a speech level which is 10 dB above the background noise level. In our example that means we need to achieve a sound pressure level (SPL) of 70 + 10 = 80 dB at 1.5 m from the source.

The sensitivity figure is derived from the manufacturer's data sheet. A typical value for a ceiling baffle loudspeaker is 90 dB. Unless otherwise stated, this means that the device will produce a sound pressure level of 90 dB for 1 watt input at a distance of 1 m from the drive unit cone or diaphragm.

Given this information the following simple 'inverse square law' distance attenuation formula can be used to calculate the sound attenuation over any required distance:

$$\text{Attenuation} = 20\log D$$

where D is the source—to—receiver distance in metres. For example, at 2 m the attenuation will be 20 log 2 or 6 dB, at 4 m it will be 12 dB and at 8 m, 18 dB. It can be seen that this gives a useful rule of thumb that sound attenuates by 6 dB each time the distance from the source is doubled.

To return to our example, 20log 1.5 gives 3.5 dB, which, added to the SPL requirement of 80 dB, gives 83.5 dB as the required source level. Given that half a decibel is not aurally detectable as a level change, this is a good opportunity to do some rounding, so we shall take 84 dB as the required source level.

To convert directly between dB level differences and input watts we can use the standard dB power ratio formula:

$$10^{\text{dB level difference}/10}$$

The level difference in our example is 90 – 84 = 6 dB which gives a power ratio of $10^{0.6}$ or 4.

Thus the power input requirement is a factor of 4 below the reference, or one quarter of a watt. There is another useful rule of thumb emerging here, in that halving or doubling the input power will always produce a 3 dB decrease or increase in SPL. Similarly a fourfold change will produce ±6 dB, an eightfold change ±9 dB, etc.

Most non-specialist specifiers would look at a ceiling baffle in a 3 m high ceiling and 'guesstimate' a power requirement of about 2W per loudspeaker, whereas we have shown by calculation that this would be an eightfold overestimate. Multiplied up to large system proportions, consider the cost implications of providing an 8 kW amplifier rack when just one amplifier rated at 1 kW is all that is required! So in addition to providing uniform loudspeaker coverage, here we have a further example of the benefits of adopting an engineering approach to system design.

6 The effects of reverberation

The general approach illustrated in the previous example can be applied to almost any type of loudspeaker installation provided the source-to-receiver distance and background noise level are not excessive, and the acoustic environment reasonably 'dead', and would produce an STI rating well above the 0.5 minimum set by *BS7443*. Once we enter into a high-noise or highly reverberant environment — e.g. an aircraft hangar, a large exhibition hall or arena, a cathedral, a machine shop or industrial assembly plant, then we enter an altogether different 'ball game'. Figure 19 illustrates such a situation where a small re-entrant horn with a maximum output level of 110 dB is used in a large space where the source-to-receiver distance is 50 m and the background noise 80 dB. Using the 20 log*D* formula, the maximum achievable direct sound level at 50 m is only 76 dB, which is 4 dB below the background level. This would be just audible but not decipherable as speech.

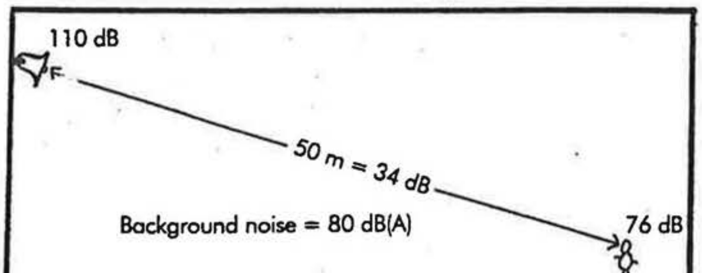

Figure 19 'No go'

Simply put, the problem lies in achieving the 10 dB headroom between the speech level and the background noise level. In this context, the background noise is an amalgam of the actual noise in the room plus the residual noise floor created by its own reverberation component, plus the reverberation generated by the speech signal itself. To illustrate the considerable influence of reverberant energy, Figure 20 shows the effect of the same hand clap in a relatively 'live' room and in a relatively 'dead' room.

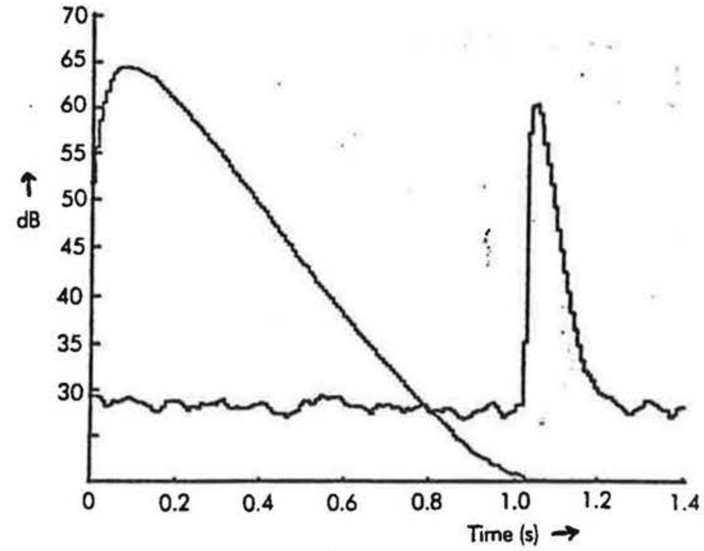

Figure 20 Handclaps in two rooms

The differences in the resultant energy—time curves are clear to see. Also, it is important to grasp the perception of reverberant sound as noise. In this context Figure 21 shows the attenuation with distance of the direct sound (the straight-line curve), compared with that in rooms with RT60 values of 0.5, 2 and 8 seconds. Note that in the 8 s room, the noise floor due to the reverberant energy alone is only 12 dB below the original source level after only 6m. In this type of instance it is not possible to achieve 10 dB headroom when the actual background noise level is also taken into account and compromises have to be sought.

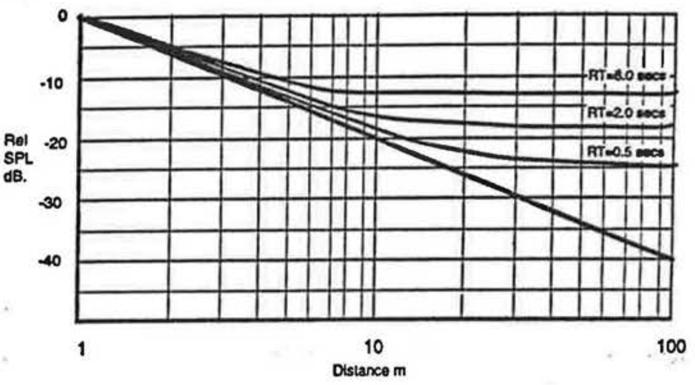

Figure 21 Sound field in a space for different reverberation times

Figure 22 illustrates the importance of the effect of reverberation on speech. It shows the greeting 'Good morning', broken

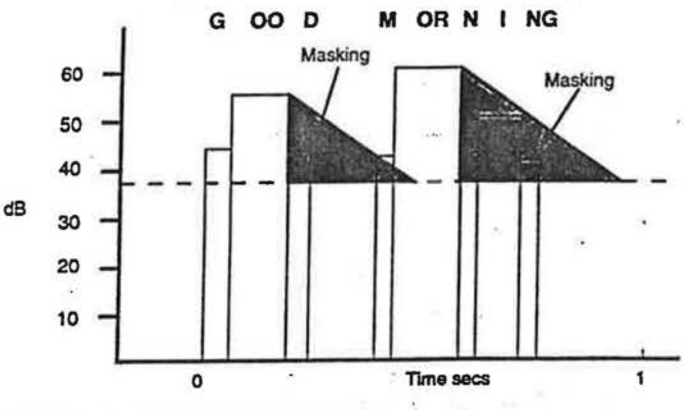

Figure 22 Effect of reverberation on speech

down into energy/time syllables and superimposed on a constant background noise of level 38dB. Because reverberation is usually frequency dependent the 'G' consonant does not excite a large amount of reverberant energy, and any that is produced is immediately masked by the full and rounded 'oo' sound. Being lower in frequency and of greater amplitude this generates significant reverberant energy which hangs on in the room to mask the 'd' of 'Good' and the opening consonant of 'morning'. This reverberant envelope has hardly subsided when along comes another in the shape of the 'or' syllable from 'morning', which completely masks 'ning'. Consequently the two words are merged into one and the greeting is heard as:

GOOOoORRING

Because of the brain's ability to decipher incomplete information by filling in the gaps, and due also to the effects of anticipation, some listeners may still be able to understand what was being said, although the STI rating would be about 0.3, at the border between 'bad' and 'poor' on the scale shown in Figure 14. So what is the solution?

Figure 23 shows the behaviour of a typical loudspeaker in a reverberant space and the effect when such a loudspeaker is placed in a room. Significant reduction in the unwanted reverberant component can be obtained by restricting the bandwidth fed to the loudspeaker to the 'useful wide-band component' segment shown. Given however that the speech bandwidth must extend down to 400 Hz, and that most ordinary loudspeakers are not directional at this frequency band, in practice the solution is to employ a specialist type of loudspeaker known as a constant-directivity horn. These are available in a variety of dispersion angle formats and are physically large and expensive. The design process involved in the selection and deployment of such a device is well outside the scope of a general article. What can be said, however, is that used with appropriate skill and expertise such devices are capable of meeting the 0.5 STI requirement within its 'useful wide-band component' segment and in some fairly hostile acoustic environments.

Figure 24 shows a computer-generated prediction of the signal-to-noise ratio and direct-to-reverberant ratio combinations which would be expected to achieve 0.5 STI or better in an RT60 of 1 s (upper) and 2 s (lower), but satisfactory results have been obtained in cases where the reverberation time extends to 8 s.

7 Other solutions

A number of electronic aids are also available to help in overcoming some of these difficulties, and each has a part to play if used in the right environment and for the right reasons. A

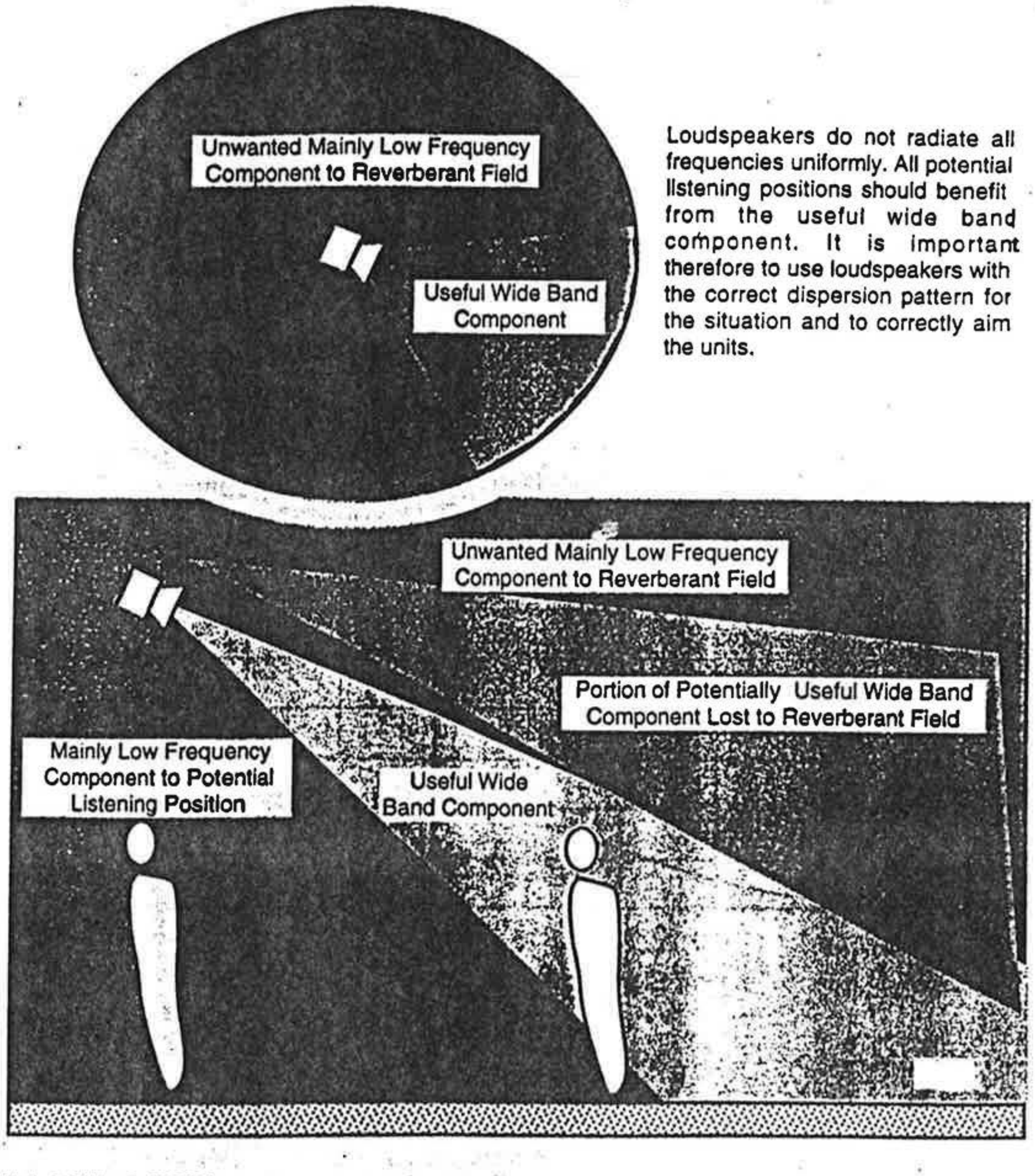

Figure 23 Behaviour of a typical loudspeaker in a reverberant space and the effect when such a loudspeaker is placed in a room

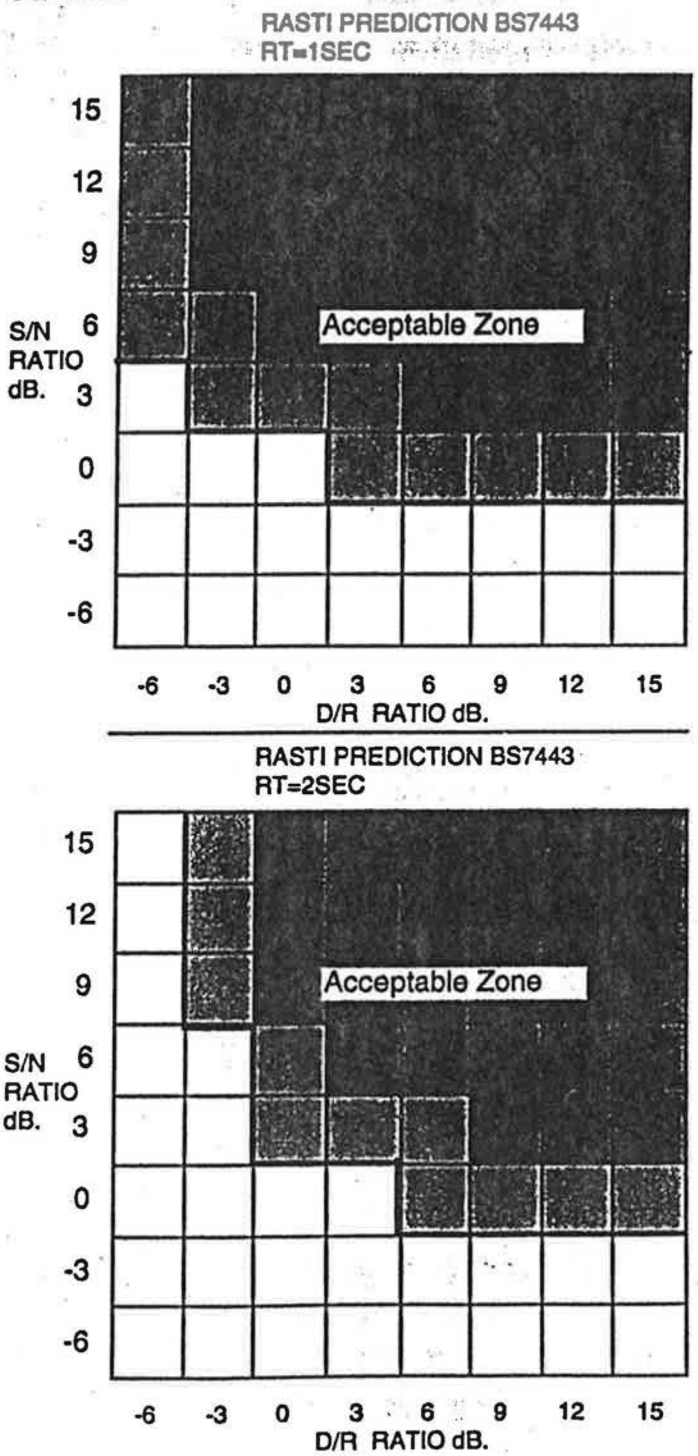

Figure 24 RASTI predictions according to BS7443

graphic or parametric equaliser, for example, will enable the spectral response of the loudspeaker system to be tuned to minimise the effect of some room resonance or to maximise headroom above the background noise spectrum. Also useful in regulating signal–to–noise conditions are automatic gain control devices which adjust system gain to maintain a specific amount of headroom above the background noise level. This is particularly useful in areas where the background noise level is likely to vary — for example a plant room with varying duty cycle, an airport concourse, railway stations, etc.

Another problem in providing a consistent performance is the variability in announcer competence. This can be largely overcome by the use of microphone processors which automatically adjust the system gain according to the proximity and/or loudness of the announcer's voice, and incorporate frequency—dependent compressors and/or limiters to prevent over—modulation and distortion.

8 Concluding comment

Audio is one of those subjects of which everyone seems to have some level of knowledge, so I have assumed that readers of this publication will already have a certain basic knowledge of public address systems. Against this assumption I have selected what I consider to be the key elements of system design and treated these to illustrate the level of engineering which is necessary if the design aims are to be achieved. Clearly there will be many more aspects of public address design that have not even been touched on in a brief review article. But I hope that this will have given some insight into what is involved and shown that there is a lot more to effective public address practice than stringing a number of loudspeaker baffles across a suspended ceiling. Other aspects which should receive equally critical attention include the control protocols, routing and zoning, distribution line losses, amplifier loading, microphone response characteristics, system surveillance, standby power supplies, integration with fire alarm systems and compliance with the *BS5839* fire alarm standard.

In my opinion the time is past when the building services engineer embraced all services under a single remit, with electrical and HVAC services, lifts, escalators and communications all lumped together as one unwieldy package. In a building where public address — either as a stand-alone provision or integrated with an evacuation and/or fire alarm system — is an identified client requirement, then the design and specification of such systems should clearly be entrusted to experts. A list of consultants specialising in this field can be obtained from the Association of British Audio Consultants, and a design/build proposal may be obtained in confidence from any installer listed under the Sound & Communications Industries Federation's 'Approved Contractor' scheme.

Acknowledgement

The assistance of AMS Acoustics for permission to reproduce Figures 21 to 24 is gratefully acknowledged.

Masking speech and noise with masking sound

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1 Introduction

A great majority of commercial and public premises are open—plan styled offices where a large number of people are grouped into a large working unit which may or may not have to welcome the general public and clients. This is the case, for instance, in insurance companies, public administrative bodies and in certain open bank premises — in Switzerland in particular, the open premises are becoming more usual. Figure 25 shows an illustrates a typical open—plan reception area often found in banks. In these large halls conversation, including that on the telephone, can often be either confidential or a source of disturbance to the other people. It is therefore essential to take adequate measures to attenuate as far as possible the negative effects of conversation without going so far as to restrain, hinder or maybe even prevent them from taking place. In modern buildings, the protective measures taken both against external noise — road and rail traffic or industrial noise, for example — and internal noise — due to activities being carried out in adjacent rooms or due to the plant in the building — are such that the level of ambient noise, with no activities going on in the room concerned, is sufficiently low — typically of the order of 35 A-weighted decibels (dBA) or even less — so as no longer to contribute to the masking of conversations, as was generally the case previously. Acoustic conditioning of the premises has therefore to be considered, in the form of deliberate emission of other noises or masking sounds, acceptable to the people round about. The general objective is to protect the confidentiality of conversations, to prevent them from interfering and to decrease the disturbance caused.

In the open premises considered, and generally speaking in all offices, certain noises cannot be avoided: telephones, printers, typewriters etc. These noises are of a particularly disagreeable nature, even at sound levels below 50 dBA. This is mainly due to their time characteristics — they 'burst in' and are unpredictable — but it is also due to their acoustical structures which accentuate the disturbance — whistling sounds, very marked tonal components, percussion sounds, shocks, rhythm and repetitiveness. The diffusion of background sound in these premises, depending on their proper design, is an efficient way to decrease the disturbance caused by these noises, still using the masking effect.

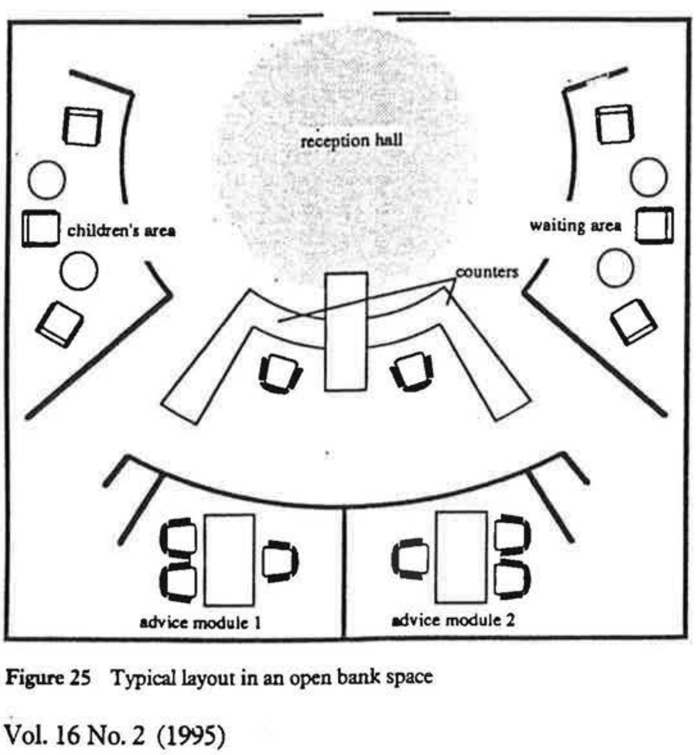

Figure 25 Typical layout in an open bank space

In certain cases, noise originating from the aerodynamics of ventilation and air conditioning installations can be enough conditioning by itself. These are stable wide-spectrum noises, without timbre or tonal pitch — they are judged to be more or less low or high pitched — resembling from this point of view 'white' or 'pink', noise, and are therefore very neutral to the ear. However, they provoke strong reactions and are rejected as soon as their level exceeds a value of 40–50 dBA.

The best solution is acoustic conditioning through the use of background sound diffused by a loudspeaker array playing specially recorded music, because it allows a sufficiently high level to be used in order efficiently to mask any undesired acoustic events and conversations without disturbing people. The problem is to know what musical material to diffuse.

2 Problem and approach

It is relatively easy to define the technical characteristics of a sound system for background 'music' — for example the sound level and spatial distribution — and therefore to select the required audio equipment — sources and amplifiers, type, location and layout of loudspeakers, etc.— but it is much more tricky to define the sound content to be diffused. Synthetic electronic noises— 'white' and 'pink' noise, etc. — and radio programmes are often poorly perceived and accepted. In practice, unless due care is taken, any musical material or ambient sounds diffused always provoke negative reactions, with the consequence that the sound system is turned off or the level is so reduced as to be of no use. Finding suitable material which will be accepted well by the people concerned is therefore crucial. For this reason we advise following the procedure set out here.

(*a*) Define the technical specifications of the sound system from the acoustic conditions prevailing in the premises, and from the masking requirements—this fixes the technical objectives and leads to the choice of equipment.

(*b*) Propose trials during a two— to four—week period with a number of pilot recordings, containing a wide spectrum of musical extracts covering all types of music, but chosen according to criteria given below. It is possible in this way to define peoples' needs more clearly.

(*c*) Gather all the reactions, impressions and comments of the people concerned, both personnel and clientele, through both discussions and questionnaires.

(*d*) Based on the reactions and information collected, make the final recordings.

3 Selection criteria of sound content

The difficulty is to make the content acceptable to the people concerned. There is no universal choice which will suit everyone, all the more so because the type of activity being carried out in the premises comes into the equation. A number of criteria can be extracted which help considerably in the choice of material:

(*a*) The material should be composed of a main background sound — music — and a secondary background sound — ambient noise. The purpose of the latter is to fill in the inevitable pauses in the music, or at least the pianissimo passages.

(*b*) The equivalent level L_{eq} — or mean energy — of a piece of sound material for the people in the area to be covered must lie in the range 50–55 dBA, with a peak factor (difference between the maximum and equivalent level) below 6 dBA. This means that the sound level is much lower and fluctuates much less than when listening normally to recorded music. The second requirement nearly always implies compressing the dynamics at the recording stage — the *fortissimo* passages are 'crushed', without distortion of course.

(*c*) The mean spectral energy density of a piece of sound material must decrease at least as fastä as a pink noise (–3dB per octave). In other words, the amplitude of high-pitched sounds is reduced as compared with that of low-pitched ones. This condition may require a low-pass filter during recording.

(*d*) The equivalent level Leq of the secondary background material must be at least 10 dBA below that of the main background material. The reason is that it must not cover the latter.

(*e*) The main background material must have a certain number of musical characteristics — such as slow to medium tempo, lightly accentuated rhythms, both melodic and rhythmic continuity, no extremes (exuberance, excessive seriousness), and soft and round sounds. In this way, its loudness is kept to a minimum compared with that of other types of sound. In view of the fact that these characteristics are more easily to be found in classical music, the latter is therefore the principal source of the main background material; other types of music such as, for instance, jazz are not, however, completely excluded.

(*f*) The main background material should be made up of scarcely known works and pieces, i.e. music that belongs neither to folklore (popular refrains), fashionable hits, nor the great repertoire most appreciated by the general public. Rejection through 'saturation' must be avoided at all costs. This excludes in particular many works from the great composers — *Eine kleine Nachtmusik, Les quattro Stagioni* etc. should be avoided.

(*g*) Preference should be given to works for instruments which allow the production of sustained sounds, without any aggressive characteristics, such as strings, wood and reed instruments and possibly the organ or synthesisers.

(*h*) Percussion instruments and brass stabs should be avoided. If the work chosen contains any such sounds, they must be eliminated or attenuated during the recording.

(*i*) Changes between different styles should be programmed — there should at least be variations in the instrumentation, tempo, nuances, etc. — with each style lasting about twenty minutes.

(*j*) Soft and smooth transitions should be inserted between the different styles, using interludes lasting between one and three minutes. One way of doing this is to keep the same well chosen secondary background material, for which the level will have been increased.

(*k*) The secondary background material should be made up of natural ambient sounds, such as sounds of waves, surf, rivers, etc. Exotic and unusual sounds should not be excluded — the chirping of birds and crickets for example — but in this case they should be used with discretion.

Recordings for open premises in banks were made applying the above criteria. Both classical music and jazz (reed instruments) were used. Thanks to compression at the recording stage, their equivalent level was 53 dBA. Subjectively the loudness is less than that of a 'pink' noise of the same equivalent level, thanks to appropriate choice of musical extracts and dynamic compression.

The recordings were tested *in situ* in order to check their appropriateness and acceptability. The opinions of the personnel and clients were, overall, positive.

4 Conclusion

It has been demonstrated that a background sound system efficiently masks speech and disturbing noises, while being well accepted by the people concerned. In order to achieve this objective, and in addition to respecting the acoustic conditions, the material recorded must have the characteristics defined in this article. The approach outlined should of course be adhered to; that is, trials of a certain number of pilot recordings should be proposed, and only after the reactions have been assessed should the final sound material be recorded. In this way, the recordings will be adapted to the local conditions not only from a technical, but more importantly, from a human point of view.