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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 1

Editorial

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

A glance at the contents list of any recent issue of *Building Services Engineering Research and Technology* will reveal that the majority of papers submitted deal with thermal and air movement topics. It is the stated intention of the current Editorial Panel to extend the subject coverage to be more representative of the interests of the building services engineer. One area of interest to all practitioners acoustics and noise control, and a considerable amount of work of relevance to the profession is being carried out. However, this work is normally published in specialised acoustics journals where it is unlikely to be seen by the typical building services Engineer. This issue of the 'Blue Pages' seeks to remedy this deficiency by presenting a number of specially commissioned brief articles describing some recent developments in the general area of the acoustics of building services.

A major source of concern is flow-generated noise. Alan Fry sets out the current position. Another problem encountered by the practitioner is that of structure-borne sound from machinery. Andy Moorhouse describes current work aimed at producing predictive techniques for structure-borne sound to complement those now existing for airborne sound. Advances in digital electronics have made available very powerful low-cost computers and Colin McCulloch describes how these can be applied to the solution of noise problems. Finally in Part 1, Geoff Leventhall and his co-workers then give an account of yet another result of the digital revolution, namely the development of active systems of noise control.

Flow and its noise in heating and ventilation engineering

A Fry (Sound Attenuators Ltd)

1 Introduction

In the following article we shall be discussing the turbulence introduced into controlled air flow, but it should be remembered that even the smoothest airflows encountered in the building services industry will be turbulent, albeit in a minor way. We shall address extra and gross turbulence over and above the basic flow. The streamlines in many of the illustrations do not really exist, but simply indicate the paths of the minor turbulence.

As an initial summary it is worth stating that to control airflow quietly, it is necessary to employ guided diversions, as with turning vanes, rather than objects of obstruction which attempt to buffet the airflow into a revised distribution. Partial but controlled deflection often serves as a compromise in application. Next best to this is an awareness of the noise consequences of any necessary duct features. Generally flow noise is of a broad-band or characterless nature and, as long as it is not too loud, is reasonably acceptable as a background sound.

As shown in Figure 1, when flow occurs around an obstacle — in this case a rod or wire — then the turbulence comes from alternate sides as whirls or vortices. These vortices emit broad-band sound, but in a fairly inefficient manner. When vortices are shed from objects they produce a back reaction on the object which is set into forced vibration, resulting in radiated noise. An example would be fan blades and, as with

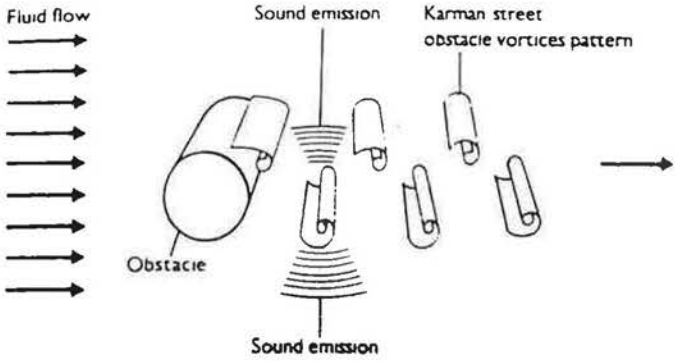

Figure 1 Formation of turbulence

musical instruments, larger objects are more efficient at radiating lower frequencies.

Nevertheless, it will also be noted that the shedding is rhythmic, travels near to the flow speed, and that this shedding rhythm can excite a resonance in the obstacle which then radiates a purer tone quite efficiently. An everyday example of this is the sound of 'singing' wires in windy environments - Aeolian tones. From the building services industry, Figure 2 shows the noise from 13 mm support stays in a 600 mm duct at 25 m s^{-1}. Tones peaking in the 500 Hz octave band are produced by the rods, and in this case it should be noted that the duct dimensions of 600 mm correspond to a wavelength of sound in this 500 Hz band.

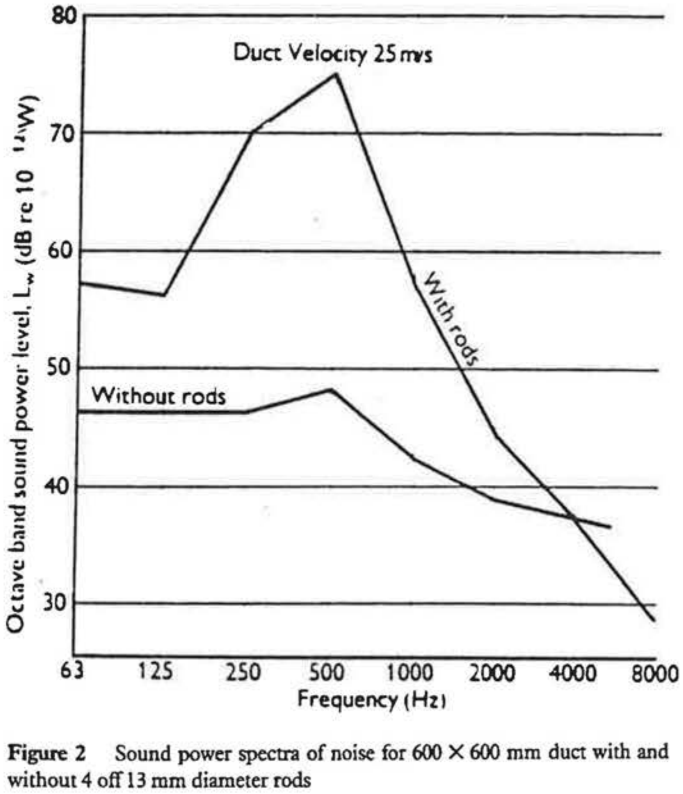

Figure 2 Sound power spectra of noise for 600 × 600 mm duct with and without 4 off 13 mm diameter rods

2 Damper noise

Since dampers are one of the few devices expected to produce noise as a result of their control duty and deliberately obstructive nature, they would seem to be an appropriate place to start. As a variable-geometry device their sound levels are difficult to predict and manufacturers' measured data should be used where available. Figure 3 shows such a table. However, attempts are reported in the literature to predict the frequency dependent noise levels as a function of the damper size, flow rate and pressure loss. A limited but useful awareness of the mathematics can be gained from Sound Research Laboratory's book *Noise Control in Building Services*.

Figure 4 shows the disturbed and less rhythmic turbulence understandably generated by a throttling damper blade, and it is appropriate at this point to mention some basic links between flow speed, sound spectrum and the size of the obstacle or opening. Putting aside any resonant effects and the noise character they may generate, the broad-band flow noise does show a broad peak in its noise spectrum as illustrated in Figure 5, which addresses the variables for a diffuser to be discussed below. In general: *The larger the object or opening, the lower is the frequency of this peak. The faster the flow rate, the higher is the frequency of this peak.* So, returning to dampers, a large single-blade butterfly damper will generate more low frequencies than a multiblade unit — a point worth remembering when noise control procedures are to be applied, as low frequencies are more difficult to attenuate than middle frequencies.

3 The system

3.1 Ductwork

Ductwork most commonly occurs in its simplest, straight form. The flow noise associated with this is generally negligible compared with that from all other features of a system. For guidance an NC sound power level of 40 dB per square metre of duct cross section may be taken for a flow speed of 10 m s^{-1}. Further to this, the scaling laws are 18 dB for each doubling of flow speed and 1.5 dB for each doubling of cross section area.

Table 1 gives guidance for low-velocity systems, but also includes allowances for the whole system. Its data should not therefore be associated with duct wall noise alone.

Table 1

NC design level	Main ducts (m s^{-1})	Branch ducts (m s^{-1})	Final run-outs (m s^{-1})
20	4.5	3.5	2.0
25	5.0	4.5	2.5
30	6.5	5.5	3.25
35	7.5	6.0	4.0
40	9.0	7.0	5.0

3.2 Transitions

Ducts of different cross section are joined by transformation pieces. Gradual rather than abrupt changes are beneficial both for low noise generation and for a low pressure drop. When a shape change is required to negotiate objects or other restrictions then a minimal change in duct velocity is arranged. The benefits of gradual or tapered transitions are considerable, as shown in Figure 6 for a velocity of 10 m s^{-1}.

3.3 Bends

90° elbow bends without turning vanes are a major source of low-frequency flow noise and of duct wall buffeting. The use of radiused bends is an improvement, but narrow-chord turning vanes are the most frequently adopted solution for 90° elbows. Figure 7 from the literature for a 200 mm square duct at 20 m s^{-1} is somewhat unrepresentative, but it does illustrate

Volume Flow Rate m³/s	Volume Flow Rate Correction dB	Single Opposed Blade Dampers SWLdB: Pressure across Damper N/m²	63	125	250	500	1000	2000	4000	Double Opposed Blade Dampers SWLdB: Pressure across Damper N/m²	63	125	250	500	1000	2000	4000
			Octave Frequency Bands in Hz								Octave Frequency Bands in Hz						
0.03	−10	981	75	76	73	75	75	71	67	981	79	80	80	79	78	74	68
0.03	−9.5	932	75	76	73	75	75	71	67	932	78	79	79	78	77	73	67
0.03	−9.0	883	74	75	72	74	74	70	66	883	78	79	79	78	77	73	67
0.04	−8.5	834	73	74	71	73	73	69	65	834	77	78	78	77	76	72	66
0.04	−8.0	785	73	74	71	73	73	69	65	785	77	78	76	77	76	72	66
0.05	−7.5	736	72	73	70	72	72	68	64	736	76	77	77	76	75	71	65
0.06	−7.0	687	71	72	69	71	71	67	63	687	75	76	76	75	74	70	64
0.06	−6.5	638	71	72	69	71	71	67	63	638	74	75	75	74	73	69	63
0.07	−6.0	588	70	71	68	70	70	66	62	588	73	74	74	73	72	68	62
0.08	−5.5	539	69	70	67	69	69	65	61	539	72	73	73	72	71	67	61
0.09	−5.0																
0.09	−5.0	490	68	69	66	68	58	64	60	490	71	72	72	71	70	66	60
0.10	−4.5	441	67	65	67	57	57	63	59	441	70	71	71	70	69	65	59
0.11	−4.0	392	65	66	63	65	55	61	57	392	69	70	70	69	68	64	58
0.12	−3.5	343	64	65	62	64	54	60	56	343	67	68	68	67	66	62	56
0.14	−3.0	294	63	64	61	62	62	58	53	294	65	66	66	65	64	60	54
0.16	−2.5	284	62	63	60	61	61	57	52	284	65	66	66	65	64	60	54
0.18	−2.0	275	62	63	60	61	61	57	52	275	65	66	66	65	64	60	54
0.20	−1.5	265	61	62	59	60	60	56	51	265	64	65	66	64	63	59	53
0.22	−1.0	255	61	62	59	60	60	56	51	255	64	65	65	64	63	59	53
0.25	−0.5	245	61	62	59	60	60	56	51	245	63	64	64	63	62	58	52
0.28	0.0	235	60	61	58	59	59	55	50	235	63	64	64	63	62	58	52
0.31	0.5	226	60	61	58	59	59	55	50	226	62	63	63	62	61	57	51
0.35	1.0	216	59	60	57	58	58	54	49	216	62	63	63	62	61	57	51
0.39	1.5	206	59	60	57	58	58	54	49	206	61	62	62	61	60	56	50
0.44	2.0	196	58	59	56	57	57	53	48	196	61	62	62	61	60	56	50
0.49	2.5	186	58	59	56	57	57	53	48	186	60	61	61	60	59	55	49
0.55	3.0	177	57	58	55	56	56	52	47	177	60	61	61	60	59	55	49
0.62	3.5	167	57	58	55	56	56	52	47	167	59	60	60	59	58	54	48
0.70	4.0	157	56	57	54	55	55	51	46	157	58	59	59	58	57	53	47
0.78	4.5	147	55	56	53	54	54	50	45	147	58	59	59	58	57	53	47
0.88	5.0	137	55	56	53	54	54	50	45	137	57	58	58	57	56	52	46
0.99	5.5	128	54	55	52	53	53	49	44	128	56	57	57	56	55	51	45
1.11	6.0	118	53	54	51	52	52	48	43	118	55	56	56	55	54	50	44
1.24	6.5	108	52	53	50	51	51	47	42	108	54	55	55	54	53	49	43
1.39	7.0	98	51	52	49	49	49	45	38	98	53	54	54	53	52	48	42
1.56	7.5	93	51	52	49	49	49	45	38	93	52	53	53	52	51	47	41
1.75	8.0	88	50	51	48	48	48	44	37	88	52	53	53	52	51	47	41
1.97	8.5	83	49	50	47	47	47	43	36	83	51	52	52	51	50	46	40
2.21	9.0	79	49	50	47	47	47	43	36	79	51	52	52	51	50	46	40
2.48	9.5	74	48	49	46	46	46	42	35	74	50	51	51	50	49	45	39
2.78	10.0	69	47	48	45	45	45	41	34	69	49	50	50	49	48	44	38
3.12	10.5	64	47	48	45	45	45	41	34	64	48	49	49	48	47	43	37
3.50	11.0	59	46	47	44	44	44	40	33	59	47	48	48	47	46	42	36
3.93	11.5	54	45	46	43	43	43	39	32	54	46	47	47	46	45	41	35
4.40	12.0	49	44	45	42	42	42	38	31	49	45	46	46	45	44	40	34
4.94	12.5	44	43	44	41	41	41	37	30	44	44	45	45	44	43	39	33
5.54	13.0	39	41	42	39	39	39	35	28	39	43	44	44	43	42	38	32
6.22	13.5	34	40	41	38	38	38	34	27	34	41	42	42	41	40	36	30
6.98	14.0	29	39	40	37	37	37	38	26	29	39	40	40	39	38	34	28
7.83	14.5	25	37	38	35	35	35	31	24	25	37	38	38	37	36	32	26

Figure 3 Opposed-blad damper noise: To find the flow-generated swl from single and double opposed-blade dampers (*a*) Find volume flow rate from left-hand table. (*b*) Add this (subtract if negative) to each octave-band value found from appropriate centre or right-hand table.

Butterfly damper

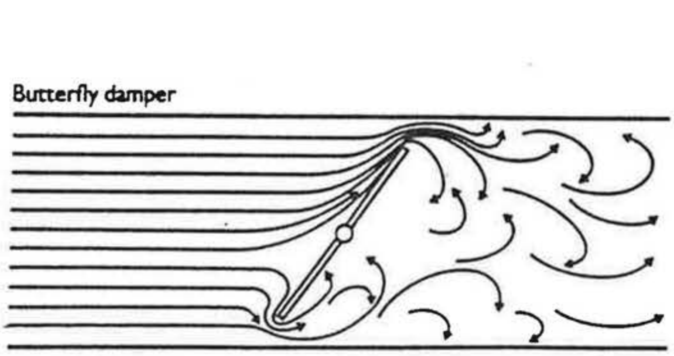

Figure 4 Turbulent flow behind butterfly dampers

10dB reduction in the 125 Hz octave bands and indicates the increased mid-frequency flow noise from the vanes. Generally this compromise is a good solution as the middle frequencies are more easily absorbed and an excellent improvement in pressure loss results.

A detrimental interaction between components of a duct work system is sometimes experienced when splitter attenuators are placed immediately upstream of a bend. The expected average duct velocity is now replaced by that of the much faster air — typically two to three times faster — jetting from the airways onto the turning vanes. Increased flow noise and pressure loss result. This led to the adoption of the composite splitter-bend attenuator incorporating the turning vanes within the airways and hence attenuating the additional flow

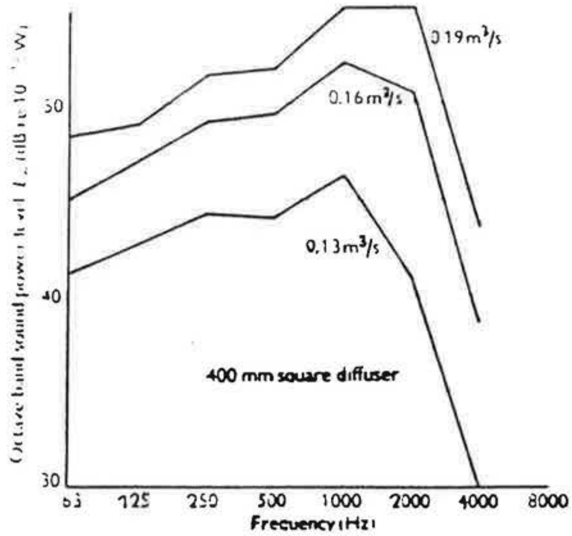

Figure 5 Diffuser sound power level

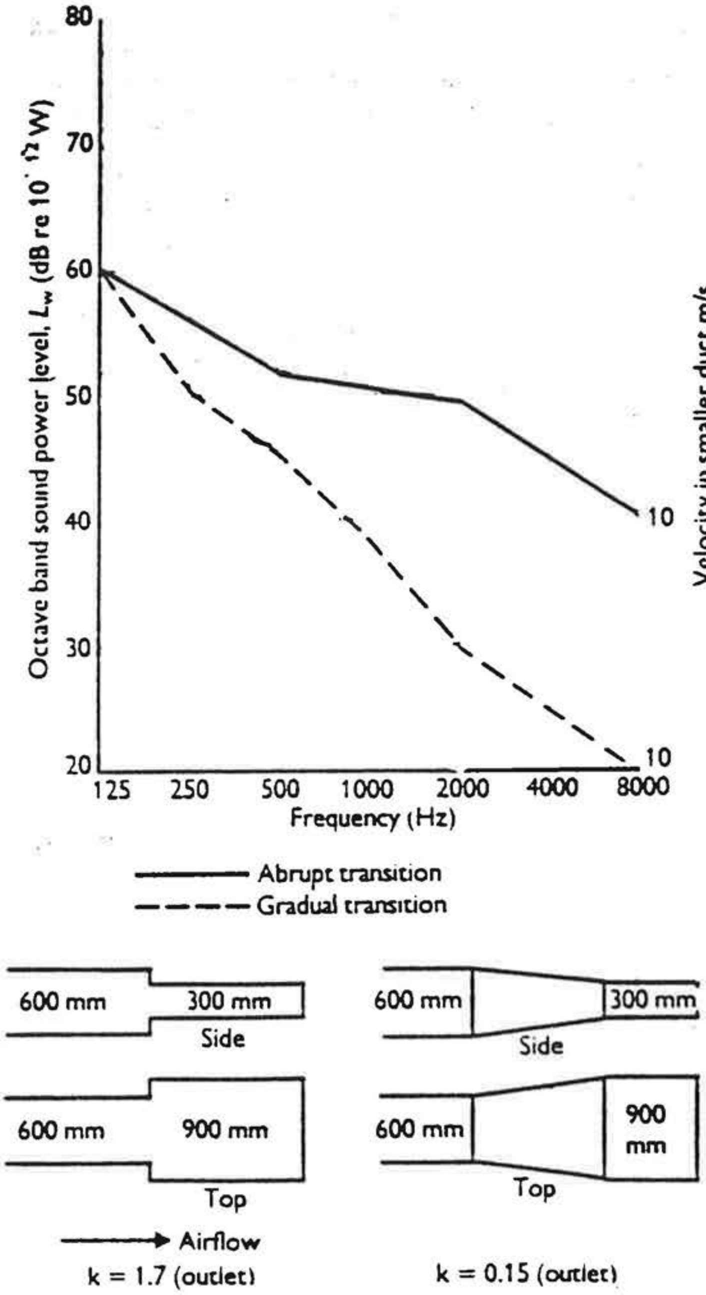

Figure 6 Sound power levels and gradual area transitions from 600 mm × 600 mm to 900 × 300 mm duct cross-section

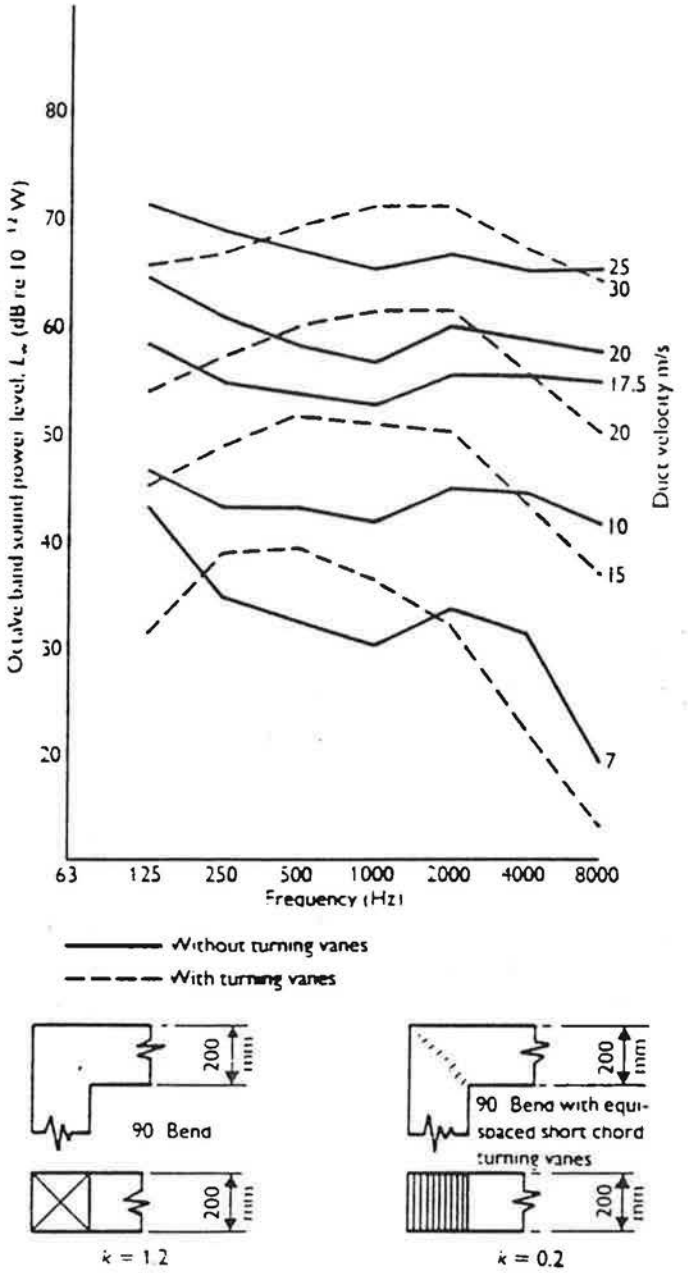

Figure 7 Sound power spectra produced by 90° elbows in a 200 mm × 200 mm duct system with and without short-chord turning vanes

noise. The documented pressure loss, insertion loss and generated flow noise are documented.

Concerning buffeting, Figure 8 illustrates the large 'whirlpools' of turbulence which will be produced by a plain 90° elbow and the corresponding sets of travelling pressure fluctuations which result. Given time and distance, these fluctuations will die away, but should they encounter a second bend, as is often the case, the pressure pulses will excite the end wall strongly and low-frequency break-out or rumbling will result at a frequency depending on the flow rate. Similarly the passage of the wall fluctuations excites the duct walls into the familiar and easily felt vibrations. These phenomena produce audible low-frequency sounds. The frequency of these may be as great as 100 Hz for high-velocity systems, but as the flow rates are reduced the frequencies also reduce, and can be as low as 10–40 Hz, in which range human fatigue rather than audible noise nuisance may be experienced. Hence the noise rating system promoted by ASHRAE includes measurements down into the 16 Hz octave band.

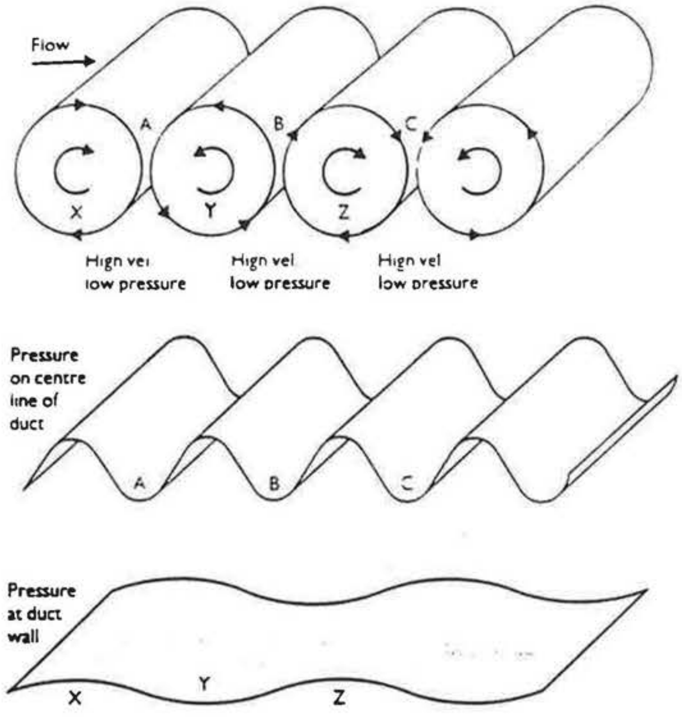

Figure 8 Duct buffeting

3.4 *Take-offs*

Take-offs and junctions represent a very complex problem magnified by the large range of combinations of the parameters involved - main duct to side branch cross-sections, main duct to side branch velocity ratios and absolute main duct velocity. These have been studied by Brockmeyer and reported in outline in the *ASHRAE Guide*. However, for those principally interested in minimising flow noise, Figure 9 indicates the optimising techniques, including the ever-useful one of maintaining a duct velocity such as to keep the airstream under control and hence minimise flow noise. Flow noise is a result of momentum change.

3.5 *Terminal devices — grilles and diffusers*

An awareness of flow noise generation at the final outlet into the conditioned area is essential as, at that stage, it is too late to offer noise attenuation procedures. While many data are available from the hardware suppliers, particular makes and types have not usually been selected at the design stage. Guidance is still however required. Figure 10 supplies such guidance for the wide range of volume flow rates 0.02–8 $m^3 s^{-1}$, related to representative neck areas. The guidance noise level is given as an overall NC (noise criteria) power level to be employed with traditional reverberant and direct-level room corrections.

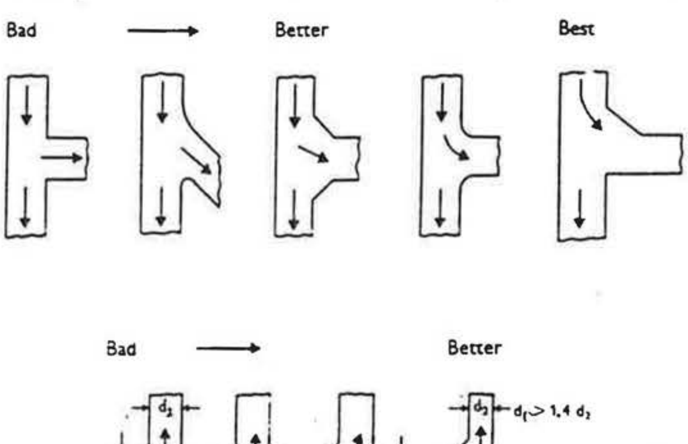

Figure 9 Guidelines for minimising flow-generated noise at branchings and take-offs

The use of NC for guidance does absolve the engineer from any requirement for detailed spectral information and any changes with duty. Figure 11 shows some indicative unweighted frequency information, showing that mid-frequency sound preponderates and that this peak frequency value is not much affected by volume flow rate.

The considerations so far have concerned the air terminal device alone, but in practice, balancing dampers are sometimes incorporated just before these room-side distribution units. The dampers are provided for system balancing and impose an additional pressure drop with some associated flow noise. When these units are close-coupled, as in Figure 12 (courtesy HEVAC), there is a case of 'what is to blame'. As can be seen from the figure, the throttling action of the damper blades produces turbulence which, if left to discharge freely, would produce a certain noise level. However, this turbulence impinges directly on the solid grille blades, where additional fluctuating pressures and forces are produced, and an increased level of radiated noise results. Removal of the grille reduces the noise level, and while it is the source it is not truly the cause. This is often the case in flow noise problems, as can be seen from other examples. If the balancing damper were removed well upstream, the throttling situation could well be acceptable, as the near field turbulence of the damper would have died away with the airstream before encountering the distribution grille. As this is often not practical, particularly when only fine adjustments are required, a prediction program is required linking noise to the additional imposed pressure drop. This is addressed in HEVAC Association publication 2/02/82 *Air Diffusion Guide*, from which some additional broad-brush conclusions can be drawn: *Doubling a discharge velocity increases noise by 18dB. Doubling the pressure drop by damper trimming increases noise by 9dB for a supply grille and by 5dB for an extract grille. Doubling the grille area for an associated doubling of volume flow rate increases noise by (only) 3 dB.*

3.5 *Silencers*

Ironic as it may seem, silencers — here assumed to be of the simple dissipative splitter construction shown in Figure 13 — do not merely attenuate noise but also generate their own flow noise. Further to this, the noise levels at the inlet and outlet are independent and broadly (at 10 dB) similar in magnitude as shown in Figures 14 and 15, but independent of length. It is often thought that the inlet flow noise will be negligible, especially on long attenuators, as it is believed to be the outlet flow noise heard through the attenuator, but reduced by its insertion loss. This is not so as the inlet flow noise can be greater than the outlet flow noise. The inlet flow noise is usually of a 'hissy' nature and reminiscent of grille or turning vane noise, even though the slot and obstacle dimensions are so much larger at around 100–200 mm. This can sometimes be an embarrassing surprise in large studio work when the total extract volume is exited directly through a large in-studio attenuator. This inlet phenomenon also serves as a convenient warning against an understandable but not fully supported belief that flow noise is simply linked to an

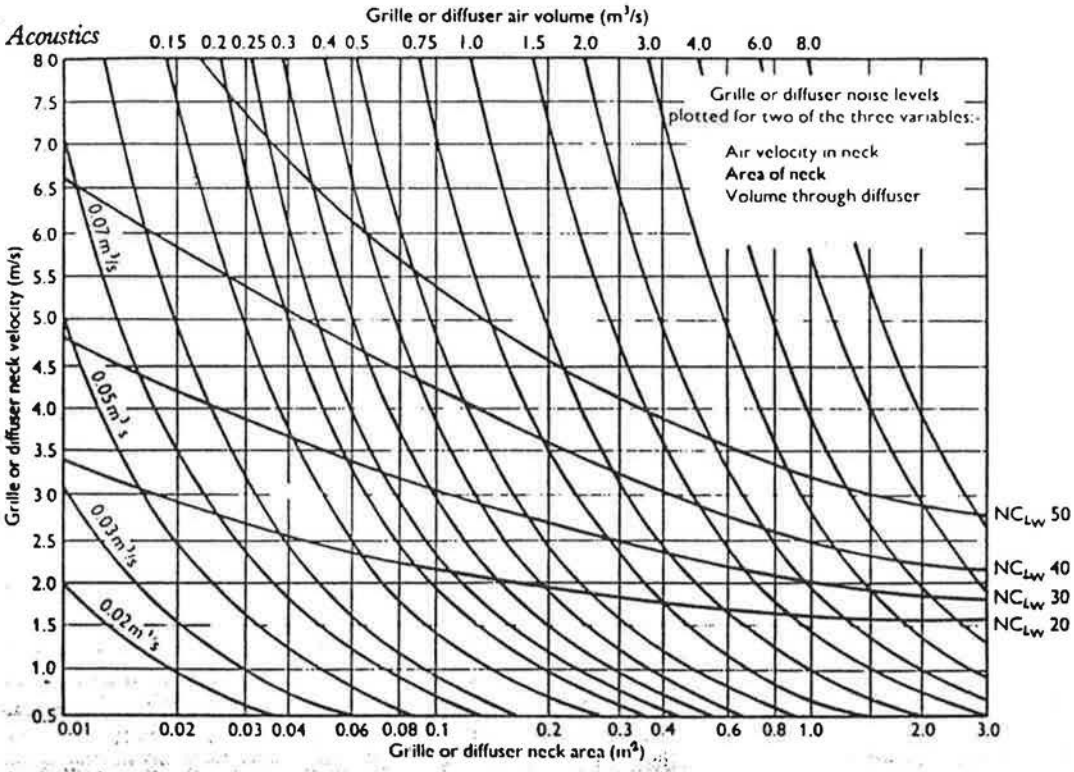

Figure 10 Guidance on flow-generated noise for diffuser/grille

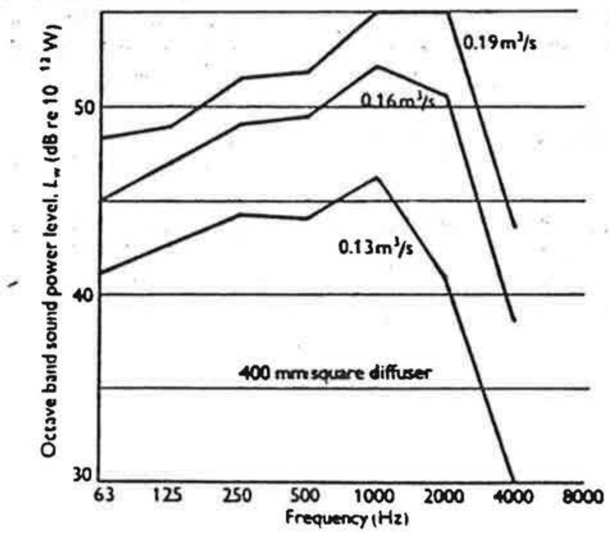

Figure 11 Diffuser sound power level

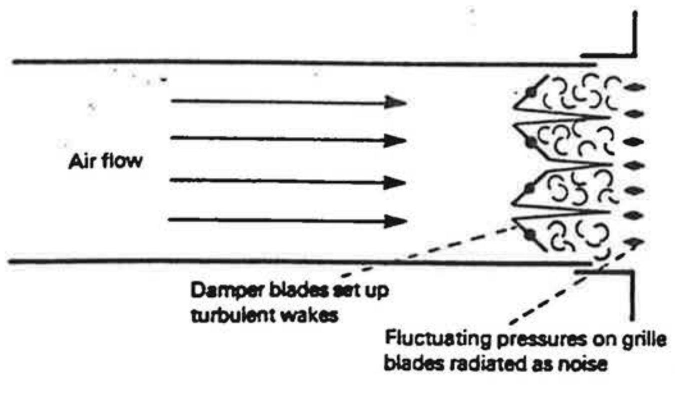

Figure 12 Mechanism of noise generation by dampers and grilles

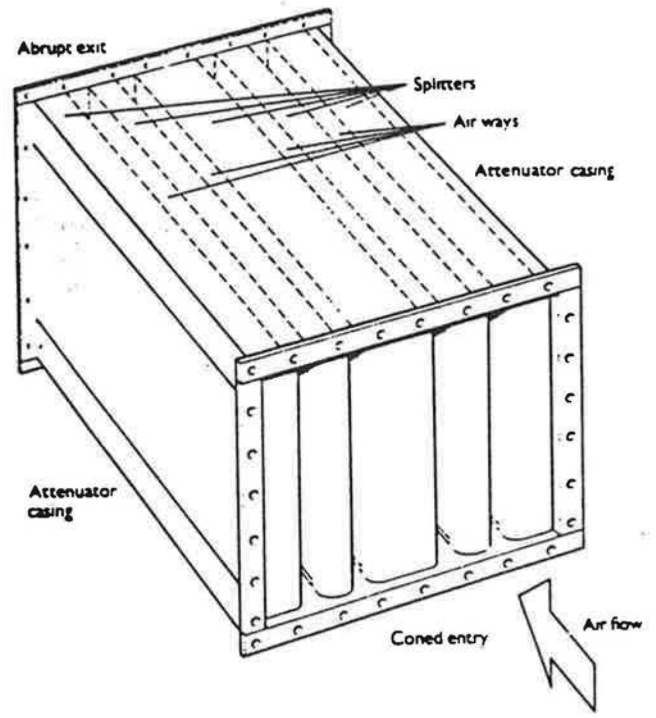

Figure 13 Rectangular attenuator

associated pressure loss. In an attenuator with bullnose inlet splitters, the discharge pressure loss is some ten times greater than the inlet loss, but the flow noise levels are not so dissimilar as would be the case for a throttling damper. This is because the generation mechanism is linked to momentum change, which for such a simple design is the same at inlet and outlet. Tapered outlets and stepped attenuator outlets reduce the discharge momentum change and hence the flow noise.

Concerning the change with flow rate, the graphs support the general law of an 18dB increase with doubling of the flow rate, but there is fine tuning for various frequencies.

4 Conclusion

Flow-generated noise is best minimised by the smooth control of the air flow and minimal step changes in velocity, i.e. a constant-velocity design approach. Where the realities of a system do not allow for this, guidance data for the generated flow noise are available. Above all it should be realised that flow noise is extremely sensitive to velocity, and that 18dB increase per doubling of velocity is a representative 'rule of thumb'.

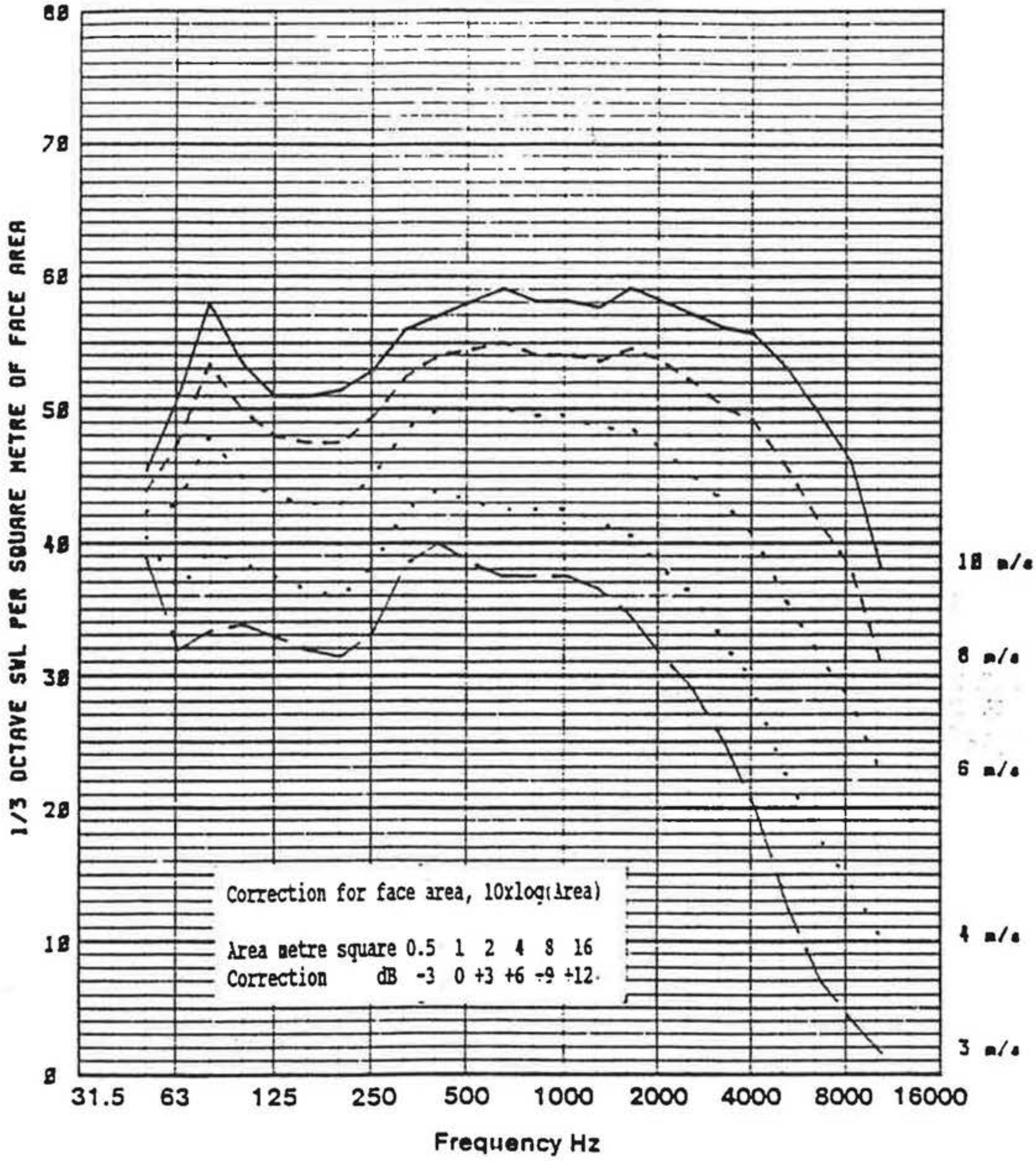

Figure 14 Rectangular attenuator inlet flow noise

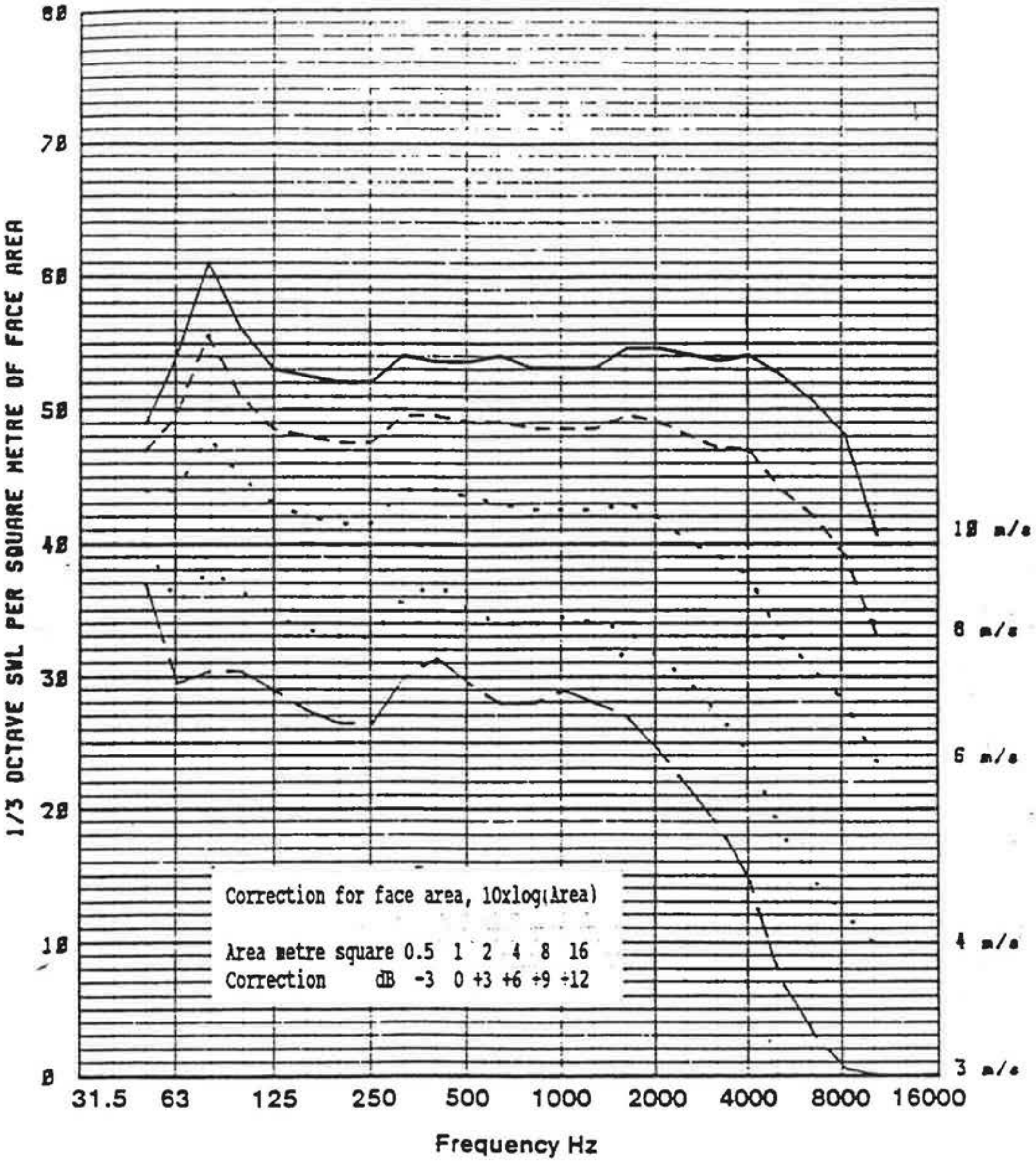

Figure 15 Rectangular attenuator outlet flow noise

Characterisation of sources of structure-borne sound

A Moorhouse (Acoustics Research Unit, School of Architecture and Building Engineering, University of Liverpool)

1 Introduction

Sound levels in occupied spaces in new buildings due to air conditioning, heating and ventilating plant are nowadays calculated as a matter of routine. For example in Figure 16 the sound level in the office due to duct-borne noise (path 1) would be calculated using established techniques: the sound power of the fan is obtained from manufacturers; duct attenuations, obtained from tables, are subtracted, a room correction applied and the calculated sound level compared with the room criterion. In a similar way, with a little more difficulty but still using established techniques, the sound level at the occupant's position due to airborne transmission (path 2) can be calculated.

In the case of structure-borne sound however (path 3), the calculations fall down at the first hurdle. How is the 'noisiness' of the fan as a source of structure-borne noise to be quantified? To date there is no simple or universally agreed answer to this question. The following review explains why characterisation of structure-borne sound sources is in fact a surprisingly difficult problem and discusses current thinking in this area.

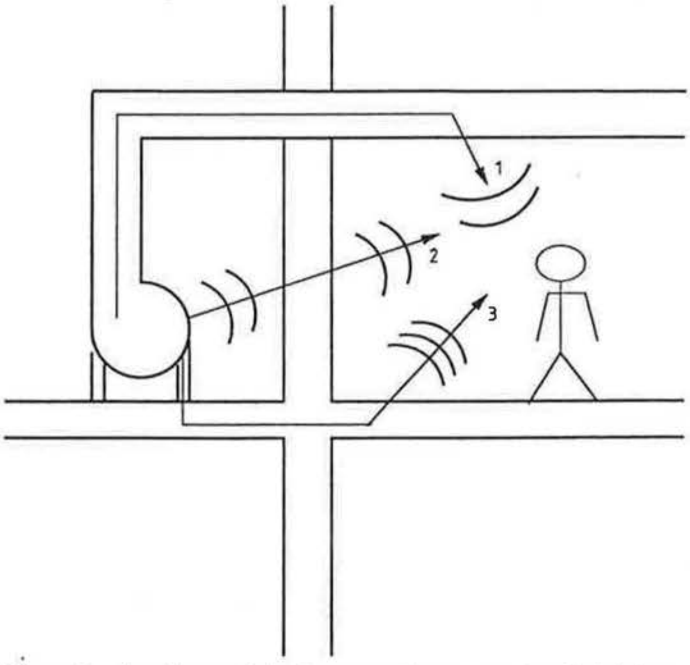

Figure 16 Sound transmission from a machine to a receiver: Path 1 duct-bourne; Path 2 airborne; Path 3 structure-bourne

2 Comparison of airborne and structure-borne sound sources

Most machines act as sources of both airborne and structure-borne sound. Consider Figure 1. The machine delivers sound energy into the air by creating pressure fluctuations (sound waves are nothing more than pressure fluctuations) at the vibrating surfaces. The noisiness of the machine is described simply and accurately by analogy with light and heat sources in terms of the power it emits, normally measured by summing the acoustic energy flow through an enclosing surface. Sound power is a property of the machine so does not vary significantly from one installation to the next; it is conveniently expressed as a single spectrum. As a source descriptor, sound power can be employed on at least two levels; for relative comparison of machines, and for calculation of absolute sound levels when the source is placed in a known acoustic environment.

In the structure-borne case (Fig 1) the energy delivered to the receiving medium, in this case the supporting floor as opposed to the air, will vary from one installation to the next. This is because different supporting structures vary in their receptivity. (Many people have had the experience of typewriters being more noisy on some tables than on others.) Thus the structure-borne noisiness of the installation is a property of both the source and the receiver. There can be then no equivalent of airborne sound power for structure-borne sound sources.

3 Voltage analogy

It is helpful to use a voltage source analogy to describe structure-borne sound sources. Consider Figure 17 which shows the simplest of all electrical circuits. It is well know that the voltage source is fully described not just by its open-circuit voltage V_0 but also by its internal resistance Z_0. These two quantities are independent properties of the source. The power delivered depends on both, and on the load impedance Z_L.

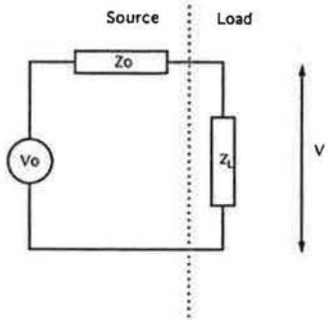

Figure 17 Voltage analogue for noise sources:
Output voltage $V = V_0 Z_L/(Z_L + Z_0)$;
Current $I = V_0/(Z_L + Z_0)$;
Power $P = |V_0|^2 Z_L/|Z_L + Z_0|^2$

For a structure-borne sound source the following analogues may be drawn:

- Open-circuit voltage is analogous to the 'free velocity' of the machine. This is the velocity (vibration) level, measured at its mounting points, when the machine is operating normally but is freely suspended in space. In practice this means that it must be supported in such a way as not to restrain vibration, typically on very soft springs. (It should be remembered that this and all the following quantities are frequency-dependent, so a spectrum is needed for each).
- Internal impedance is analogous to the 'mechanical mobility' of the source structure at the mounting point. Mobility is inversely proportional to the dynamic stiffness; i.e. the stiffer the structure the lower the mechanical mobility and the lower the internal impedance in the equivalent circuit. (More strictly, mobility is the ratio of velocity to applied force at a point.)

— Load resistance is analogous to the mechanical mobility of the supporting structure.

— The current flowing and the voltage across the terminals of the source when loaded are analogous respectively to the force and velocity (vibration) at the point of contact with the supporting floor when the source is installed.

— The power delivered to the load resistance is analogous to the structure-borne sound power delivered to the supporting structure.

The power delivered is the nearest equivalent quantity to airborne sound power and provides the single best measure of the source strength of the installation or its 'noisiness'. However, as we have already seen it depends on both source *and* supporting structure. A machine supplier cannot therefore specify an independent structure-borne source strength for a machine.

The question then arises — can the source be independently characterised? As for airborne sources, a meaningful characterisation would provide a potential purchaser with sufficient information that for a given installation they could (*a*) compare machines for relative 'noisiness' and (*b*) calculate absolute sound levels for comparison with some acceptance criterion. The obvious answer, with reference to the equivalent circuit, is that the source can be characterised independently by the properties analogous to open-circuit voltage and internal resistance. This is a perfectly valid answer and it is generally true that a structure-borne sound source is characterised by its free velocity and mechanical mobility.

Unfortunately, the equivalent circuit is a considerable simplification because in all practical installations the source is mounted at multiple points. Thus every point of contact with the supporting structure acts like a separate voltage source, and the true equivalent circuit consists of many voltage sources all interconnected by coupling impedances. To make matters even more complicated, structure-borne sound power is delivered not just by normal forces at the interface, but also by bending moments which again act as separate voltage sources.

None of these complications arise in the case of airborne sources because the load impedance (analogous to the conductance of the receiving medium, the air) is to all intents and purposes constant, and so, for a given machine, the sound power delivered is invariant.

Thus structure-borne sound sources *can* be rigorously characterised by the free velocity and mechanical mobility, but, except in high-technology applications it is not practical to do this because such a large number of data are involved. The machine supplier would need to obtain free-velocity spectra at each contact point and the full matrix of mechanical mobilities of the source. (For a typical machine mounted at four points, with a force and two perpendicular bending moments at each, this would mean a 12×12 square matrix with 144 elements!). A purchaser armed with such information would still need to obtain the full matrix of mechanical mobilities of the supporting structure and carry out complicated calculations before obtaining a single figure for source strength.

Because of these practical difficulties, research into various simplified methods of characterisation has been continuing over recent years[1], and understanding in this area has improved. However, with any simplification a loss in accuracy is inevitable, and to date all the simplified characterisations are valid only in quite restricted circumstances. Two of the more promising methods are now described.

4 Reception plate methods

A few variations on reception plate methods exist, in which the vibration levels in a standardised supporting structure are measured[2,3]. In a typical test arrangement the test machine is bolted to a standardised test plate and the average velocity of the plate is obtained by measurement with the machine running normally. (In some variants the radiated sound power or sound pressure is measured.) The method thus provides a single-figure rating representative of the 'activity' within the machine, and is increasingly being used to compare machines.

However, reference to the equivalent circuit in Figure 17 will make clear that even such apparently simple comparisons are beset with difficulties. The test described in the previous paragraph is equivalent to measuring the voltage across a standardised, known load impedance Z_L. The source strength is represented by the power delivered, but this power is not directly proportional to the load impedance (as seen in Figure 17). This means that two machines with the same reception plate rating and installed on identical support structures do not necessarily have the same source strength. In fact, strictly speaking, a rank ordering by this method is only valid if the support structure is identical to the standard plate (see Figure 17).

In practice, it has proved possible to obtain acceptable accuracy using such a methodology within a narrow range of machines intended for a narrow range of supporting structures[2] but only after extensive research to establish the validity of the simplifying assumptions. Reliance on such assumptions is a fundamental disadvantage of reception plate methods and it is unlikely that a general rating method along these lines will ever be practical.

5 Mobility methods

Methods using the free velocity and mechanical mobility to characterise the source are known generally as mobility methods. (Most papers published on this subject are rather technical, but Reference 4 is useful background.) A significant advantage of such methods over reception plate methods is that they have a sound theoretical basis. The importance of free velocity is now widely recognised to the extent that measurement standards exist[5,6]. (In fact free velocity has long been used to grade the quality of balance in rotating machines.) Although the free velocity is an incomplete characterisation it does provide a rank ordering of sources in terms of 'noisiness' under certain conditions. Equation 1 shows that the free velocity (open circuit voltage) is proportional to the power, provided Z_0 is much less than Z_L, a condition which is likely to be met for small machines on concrete floors but not for machines rigidly mounted on supporting structures of construction similar to that of the machine itself.

In predicting absolute source strength the difficulties associated with mobility methods are practical ones, due to the large number of data required. However, in the special but common case of resiliently mounted machines it has been shown that reasonably accurate prediction is possible and simple enough to be practical[7]. The structure-borne sound power is given approximately by:

$$P = (1/2)\,N\,|v_{fs}|^2\,|Z|^2\mathrm{Re}(Y)$$

in which N is the number of contact points, $|v_{fs}|^2$ the mean square free velocity of the source (above the resilient mounts), Z the mechanical impedance of the resilient mounts (which is approximated at low frequencies by the stiffness divided by the angular frequency), and Y the mobility of the supporting floor. Again, certain assumptions are necessary which cannot be described in detail here but are given in Reference 7.

6 Case study

A brief case study will illustrate some of the benefits of quantifying the structure-borne source strength. A typical fan installation as shown in Figure 18 was studied. The fan was mounted on steel spring mounts, which provided reasonable vibration isolation, but of course some sound power is still transmitted through the springs into the floor. The power injected into the floor via the spring mounts was calculated by equation 1, and the spectrum obtained is shown in Figure 18 (lower curve). It is interesting to note that this is quite 'spiky', because most vibrational energy is generated at the rotational speed of the fan and motor and at its harmonics.

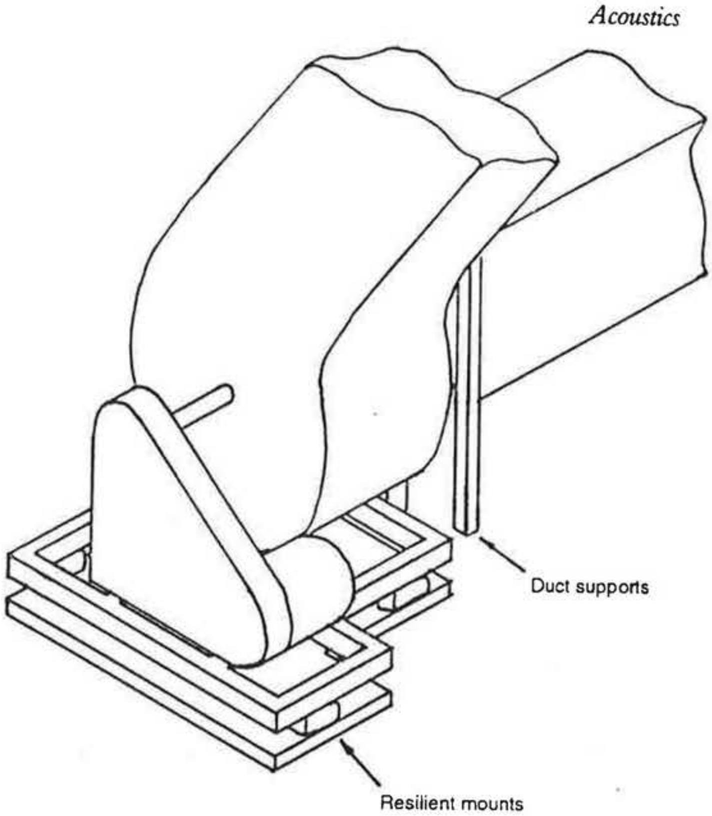

Figure 18 Fan installation used in case study

Once the power passing through the mounts had been quantified it became obvious that the vibration levels in the floor could not be due to this transmission path alone. It was suspected that a ductwork support bracket (Figure 18) was also acting as a sound transmission path, so the power flow down the bracket and into the floor was measured in situ. (Details of the measurement will not be described here.) In Figure 19, it is seen that the power passing into the floor via the bracket dominated that through the spring mounts; this confirmed the subjective impression. Furthermore, Figure 19 provided a basis for quantifying the benefit of modifications such as replacing the duct support with a spring hanger.

This case study also illustrates some of the previous points. The structure-borne source strength of the installation is the combined power through the mounts and the support bracket, i.e. the decibel sum of the two curves in Figure 19. Clearly, the source strength would be completely different were the same machine to be installed with the duct support replaced by a resilient hanger. This illustrates clearly that although the machine is independently *characterised* by its free velocity, its structure-borne *source strength* depends on the installation.

The transmission paths can be compared in this way because they are both quantified in the same units, those of power. Note that these same units are also used for airborne sound

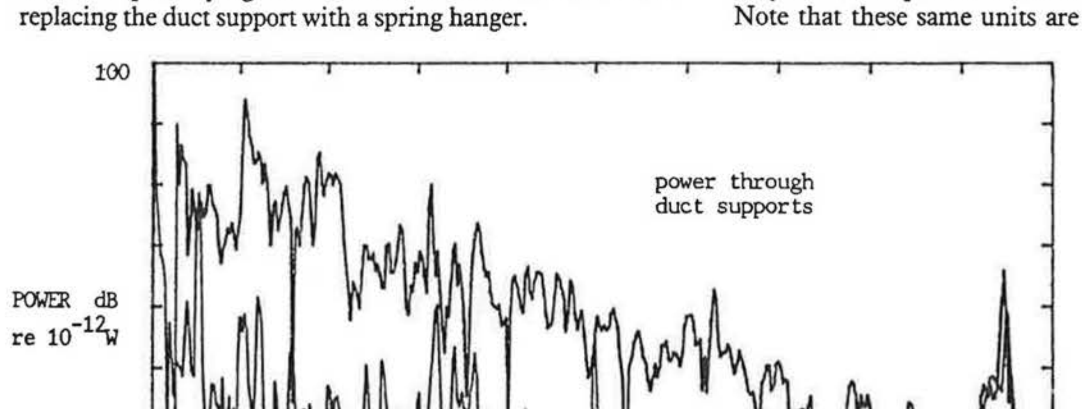

Figure 19 Comparison of power flow through mounts and through duct supports

power (dB_{re} 10–12 W), so in theory one can also compare the structure-borne source strength of the machine with its air-borne source strength (although such a comparison is not always meaningful because in the latter case the propagating medium is the structure, and in the former the air).

7 Summary and conclusions

The structure-borne 'noisiness' of an installation depends both on the machine *and* on the supporting structure. Therefore, a machine supplier cannot specify an independent source strength for a machine. However, one can, in theory, independently *characterise* a source such that a purchaser has sufficient information to calculate its source strength when installed on a given supporting structure. In practice, full characterisation is seldom practicable because a large number of data are required, and to date, no universally accepted simplified characterisation exists.

Reception plate methods, in which the machine is mounted on a standard support plate, are increasingly used to quantify the 'activity' within the source, but the results are valid only under certain conditions. The most scientific of the simplified characterisations is the free velocity, for which measurement standards exist. Free velocity data are a subset of those required for complete characterisation, but allow useful estimates of source strength for resiliently mounted machines. Currently, understanding of this complex problem is insufficient for source descriptors to be standardised reliably, and so in the short term, experience of similar installations will remain an invaluable means of assessment. However, it will increasingly be possible to back such experience up by calculation, bringing about a general improvement in understanding, hopefully leading to standard methods in the future.

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Acoustic modelling techniques in building services

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1 Introduction

This article is mainly concerned with numerical modelling which can be used in design analysis and research on heating, ventilating and air-conditioning (HVAC) systems, although the methods can be applied to many other situations in mechanical, civil and marine engineering and architecture. An overview of the available numerical techniques (without extensive mathematics!) is followed by a review of their different applications and selected examples.

In the design and installation of HVAC and related services, we may be concerned with various aspects of acoustic behaviour, all of which relate to sound which is almost always unwanted, i.e. noise. Noise emanates from equipment itself (fans, other mechanical plant, ducts), is transmitted through the system (down ducts, out of grilles), and is finally distributed in occupied spaces (rooms, atria, theatres, ...). We may also be concerned with structure-borne noise paths, in which noise is radiated from vibrating structures which may be excited directly by fluctuating forces transmitted through the structure (e.g. from equipment mounts) or excited indirectly by acoustic pressures acting on one part, which then re-radiates noise elsewhere (e.g. duct noise break-out).

These issues have all been addressed in the past by combinations of experience, the gradual development of products and isolation techniques, practical noise control treatments, and by empirical or analytical calculations simple enough to be done 'by hand'. With the rapid development of computers in recent years, and some developments in software and numerical methods, it is now possible to do much more predictive and diagnostic work using computer models. This can supplant parts of the 'cut and try' approach and enable practical work to begin closer to an optimum solution.

2 Available numerical methods

The numerical techniques can be divided into two general classes: those which are *deterministic* (i.e. they are based on a numerical model of the physical phenomena of acoustic waves, possibly also including structural vibration waves) and those which are *energy-based* (i.e. using a model of the acoustic and/or structural system within which the input acoustic and vibrational energy is distributed according to the geometry and the energy transmitting or dissipating properties of the components and their connections). In the deterministic class, finite-element and boundary element methods are the most effective. In the class of energy-based approaches, there are various geometrical acoustics techniques such as ray tracing, and statistical system modelling approaches such as statistical energy analysis.

3 Deterministic methods

3.1 Finite-element models

Finite-element analysis (FEA) of structures has been used for some time and is now a mature science, including its use for modelling vibrations. Structural FE models may be used in conjunction with acoustic FE or boundary element models to generate surface vibrations as the sources of radiated noise, or in fully coupled analyses of vibro-acoustic interaction.

FE acoustics is essentially an adaptation of the FE technique in which the elements represent small divisions of the fluid medium (e.g. air) forming the acoustic field. The elements are, of course, finite —, so the model has a limited extent and a finite, closed boundary. Different boundary conditions can be applied (known vibration, pressure release representing an open face, admittance representing an absorbent coating, etc.) and each element can have different properties (some may represent absorption within their volume, as in foam or wool packings). The modes of the closed volume can be calculated, as well as the response to known vibrations or acoustic sources. The results are mode frequencies and shapes, and acoustic pressures, particle velocities and intensities (acoustic energy flux vectors) at all the nodes of the FE mesh, at each calculated frequency.

3.2 Boundary element methods

Boundary element (BE) acoustic models employ a mesh of elements representing only the surface of the object(s) considered. The fluid (air) is considered as linking all these surface elements together, and mathematical relationships are set up between them by the computer program on the basis of the equations for the acoustic waves in the fluid (Helmholtz's wave equation) and the particular geometry. Effectively a 3D volume problem is condensed mathematically to a 2D surface problem. it is therefore possible to consider not only a finite, closed region, but also an infinite, open region, with acoustic radiation to infinity automatically included in the latter case.

There are two forms of the BE method used in acoustics: the collocation and variational approaches. In the former, the fluid is considered to be on one side of the boundary element only, so the model must have a closed surface and the behaviour either inside or outside is calculated. In the variational approach, however, fluid is considered on both sides of the element simultaneously: this allows an easy analysis of open structures (such as a duct with an open end) and of coupled structural-acoustic behaviour (such as breakout through flexible duct walls).

4 Energy/geometric methods

4.1 Ray tracing

Acoustic waves can be considered as rays or beams spreading out from each source. They are reflected (as in a mirror) at each surface they strike, losing energy each time according to the absorption characteristics of the surface (the Sabine coefficient). The sound pressure level at any point is then the summation of the energy contributed by all the rays or beams as they pass the point. The model consists of a geometric representation of the surfaces, as planes with defined absorptions (usually varying between octave bands) and point sources with defined powers (and directional characteristics if required; line and plane sources can be represented by arrays of points). Usually no phase or other wave-interaction effects are taken into account, although standard formulae based on path-length differences can be used to account for simple diffraction over screens and similar barriers. Open exterior and closed interior regions can be considered, as can mixed situations. The lack of phase-related data means that standing-wave mode effects cannot be considered. (However, by using the path length of each beam, a development of the method allows constructive or destructive interference between reflections to be considered).

4.2 Statistical energy analysis

This approach is not regularly used by the author's company or by other industrial contacts and will not be considered in detail, although it is an area of continued research. The approach employs a model of the structural and acoustic systems made by assembling components or subsystems. These are characterised by their response to vibrational or acoustic energy, and its transmission and dissipation. The interfaces between the components are similarly characterised. The model then allows prediction of the distribution of energy between the components due to defined inputs, and the consequent vibration and sound pressure levels. The method is sometimes criticised for a lack of direct, physical characteristics (either as vibro-acoustic waves or geometric shapes of components) and uncertainties over input data such as the coupling between components and the related loss factors (damping). Its main application to date has been in structural dynamics rather than in acoustics.

5 Features, limitations and applicability

The principal limitation of FE and BE methods is the size of the numerical model which can be created (i.e. the size of the equation system) and the consequent computer processing time and memory required for calculating and storing results. Even with current computers, this can be significant. The objects being modelled are divided into discrete elements, and it is the size of the elements (the refinement of the mesh) which has a large influence on the overall accuracy. A rule-of-thumb requires six elements per acoustic wavelength for good results, hence an upper limit on the valid frequency range can be set for any particular mesh. A finer mesh, with more but smaller elements, would cover a higher frequency, but requires more computer resources. It is usually impractical to model rooms, other than fairly small ones, with FE or BE, so these methods are generally confined to models of more localised effects, such as radiation from vibrating equipment, silencers/mufflers, duct details and component design.

The disadvantages of FE compared to BE are the time taken to make the mesh (a 3D volume mesh is much more difficult to generate than a 'surface' BE mesh, with elements like thin shells) and the inability of FE to model free-field radiation effects. However, FE can model varying fluid properties and volume absorbers, and transfer impedance layers between one region and another (e.g. representing perforated panels). The BE methods offer easier, faster mesh generation, free-field boundary conditions and (with the variational approach) the ability to handle 'free edges' (sharp edges projecting into the field, such as thin ribs or a duct opening).

Geometric approaches such as ray or beam tracing are relevant at the higher frequencies, where FE and BE are impractical, and can be used (with caution) at lower frequencies. This results in an overlap of the applicable frequency ranges, for which both approaches can be used. Since the ray paths and their relative contributions to the noise level at any point can be plotted, the method is very useful in room acoustics, whereby different treatments can be applied to different surfaces to optimise the design. Where phase-dependent effects dominate at lower frequencies (e.g. modes in small rooms)

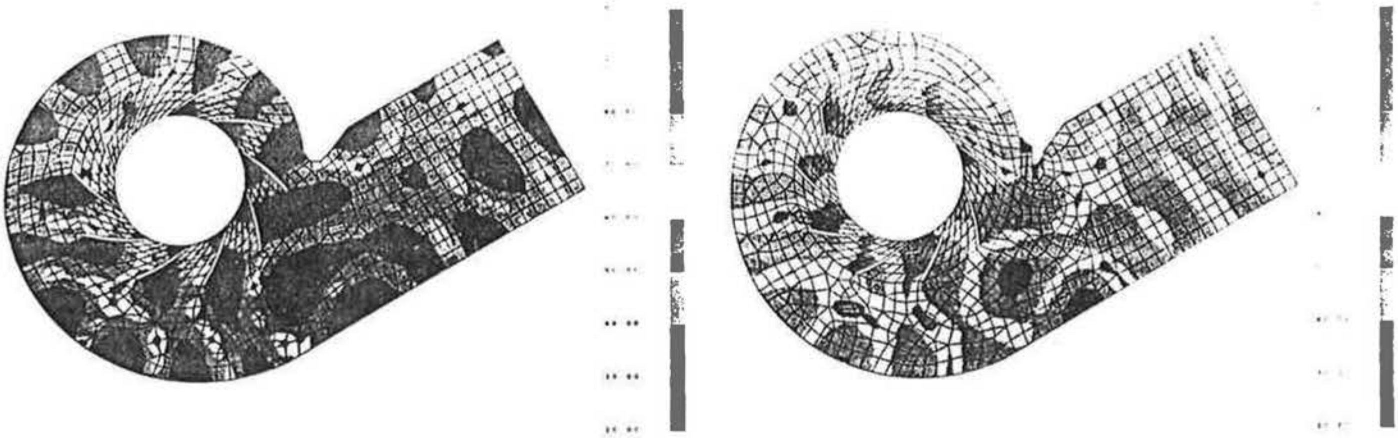

Figure 20 Sound pressure levels at 167 Hz inside a centrifugal fan

however, the deterministic approaches of FE or BE can be preferred.

4 Applications and examples

In the following, some typical examples of applications are shown, with remarks on their main features and other related applications.

4.1 Air-handling plant: Fan noise

Figure 20 shows the sound pressure levels inside a centrifugal fan, from which noise was passing down the outlet duct. The problem was essentially tonal (as might be expected, related to blade-passing frequency). A finite-element acoustics model was used to assess the effect of a tuned absorber ('Helmholtz chamber') attached to the fan scroll at a suitable location. The problem was eliminated by this device.

4.2 Duct design: Plenum/ silencer design

Silencers can be modelled by finite-element and boundary element methods. FE models have the advantage of modelling absorptive as well as reactive types. Figure 21 shows a model of a simple plenum silencer (based on an example given in Reference 8). By computing the pressures and acoustic particle velocities at the inlet and outlet for the opposite cases of closed (zero velocity) and open (zero pressure) outlets, and unit inlet velocity, the transmission loss through the silencer can be found at each frequency. A curve of transmission loss can then be drawn: Figures 22 and 23 compare the results for the plenum of Figure 21, from Reference 8 (where it appears that negative TLs have been ignored) and from the SYSNOISE program[9].

4.3 Duct design: Outlet grille behaviour

The transmission of noise from an air-supply system into the room may be of interest. The effects of the geometry and other features of the grill can be analysed using boundary elements. An example is shown in Figures 24 and 25, showing the directional effects (at one particular frequency) of the grille construction. Diffraction and phase-dependent interactions are taken into account as well as reflections.

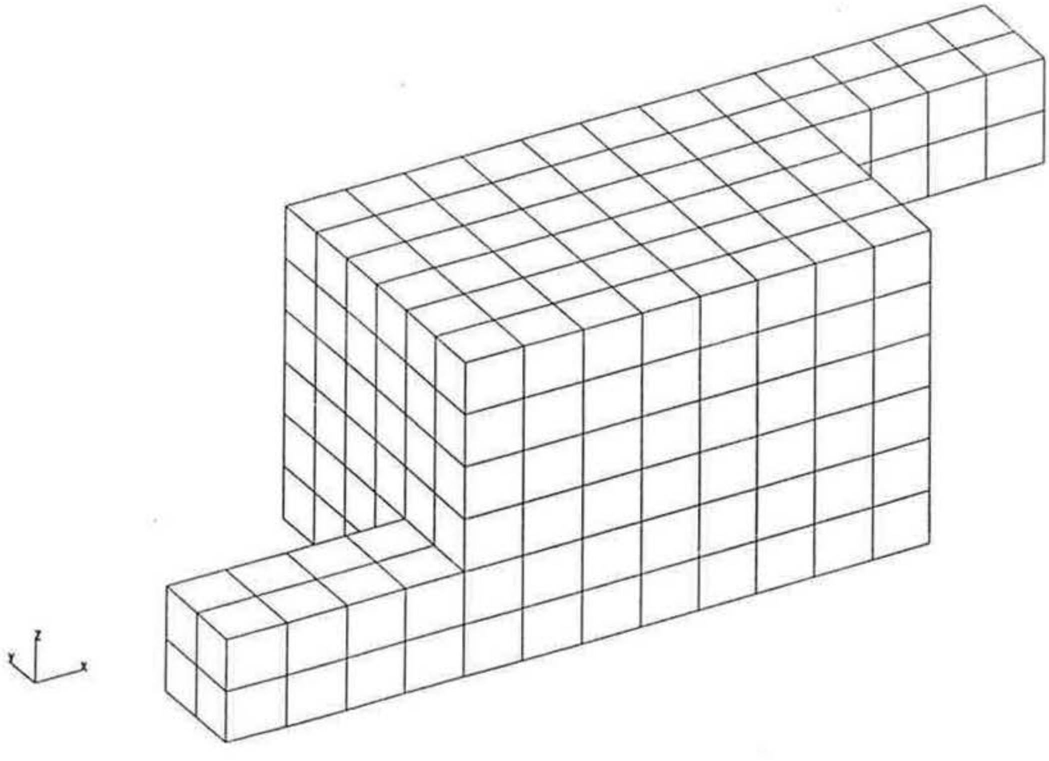

Figure 21 Model of simple plenum silencer

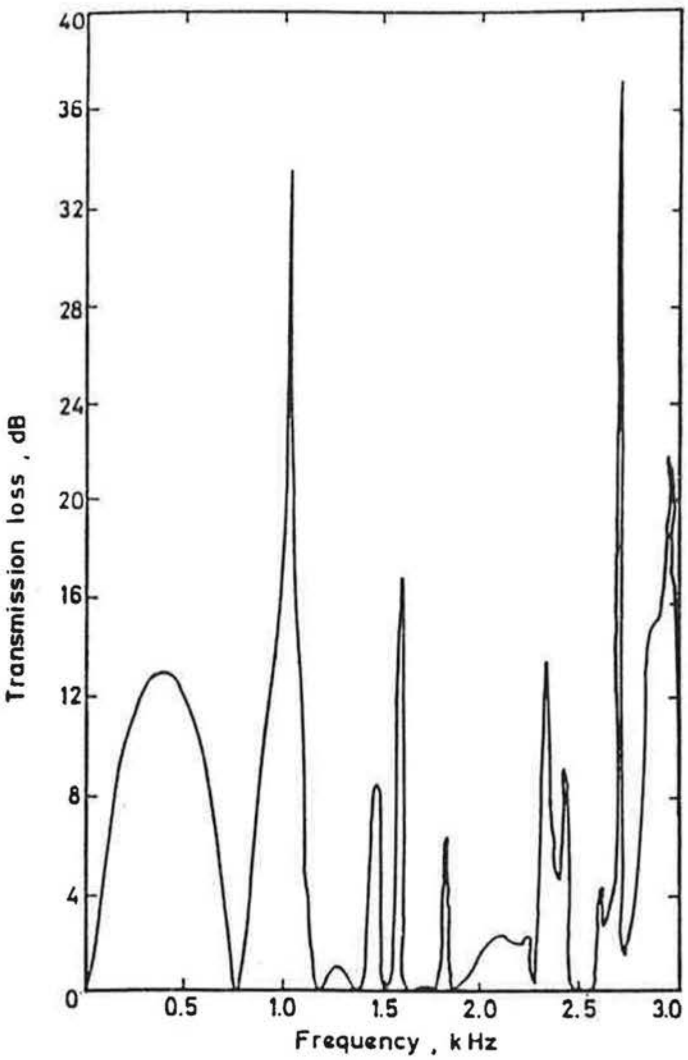

Figure 22 Results for plenum of Figure 21, from Reference 8

4.4 Equipment noise radiation

The noise radiated by vibrating machinery such as pumps, fans, gearboxes etc. can be assessed using boundary elements. Structural vibration data are used as an input to the calculation (derived from measurements on an existing machine or from structural vibration calculations, for example using finite elements). The total radiated sound power can be calculated, as well as the acoustic field in free-field or partially-enclosed situations. Figure 26 shows a directivity diagram of sound pressures around a gearbox, superimposed on the BE mesh of the component. The contribution from the vibration of a selected surface to the sound power or the SPL at a selected point can be determined, to identify those parts where treatment will be most effective.

4.5 Room acoustics: Modes

A further application of FE acoustics is the calculation of modes of enclosed cavities, such as small rooms. Figure 27 illustrates such a case.

4.6 Room acoustics: Sound distribution in auditorium

The case shown in Figure 27 is a mode at a specific frequency. For larger spaces, the distribution of sound energy is better determined by a geometric approach. This is shown in Figure 28, where the trace of a sound beam around an auditorium is shown. The sound source may be directional, so — for example — the dispersion of noise from a ventilation grille can be assessed. Such models can be of virtually unlimited size, number of surfaces, sources, etc. [10]

4.7 Room acoustics: Equipment spaces

The beam-tracing approach can also be used to determine noise levels for safety purposes in equipment spaces. As shown in Figure 29, each item of plant is specified in terms of sound power in each octave band (with special directivities if desired) and the resultant SPLs due to direct and reverberant energy can be found. Contour maps show those areas where noise levels may be unacceptable. Recalculation with changed absorbent properties for certain surfaces, added absorbent panels or baffles, etc., allows remedial measures to be proposed and verified before any commitments to particular hardware. The calculation technique gives a much more precise assessment of the noise distribution than simplified formulae (Sabine, Eyring, etc.). The latter do not take the geometric distribution of sources and absorbers properly into account, and can therefore suggest a need for more treatments, added absorption, etc., than is really necessary.

5 Conclusion

Both deterministic (FE, BE) and energy/geometrical (ray-beam tracing) approaches to acoustic modelling can be used to study the behaviour of building services plant and installations. The design of individual components can be optimised without recourse to time-consuming trials and modifications on prototypes. The increasing power of computers and enhancements to software and calculation techniques will undoubtedly make these methods more common in research, development, product design and installation in future.

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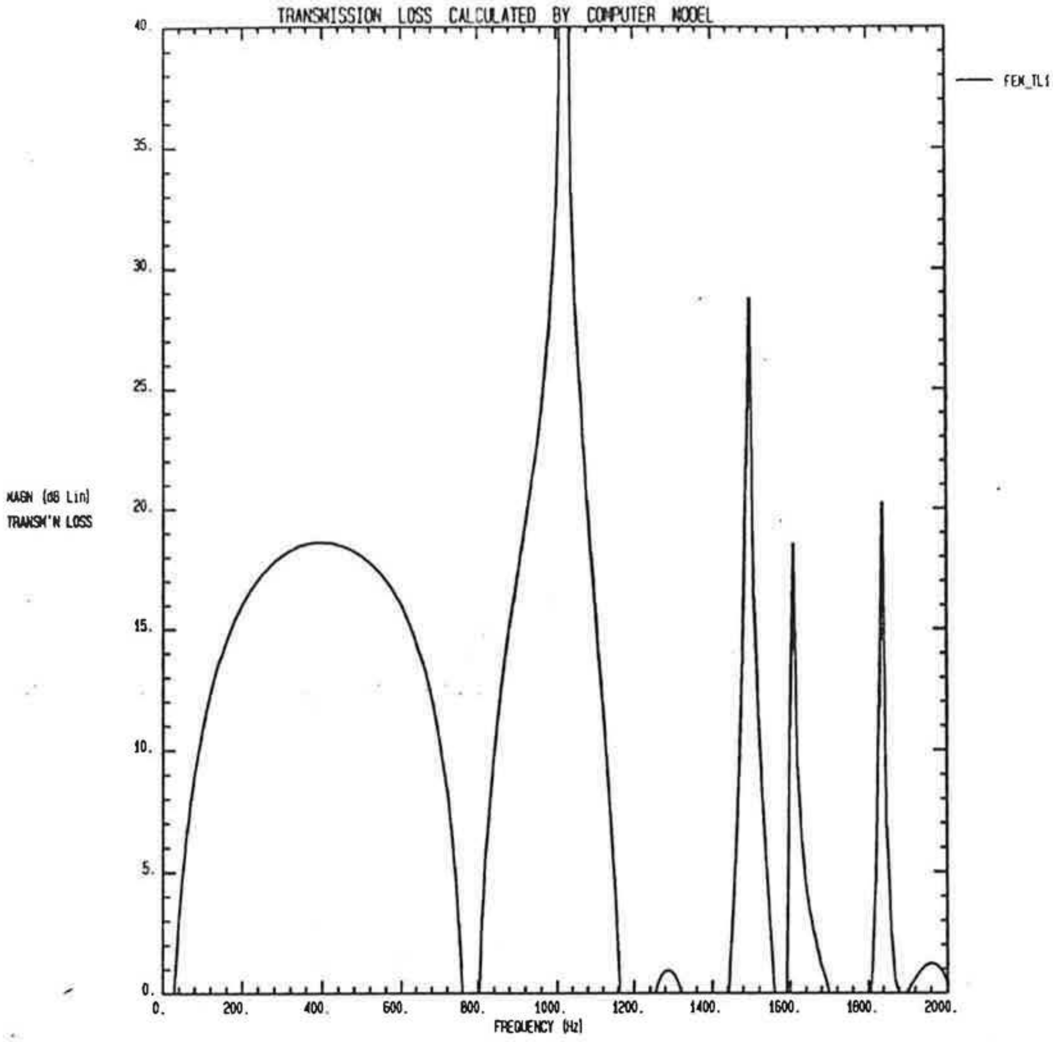

Figure 23 Results for plenum of Figure 21 from SYSNOISE program[9]

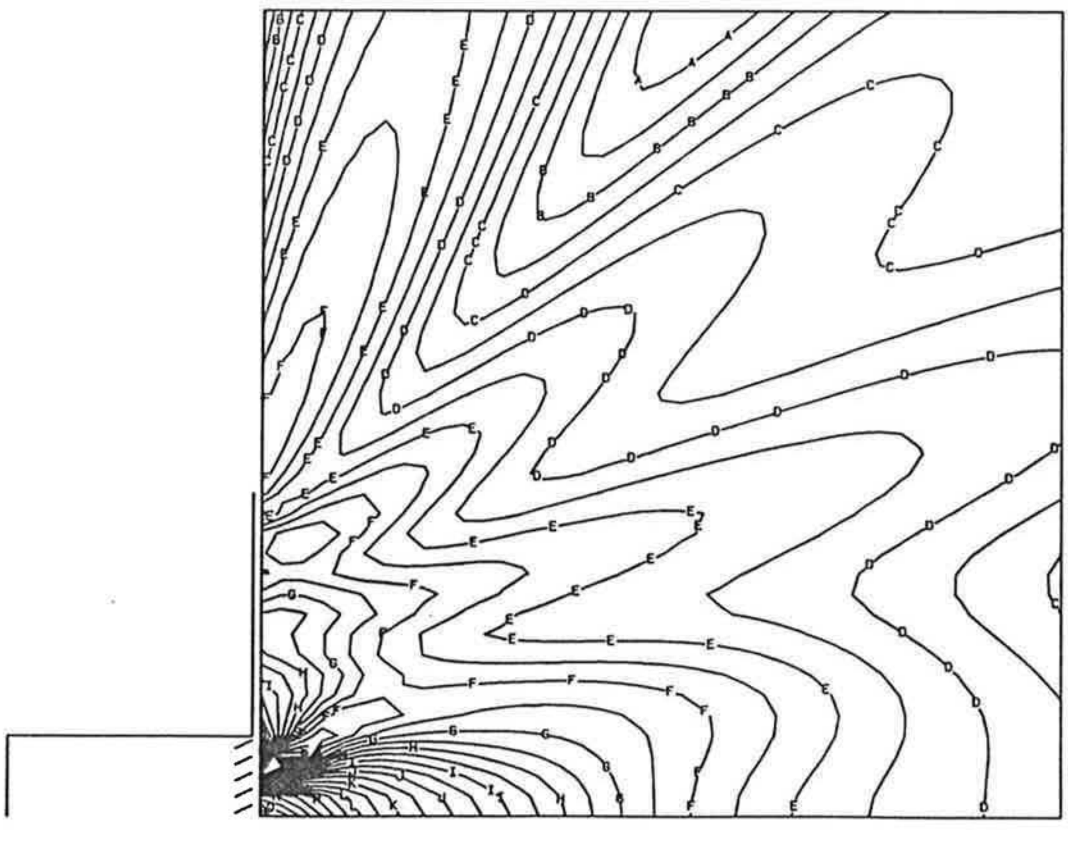

Figure 24 Sound pressure around outlet grille — 2 kHz

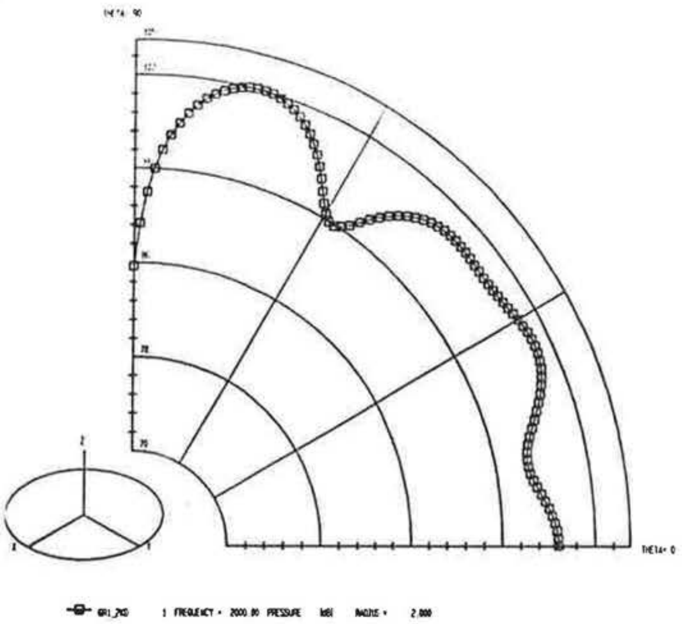

Figure 25 Directional effects of outlet grille construction — 2kHz

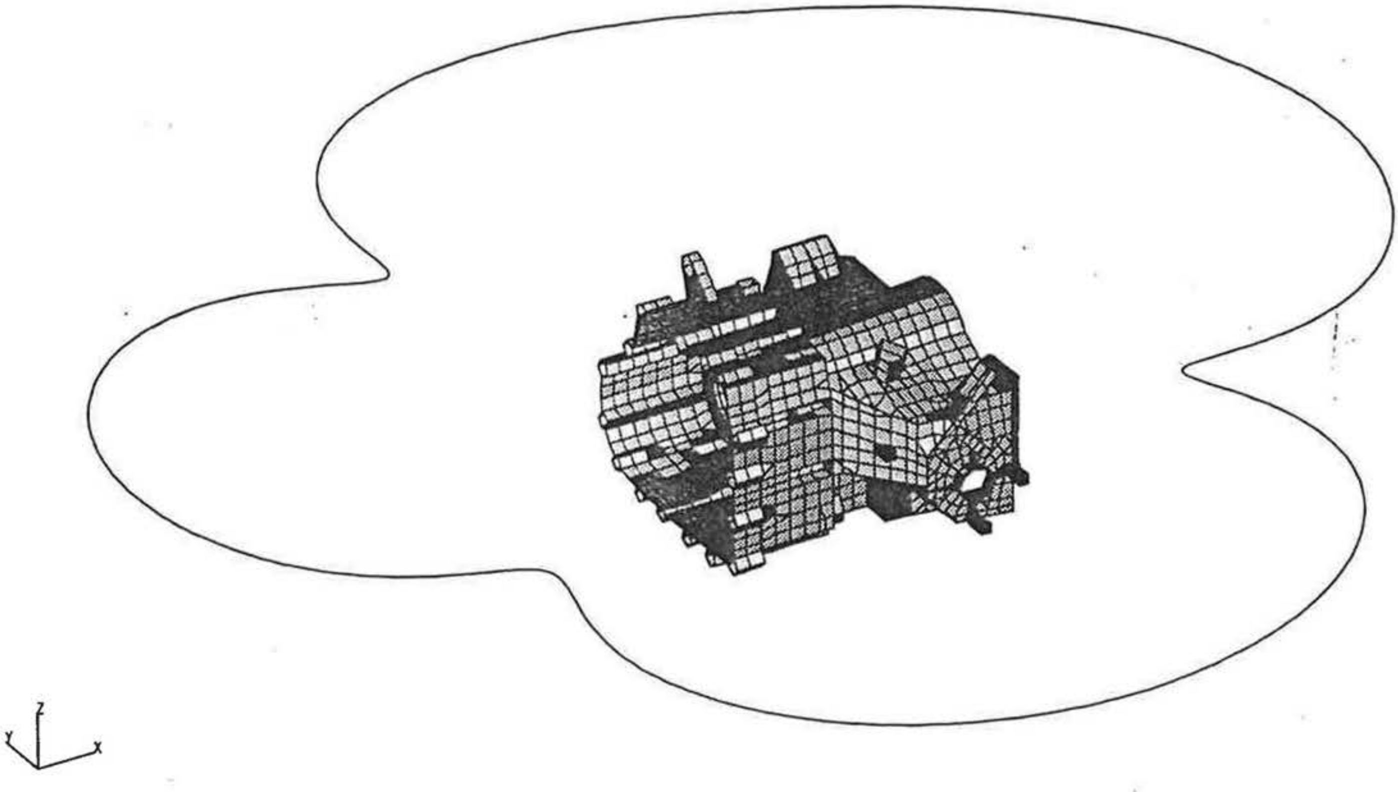

Figure 26 Directivity diagram of sound pressures around a gearbox, superimposed on a BE mesh of the component

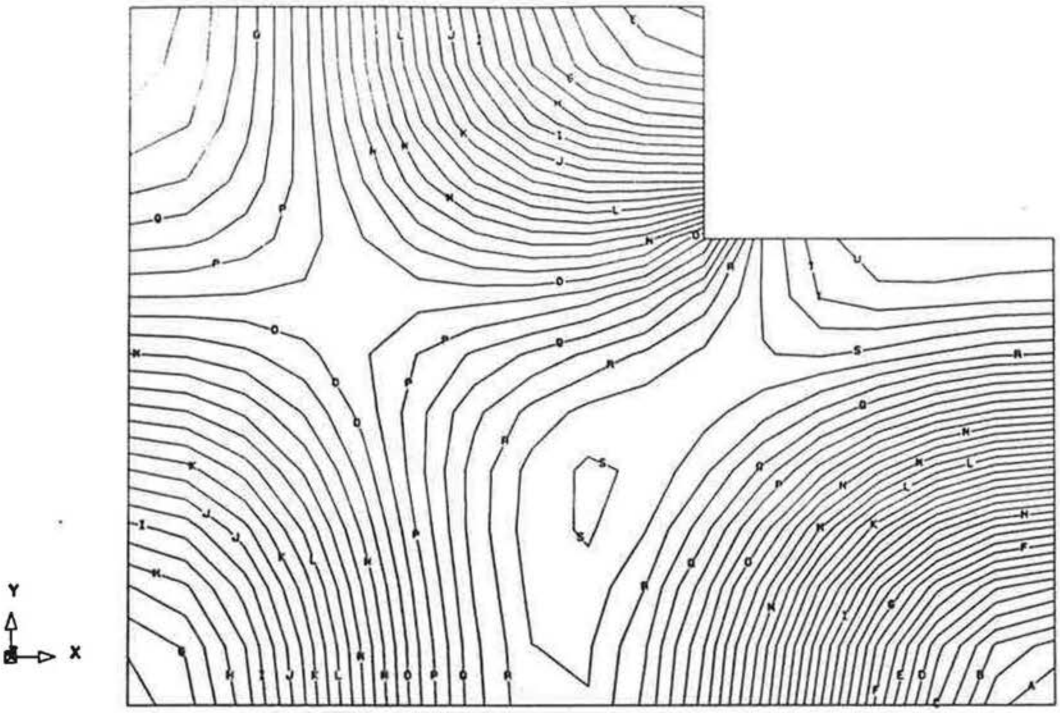

Figure 27 Acoustic finite-element analysis: Mode in a small room (101 Hz)

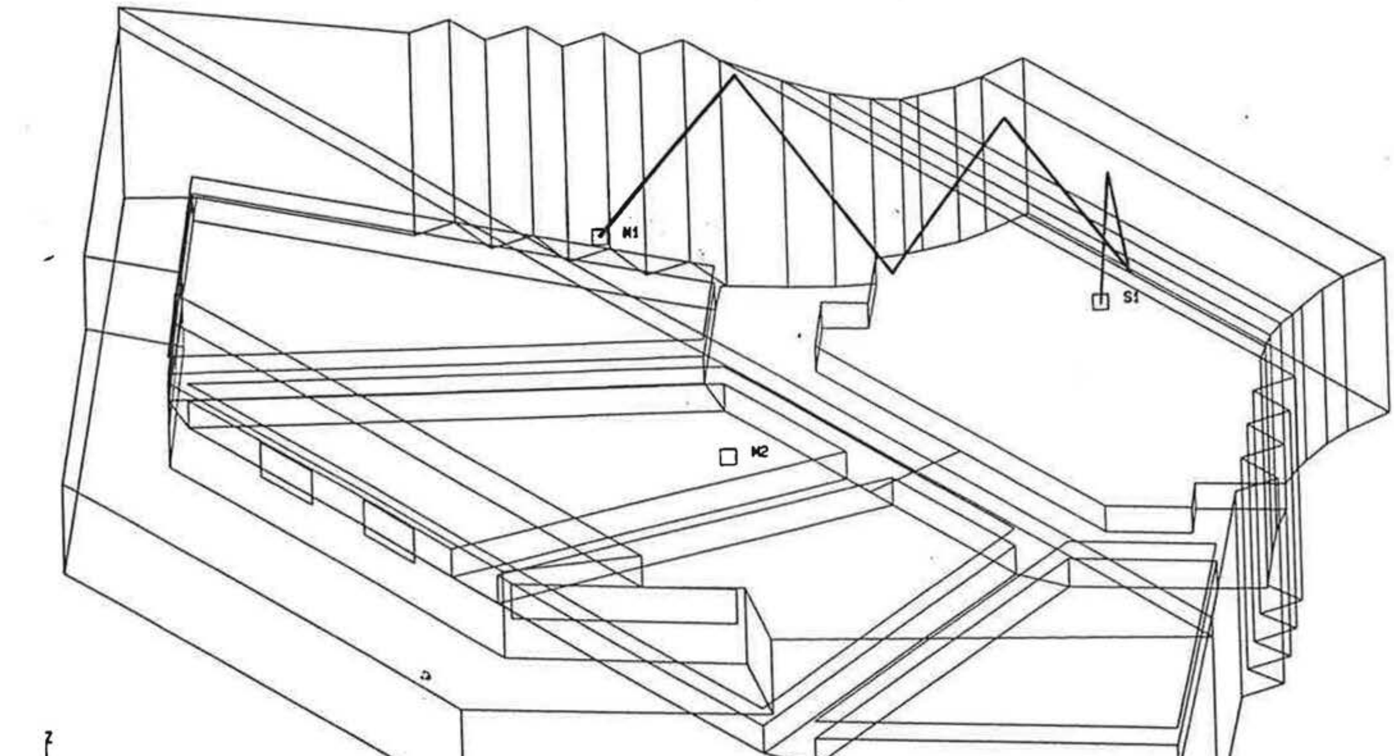

Figure 28 Auditorium with corrugated side walls

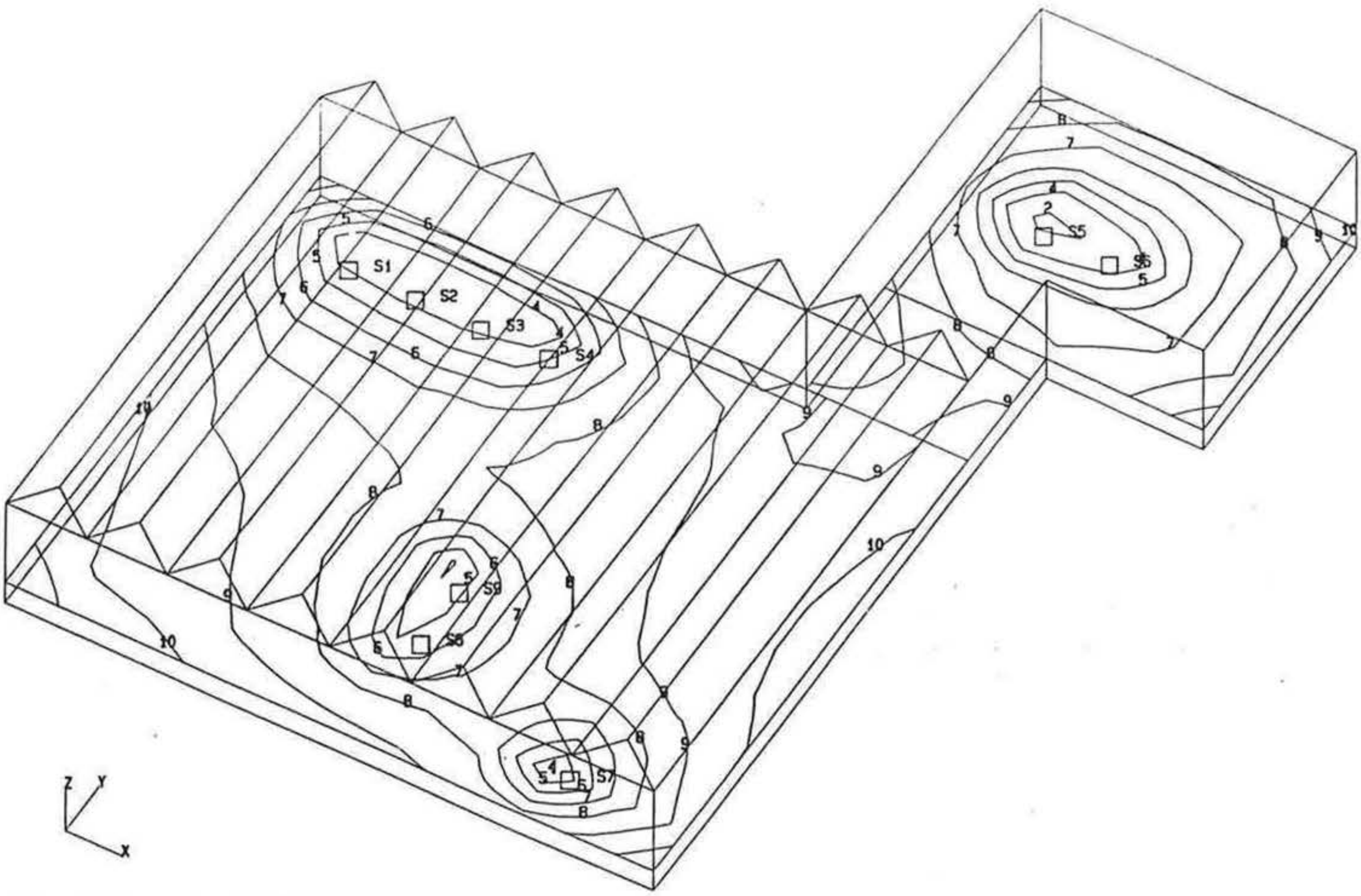

Figure 29 Noise contours for plant room (See Table 2 for key)

Table 2 Key to Figure 29

Contour no.	1	2	3	4	5	6	7	8	9	10	11
dBA	85	84	83	82	81	80	79	78	77	76	75

Active attenuation of noise in HVAC systems

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1 Introduction to active noise control

The basic concept of active noise control is to create an 'anti-noise' acoustic field in a space in order to cancel the existing noise and produce a quieter space. The best results obtained so far are for cancellation of one-dimensional, plane waves travelling down a duct. As shown in Figure 30, the upstream microphone detects the noise and converts the sound pressure waves generated by the fan to an equivalent electrical signal. Using adaptive filtering methods,the controller creates an electronic representation, or model, of the duct. The cancellation signal is sent to the loudspeaker, which radiates the 180° out-of-phase sound wave to mix with, and cancel, the noise propagating downstream from the loudspeaker. The second microphone, also downstream from the loudspeaker, monitors the residual acoustic pressure after cancellation and returns information to the controller to enable it to adjust itself for optimum results.

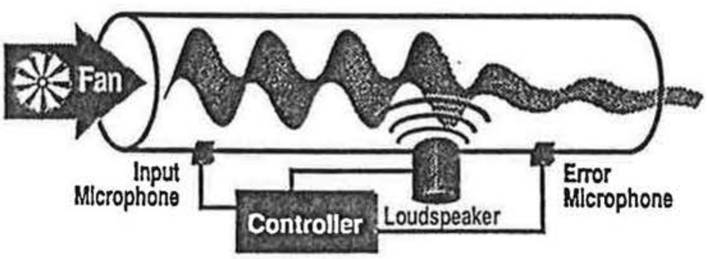

Figure 30 Active noise control in a duct

The loudspeaker radiation not only propagates downstream, where it cancels the noise, but upstream, where it reaches the input microphone and contaminates the noise signal. One approach to this problem is the use of an infinite impulse response (IIR) adaptive filter to compensate for the acoustic feedback which occurs between the cancellation loudspeaker and the input microphone[11]. The IIR adaptive filter creates an electronic representation, or numerical model, of the duct acoustical systems. Once the delay and amplitude changes from the input microphone to the loudspeaker have been found, any pressure wave which reaches the input microphone will be attenuated when it arrives at the loudspeaker. This technique cancels both periodic (tonal) and random plane waves, with preferential attenuation of tones.

The controller also assumes that any pressure fluctuation appearing at the input microphone represents a plane wave which will appear, after a delay, at the cancellation loudspeaker. The coherence between the two microphone signals is a convenient measure of the amount of plane wave energy compared with local pressure fluctuations, e.g. turbulence. Coherence of 0.95 or greater usually ensures good broadband cancellation in excess of 10 dB.

Another critical feature for a successful active noise control system is the ability continuously to update the electronic model of the duct acoustics. This is accomplished by continuously playing low-level pseudo-random noise through the cancellation loudspeaker. On-line calibration uses this known signal to allow the controller to determine the characteristics of the loudspeakers and microphones and those of the environment, such as duct temperature, which affect the speed of sound[12]. This information is included in the transfer function which the controller applies to the input microphone signal so that it can create the correct cancelling sound wave.

A final important requirement for cancellation of broadband (random) noise is that the acoustic delay from input microphone to cancellation loudspeaker be at least as long as the electrical delay from the input microphone to the radiated cancelling noise of the loudspeaker. Periodic signals (tones) are a special case where such causality is not required and shorter systems are possible.

In Figure 31, the duct is the acoustical domain. The source microphone converts the sound pressure to an electrical signal. The speaker converts the electrical output of the controller into the cancelling sound. A computer model is made of the error plant, which is the physical system of cancellation speaker, duct, and error microphone. The model is therefore a transfer function of the speaker, duct, and microphone response. Similarly, a model is made of the forward plant from source microphone to loudspeaker and a third model is of the acoustic feedback, from loudspeaker to source microphone. These three models characterise the whole active system and are held in the controller, where they are updated continuously in order to allow for changes in the duct system.

2 Heating, ventilating and air conditioning applications

The use of duct silencers is the conventional means of attenuating low frequencies, as shown in Figure 32. Passive (absorptive) silencers use various types of fibrous packing materials

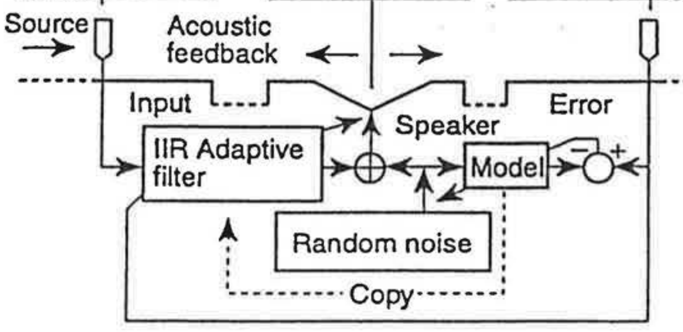

Figure 31 Active noise control schematic and block diagram

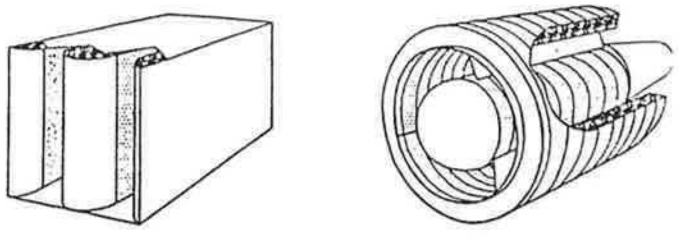

Figure 32 Conventional passive silencers

so that the noise energy is dissipated. Passive silencers normally have a rectangular or circular cross section and impose a pressure loss on the system from, say, 100 to 625 Pa depending on silencer design and air flow conditions[13].

3 Active/passive (hybrid) designs

Figure 33 shows an active/passive hybrid approach, designed to work in HVAC systems and which can provide attenuation over the full eight octave-band range[14]. The unit is typically sized from 2.4 to 3.0 m long, 0.6 m × 0.6 m to 0.6 m × 2.0 m cross section and requires only 25 mm of sound absorptive lining in order to control the high frequencies. Low frequencies are cancelled actively. This gives full-range noise control in a zero-pressure-loss system, thus saving the fan power which would have been required to move air against the silencer resistance. Both input and error microphones are designed to reject flow noise. Heavy duct wall material (18g steel) reduces break-in and break-out effects, allowing the unit to operate in both noisy and quiet external environments. In some cases, rectangular ducts may need to be reinforced upstream of the cancellation loudspeaker to further reduce break-in and break-out of low-frequency sound. Active silencers can also be fitted to round ducts which have superior break-in, break-out characteristics at low frequencies.

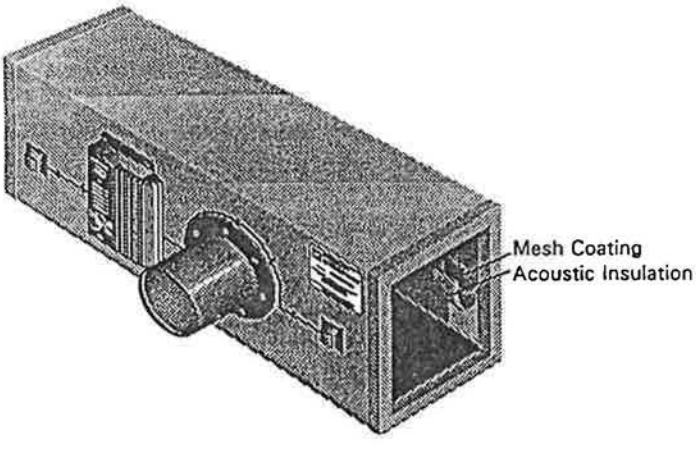

Figure 33 Active/passive hybrid duct silencer

Figure 34 shows a narrow-band frequency spectrum and octave band analysis of the fan noise typically generated by forward-curved centrifugal fans. The dark curve is the uncancelled low frequency fan noise, and the lighter curve is the residual noise after active cancellation. The measurements were taken in the duct. Typically, the active part of the silencer works in the first three to four octave bands and the passive lining attenuates noise in the third to eighth bands. Only a modest amount of mid- to high-frequency noise attenuation is normally required and this is achieved with the absorptive lining around the inside perimeter of the duct in the region of the active system.

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MESSAGE:

Dear Mark,

Thank you for your phone call earlier today. The publications which I am interested in are:

- Reynolds and Bledsoe (1991), and Schaffer (#10227,1993) "Sound and Vibration Control"
- Schaffer (#10227,1993) "A practical Guide to noise & vibration control for HVAC Systems"
- Fry (#10222,1995) also discusses the noise generated by air flow within ductwork.

Incidentally the ducting which I am concerned with is a fabric permeable material (7m long)

I can't locate any references/publications on attenuation or absorption for these fabric ducts.

I would be deeply appreciated if you are aware of the locations of such material.

Thank you for your valuable time.

Dan O'Brien..................

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