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THE PREDICTION AND CONTROL OF NOISE FROM HVAC SYSTEMS

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Methods for predicting noise from air flow systems in buildings are reviewed and compared. The accuracy of the methods is considered but insufficient information is available to produce an overall estimate of uncertainty in the final result. A questionnare survey of users indicates confidence in the prediction techniques when supplemented by the users own practical experience. There is a widespread feeling that current prediction methods 'play safe' by overestimating noise levels and that there is a need for further research to reduce uncertainty and to save on unnecessary noise control. Noise breakout from ducts is identified as an increasing problem and methods of prediction and control are reviewed, and some test data presented.

Introduction

If the noise level from the HVAC system in a building is too high it can lead to annoyance and complaint. A certain minimum noise level is, however, useful in masking other unwanted sounds, such as speech from a neighbouring office, and if building services noise level is too low expensive improvements to the sound insulation of the building may be required. The prediction of noise levels allows control features to reduce excessive noise to be built into the system at the design stage, much more conveniently and economically than retrospective remedial measures. Uncertainties in the prediction can lead to cautious overdesign of noise control, which may lead to the system design being uncompetitive at the tender stage, and which is anyway wasteful in terms of financial and enegy costs. Alternatively uncertainties in prediction can result in inadequate noise specifications and to contractual disputes about responsibilty if noise levels are too high.

The prediction of noise levels

A number of noise prediction schemes are available to the designer (1-6) in the form of graphs, charts, formulae, and software packages. They all follow a similar format, based on the source - path - receiver model of noise transmission. Starting with the sound power level produced by the fan a series of attenuations are subtracted to allow for the effect of the various parts of the duct system, in order to arrive at the sound power level entering the ventilated space, via a grille or diffuser. From this the sound pressure level at the reception point in the room may be found, and compared with the noise target level required by the client, and, if necessary, adjustments may be made to the system design. In many cases the end product of the prediction is the specification of the insertion loss required from any silencers or other noise control measures built into the system.

The prediction scheme also takes into account the effect of additional sound energy generated by airflow through the system, for example at bends, dampers, grilles and diffusers, and other sound transmission paths such as noise breakout from the ductwork or from the plant room.

Since all aspects of noise generation, transmission and control are frequency dependent the prediction has to be carried out separately for a range of octave frequency bands. Fan noise tends to be most important at low frequencies, with air flow noise predominating at medium and high frequencies. For well known theoretical reasons the accuracy of noise measurements and predictions is usually lower at the low frequencies and it is also more difficult to achieve noise reductions at these bands than for higher frequencies. The noise level target for the system is usually expressed as either an NC, NR or dB(A) value, all of which depend on the frequency content as well as the level of the noise.

The various prediction schemes all contain guidelines for good practice to minimise noise, including advice on the selection, installation and operation of fans, the layout of ductwork in order to maintain smooth airflow and minimise turbulence. The penalty for ignoring these rules of good practice is often an increase of several decibels in excess of predicted noise levels.

Henson (7) has compared a number of prediction schemes, focussing mainly on the CIBSE and ASHRAE methods, but also including the Australian IRAH method (17). The various corrections to fan sound power level may, following Henson, be collected into four groups: the duct system attenuation, additional sound power generation due to air flow through the system, the estimation of room sound pressure level, and the calculation of noise transmission (i.e. break-out) through duct walls.

Even when a duct system does not include any specialist noise reduction elements, such as in-duct silencers or acoustic duct linings, only a proportion of the acoustic energy generated by the fan or by the air movement across duct fittings reaches the ventilated room. This natural attenuation provided by the duct system arises from a combination of the division of sound energy at branches, the reflection of sound energy at bends and at duct terminations, and sound energy losses due to duct wall vibration. When the system contains long lengths of ductwork the attenuation due to duct wall vibration may become significant, particularly for rectangular ducts at low frequencies. The CIBSE guide gives values for the attenuation of unlined ducts (in dB/m) for various sizes of duct, expressed as a mean duct dimension, and for three frequency bands of 125Hz, 250Hz and 500Hz and above. The ASHRAE guide uses a different method, based on the ratio of duct section perimeter to cross sectional area, and frequency bands of 63Hz, 125Hz and 250Hz and above. However, the two methods give very similar results (0.3 dB/m) for a wide range of duct sizes, from 300-900mm mean duct diameter, but there are differences at low frequencies for larger ducts and at the higher frequencies for smaller ducts. The two methods also give very similar predictions, within 1dB in most situations, for the attenuation of unlined rectangular bands. The ASHRAE method, reflecting American practice, also gives information about the attenuation provided by ducts and bends which are lined with sound absorbing material. The attenuation of sound at a junction in the ductwork is based on the division of sound power between the feed and take-off ducts, on the basis of the ratio of their cross-sectional areas. This calculation is identical for both the ASHRAE and CIBSE methods, but the ASHRAE method also suggests making an additional attenuation allowance where the junction incorporates a bend. Fairhall (8) has confirmed the need for such a correction. Both methods also give very similar results, usually within 1 or 2 dB, for the correction to be applied for reflection at the end of a duct, but the ASHRAE method gives more guidance about conditions under which this correction should be applied. The full end correction should not be applied unless the duct termination is preceeded by a straight section of ductwork of at least 3 to 5 diameters long, and is without any diffuser or grille. It should also not be applied where linear diffusers are tapped directly into plenums, or for diffusers tapped directly into primary ductwork. A reduction of at least 6dB in the end reflection loss is recommended if the duct terminates in a diffuser.
The CIBSE method also gives guidance about the effect of grilles or diffusers on end reflections corrections. The Guide advises that where a bend immediately precedes a grille or diffuser the end reflection loss may be reduced and recommends halving the usual figure in such cases, but is not specific about minimum distances between the duct termination and the bend or junction. These considerations may have important financial implications, since end-reflection is a major contributor to system attenuation at low frequencies, which would be otherwise difficult and expensive to provide using acoustic silencers. Fairhall (9) has also investigated end reflections. Henson (7) concludes, with regard to system attenuation, that in comparison with ASHRAE the CIBSE method probably underestimates the attenuation due to branches, but may sometimes overestimate that due to end reflections.

The methods for predicting airflow noise are more detailed in the ASHRAE method but they require more comprehensive input data such as the pressure drop across elements, whereas the CIBSE guide gives simpler methods based only on flow velocity. Oldham (10) has argued for noise prediction based on pressure drop information.

An important difference between methods is in the calculation of room sound pressure level. The CIBSE method uses the standard Sabine room acoustic theory based on the concepts of direct and reverberant sound fields, whereas the ASHRAE guide adopts alternative sound propagation models of Schultz and Thompson. For 'live' rooms, such as plantrooms or bathrooms the two methods give similar results, but Henson has shown that for typical office conditions the CIBSE method predicts higher levels, by up to 5dB in some frequency bands. The ASHRAE Guide claims that the Schultz method is accurate to $\pm$ 2dB within the quoted range of room size and acoustic absorption.

The ASHRAE system gives a more detailed treatment of duct break-out, (including break-in and for flat-oval ducts), acoustic duct lagging and the breakout of plant room noise and radiation to the external environment. Duct breakout is dicussed in more detail later in this paper. Reynolds and Bledsoe, (11), have reviewed the experimental and theoretical basis for the various prediction algorithms used in the ASHRAE scheme, indicating where they think that the evidence is limited and requires further research. A number of such areas are identified, including: noise generation by diffusers; system attenuation from unlined rectangular ducts and bends, and from plenum chambers; end reflection losses, particularly for high aspect ratio rectangular ducts, and ducts of short length and fitted with diffusers; breakout from ducts under airflow conditions (current methods being based on loudspeaker tests); insertion losses of externally lagged ducts; and the extension of room sound pressure level prediction to lower frequencies (63Hz). Henson, (7) also suggests the need for further research in similar areas to those listed by Reynolds and Bledsoe, but also including the need to extend the work of Schultz, on room sound pressure levels to a wider range of rooms and noise sources, and on the interaction between the various parts of the duct system and the effect on noise attenuation and generation. Neither Reynolds and Bledsoe nor Henson give estimates of error, or uncertainty for the various components of the prediction schemes, but Henson has suggested, on the basis of his own experience and that of colleagues, that, provided the duct system is fairly extensive then existing prediction methods allow the design of HVAC systems which are usually free from noise problems. This is because, it is suggested, the prediction of system attenuation is usually rather conservative, so that predicted sound pressure levels tend to be overestimates, thus, in effect, incorporating a factor of safety in the design. This view is strongly supported by the user questionnaire survey reported later. What is not known is exactly how conservative the prediction methods are, and which parts of the system are responsible.

The accuracy of noise level prediction

Several factors may affect the accuracy of noise predictions. Parts of the prediction process may be based on simplistic theoretical acoustic models which may not always relate to actual conditions. Other parts of the method may be based on laboratory test data which may not relate to site conditions. The author's own experience has shown that, even under laboratory test conditions (BS4718) relatively small fabrication changes and attention to sealing small gaps in silencers can make significant differences to insertion loss test values, and Bowdler, (12), has reported that poorly made or fitted ductwork, with sharp edges, for example can increase noise levels above the laboratory test values. Since the graphs and charts on which some predictions are based can only include a limited number and range of parameter values, errors can arise in interpretation and interpolation. Finally, but perhaps most important are the limitations in the accuracy of the input data, most notably in the noise emission data from fans and other items of plant and equipment, but also in acoustic data such is absorption coefficents of surfaces, transmission loss values of partitions, ductwork, ceilings.

Much research has been carried out (13), and is continuing into understanding the factors which can affect sound power levels of fans, in order to improve the accuracy of standard test procedures and to more accurately relate test values to the sound power actually emitted by the fan when installed in the HVAC system. The fan sound power will depend on its operating point on the fan-performance curve and on the installation conditions, which can affect the inlet air flow conditions and the acoustic impedance loading of the fan. The ASHRAE guide indicates that the accuracy of fan measurements ranges from (plus or minus) 6 to 8 dB at 63Hz, to 2dB in the mid-frequency range. BS848 Part 2 (1985) gives slightly lower estimates of error, in line with BS4196 Part 1 (1991) for sound power levels of other plant tested under precision laboratory test conditions. The accuracy of measurement of silencer insertion loss, according to BS4718, is ± 3dB in the 125Hz band, and ± 2dB in higher bands. When measurement data is not available, often the case in the preliminary stages of system design, the prediction of fan sound power levels is less accurate, with errors estimates varying from (plus or minus) 5 dB in mid frequencies to 10 dB at 63Hz (5), with other sources giving slightly different estimates (1,11). The estimated uncertainty in the prediction of noise levels from other items of plant such as pumps, chillers, fan casing radiation is also about ± 5 dB (1).

Estimates of the uncertainty of the various other stages in the prediction process are not usually given in the literature, with one or two exceptions. It is not therefore possible to estimate the likely overall accuracy of the predicted noise levels from the accuracy of the components. There is also a similar lack of information in the published literature about systematic and detailed comparisons of predicted and measured sound pressure levels from entire HVAC systems although many engineers and companies will undoubtedly possess their own private store of comparative data. It is because of this lack of available information on the accuracy and reliability of prediction schemes that it was decided to seek the views of experienced practitioners about prediction methods, using a questionnaire survey.

Survey of use of noise prediction methods

A questionnaire about the use of noise prediction methods was sent to a carefully selected sample of 44 manufacturers of noise control equipment and noise consultants specialising in building services noise. A copy of the questionnaire is given in the Appendix. The return and completion rate was 77%. The questionnaire was followed up by detailed discussions with a third of the sample.

The following is a brief summary of the responses to the twelve questions.

The replies indicate that respondents to the questionnaire carry out thousands of noise level predictions each year (Q(1)). They usually use a combination of different methods, modified by their own experience (Q(2)). They are, on the whole, fairly confident about the effectiveness of their predictions although there is a feeling that the standard methods overpredict noise levels (Q(4)). Although the use of computerised prediction methods is widespread many continue to use the chart/graphical methods in combination with computer software (Q(3)). Noise level measurements are not usually carried out to check predictions (Q(5)), and noise problems do not occur very often provided that good practice is observed (Q(6)). When problems do occur the most frequently reported causes are incorrectly installed, or faulty equipment. Other sources of problems are flow noise, often linked to excessive flow velocity, or poor system design, and inaccurate noise level data (Q(7)).

The determination of the fan sound power level was considered by the majority to be the most critical stage in the prediction process (Q(11)), and manufacturers data is nearly always used when available (Q(12)). Duct breakout and structure borne noise were also important. Many respondents often build a margin of error, typically 3 dB, into their prediction, depending on the application and the frequency band (Q(8)). The 125Hz band was considered by the majority to be the most critical (Q(9)), and also the lowest frequency band which could be confidently predicted, with an expected accuracy of ± 3dB (Q(10)). A detailed analysis of the replies is given in the Appendix.

In considering these replies it should be borne in mind that the respondents to the questionnaire were, on the whole very experienced in noise prediction and control, and a different set of responses may be obtained with a wider sample. The respondents felt confident to use the various prediction schemes selectively, and the modular nature of the prediction process facilitates this approach, so that, for example it would be possible to select CIBSE data for one part of the process and the ASHRAE method, or one's own data for another. Most of the respondents were well aware that the accuracy of the fan sound power level data underpins the whole process, and that values based on prediction forumalae are much less accurate than manufacturer's own test data. The follow up discussions indicated that confidence in existing prediction methods was because they tend to overestimate noise levels and that further research was still needed to improve accuracy and to reduce uncertainty. Discussions also indicated that changes in system design brought about by increasing demands to reduce the space available to the HVAC system has led to increased problems from noise breakout from ducts.

The prediction and control of duct breakout noise

A well known method for predicting noise levels breaking out from a length of ductwork is using the Allen formula (5), which is based on the theory of sound transmission through panels, adapted to account for the difference in the areas over which sound enters the duct (ie its cross sectional area) and breaks out of the duct (ie the duct surface area). This is the approach used in the CIBSE guide. It is generally agreed that this method can sometimes give very inaccurate results, particularly for low frequencies and long lengths of duct. One reason for this is that the method does not take into account that breakout from the duct will lead to a reduction in sound power within the duct along its length. The formula can lead to the prediction of negative attenuations, ie with more sound power apparently leaving the duct than entering it. This is obviously impossible and in such cases it is usual to assume that under worst breakout conditions the sound power radiated from the duct is 3dB below that in the duct. Another difficulty arises from using sound reduction index (or transmission loss) data for duct walls which has been obtained from sound transmission loss tests on plane panels rather than on in-duct tests.

Prediction methods which attempt to overcome both of these difficulties are available (5,14) and these also allow for the effect of direct sound radiation from the duct into the room, using a line source approach, as well as the reverberant component. These methods use a modified definition of transmission loss which will depend on the duct size and shape, as well as on the material type and thickness. The method of predicting duct breakout transmission loss given by Ver (14) and used in the ASHRAE guide gives much higher values than the usual data (eg in CIBSE guide) based on panel tests.

Dutfield (15) has carried out measurements of duct breakout noise using the test arrangement of Figure 1. There is no standard method for breakout testing and this arrangement, using a short length of duct, allowed the sound power radiated from the duct to be measured using the reverberant room method. Microphones inserted into the duct enabled the sound power level within the duct to be measured. The difference between these two values is whown in Table 1 together with the predictions of the Allen formula and of the Ver method. In this case the difference between the two predictions is mainly due to the use of different transmission loss figures, since for such a short length of duct the attenuation of sound power along the duct length is negligible. It can be seen that the Ver method gives better agreement with the measured results.

The usual method for reducing duct breakout is to reduce the sound power in the duct by use of an upstream silencer. Alternative noise control treatments to the duct itself include stiffening, damping, acoustic lagging and using a duct of thicker gauge. There are no standard tests for determining the effectivemess of such measures, which are difficult to predict. The results of test of different treatments carried out by Dutfield (15) using the arrangements shown in Figures 1 and 2 are shown in Table 2, in the form of insertion losses, ie the difference between the sound pressure level measurements from the untreated and the treated ducts.

Stiffening a noise radiating structure will change its vibration response, increasing the frequency of any resonances. It will also increase the acoustic radiation efficiency of the structure. These changes are, in general, difficult to predict. Although good noise reductions can sometimes be acheived by stiffening, on this occasion it has been ineffective.

The damping treatment consisted of 13mm thick bitumastic pads of 10kg/m^2 surface density. This was the most effective treatment at low frequencies. The pads add extra mass as well as damping and the lower noise reductions at the higher frequencies are probably due to this increased mass.

On the basis of simple mass-law theory of sound insulation a change from 22 to 16 gauge ducting should produce a reduction of about 5dB at all frequencies. This fails to take into account that changing the gauge also changes the duct stiffness and damping properties. The test results show that the mass law predictions are not achieved.

The duct lagging treatment gave the best overall reduction. It consisted of a 31mm thick laminate blanket comprising two layers of 25mm thick inner, and 5mm thick outer cellular foam having a density of $100kg/m^3$ and an intermediate, 1mm thick lead core having a surface density of $10kg/m^2$, and an aluminium foil facing. Treatments of this type work in a complex way. The main noise reduction mechanism is probably the isolation of the massive lead layer from the vibrating duct by the resilient foam layer, but damping, sound absorption and insulation also probably play a part. Unfortunately as with the other treatments the insertion loss is difficult to predict, as indicated by Bies and Hansen (16) and Reynolds and Bledsoe (11). The variation with frequency shown in Table 2 is typical of this type of treatment with the minimum value at 125Hz probably corresponding to a resonance of the lagging. It is not unknown for negative attenuations to be measured in some cases. Above resonance there is a steady increase in insertion loss with frequency. The results of Table 2 do not agree with either Bies and Hansen or Reynolds and Bledsoe prediction methods, but they are of similar magnitude to results reported by Fry (5).

CONCLUSION

Although there is confidence among experienced users about noise prediction methods there is still much uncertainty about the accuracy of individual parts of the prediction and the overall accuracy. There is a widespread belief that existing methods overestimate noise levels in many cases. Much of the uncertainty in the past has been due to imprecise knowledge of fan sound power, but recent advances in fan test methods will place increasing emphasis on the need for better prediction of system attenuation and noise radiation, if potential cost and energy savings from avoidance of over design of noise control are to be achieved. Further research is needed to establish standard test methods for measuring breakout noise from ducts and into the effectiveness of noise reduction treatments.

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TABLE 1

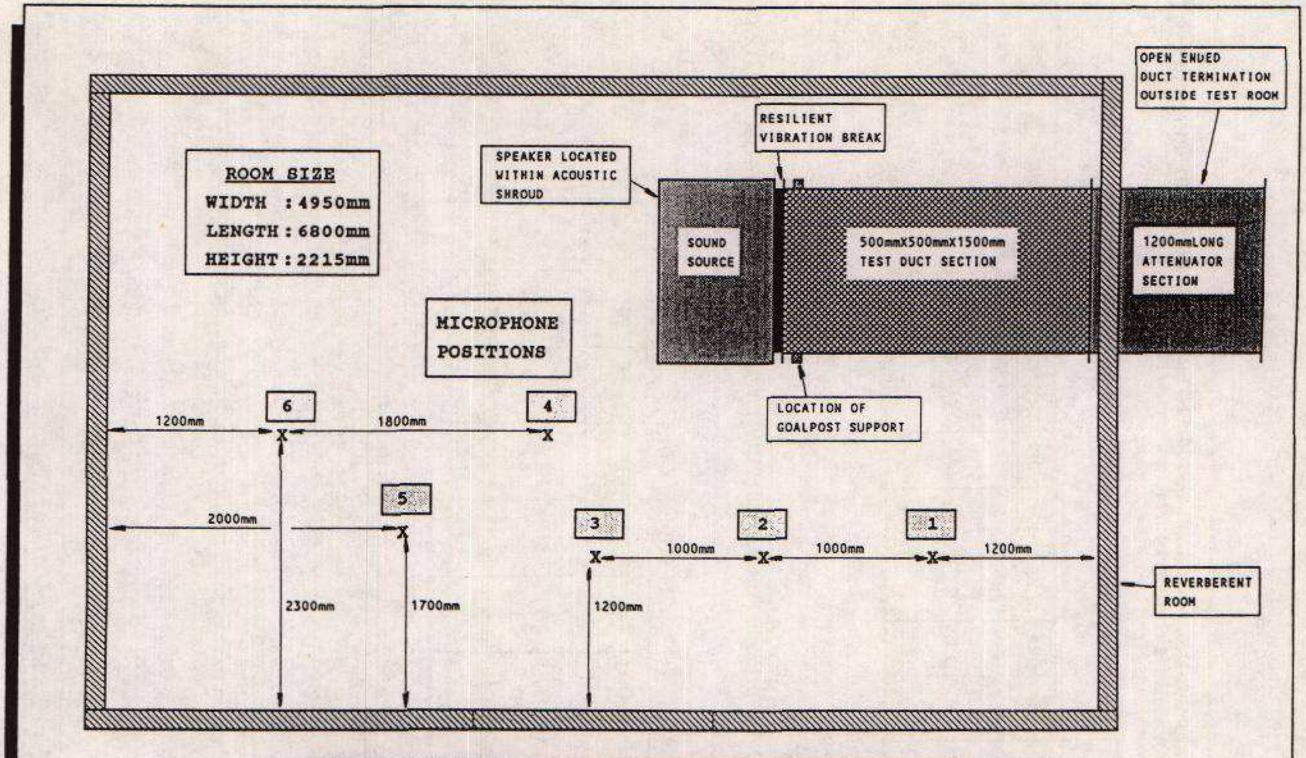
DIFFERENCES BETWEEN SOUND POWER LEVELS (in dB) ENTERING THE DUCT AND BREAKING OUT THROUGH THE DUCT WALLS (22 GAUGE DUCT)

	63	125	250	500	1k	2k	4k	8k
MEASURED VALUE	13	20	20	25	26	33	34	38
PREDICTED: VER	11	14	17	20	23	27	33	34
PREDICTED: ALLEN	3	3	3	9	12	15	16	24

TABLE 2

MEASURED INSERTION LOSS (IN dB) FOR VARIOUS DUCT TREATMENTS APPLIED TO A 22 GAUGE DUCT

	FREQUENCY (Hz)							
DUCT TREATMENT	63	125	250	500	1k	2k	4k	8k
STIFFENING	0	0	0	-1	-1	-2	-1	-3
DAMPING	7	9	4	3	4	2	2	2
LAGGING	6	1	2	4	9	10	11	11
16 GAUGE DUCT	0	5	1	4	4	2	0	-9



SKETCH SHOWING GENERAL ARRANGEMENT OF TEST EQUIPMENT AND MEASURING POSITIONS (LP & RT) WITHIN THE REVERBERENT ROOM.

FIG 1

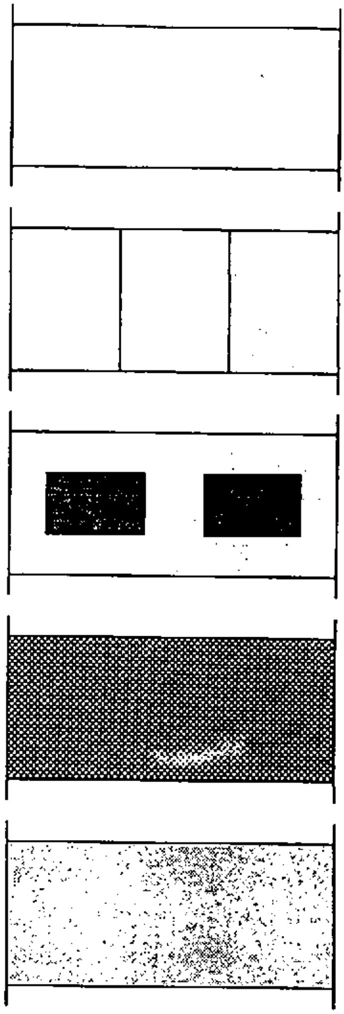
TEST 1: PLAIN 22 GAUGE DUCTING.

TEST 2: PLAIN 22 GAUGE DUCTING STIFFENED USING PERIPHERAL GIRTHS AT 500mm CENTRES

TEST 3: PLAIN 22 GAUGE DUCTING DAMPED USING 500mm x 400mm "SEPTUM" PADS.

TEST 4: PLAIN 22 GAUGE DUCTING LAGGED USING "COUSTILAM" CLADDING

TEST 5: PLAIN 16 GAUGE DUCTING.



TEST SAMPLE:500mm wide,x 500mm high,x1500mm long, galvanised sheet steel duct manufactured in accordance with ductwork specification DW142.

TEST SAMPLES

A COMPARISON OF THE EFFECTS OF VARIOUS COMMON TREATMENTS UPON "DUCT BREAKOUT" NOISE TRANSMISSION.

FIG 2

APPENDIX

THE PREDICTION OF BUILDING SERVICES NOISE

Please will you assist my research into the above area by completing this questionnaire about user confidence in current prediction methods.

Bob Peters, Nescot, February 1994.

Personal Details: (All informational will be treated in confidence and used anonymously. Please leave blank if confidentiality is a problem.)

Name: ____________________

Organisation/Firm: ____________________

Position in Organisation: ____________________

Questionnaire: Please tick as appropriate.

1. How frequently does your organisation carry out predictions of levels of Building Services noise? (number of predictions per year)

   Rarely (1-10)......, quite frequently (10-20...... frequently (more than 20)......

2. What is the basis of the prediction method used, eg.:

   CIBSE guide......, ASHRAE guide......, Woods Guide......,
   Other (eg modified versions of above, company's own etc. Please specify)..
   Don't know..........

3. Is your prediction method:

   Chart or graph based......, computerised......

4. How confident are you about the effectiveness of the predictions:

   Not confident......, quite confident......, fairly confident......, very confident......

5. How often do you carry out noise level measurements to check on the effectiveness of predictions:

   Hardly ever......, (eg less than 2 in 10) sometimes...... (3-5 in 10), fairly often...... (6-8 in 10) frequently...... (more than 8 in 10)

6. How frequently do noise problems arise in connection with installations designed on the basis of noise level predictions:

   Hardly ever......, (eg less than in 10) sometimes...... (3-5 in 10), fairly often......, (6-8 in 10) frequently..... (more than 8 in 10)

7. **When noise problems do arise with installations, what are the most frequent causes - please indicate in rank order i.e. 1 (most frequent), 2, 3, etc., where possible:**

Faulty equipment......, incorrect installation......, inaccurate prediction............, inappropriate noise level target........., other, please specify... Don't know.................

8. **When designing Building Services systems on the basis of noise level predictions, do you leave a margin of error to allow for inaccuracy in the prediction method?:**

Always......., sometimes......, never......

Please indicate magnitude of allowance..............

9. **What are the most critical octave frequency bands in the prediction process for your purposes:**

63Hz......, 125Hz......, others (please specify)..............

10. **What is the lowest octave band which you confidently expect to be able to predict?.........................**

With what degree of accuracy, (in dB)?........................

11. **What are the most critical stages in the prediction process, for your purposes. Please indicate in rank order ie 1 (most critical), 2, 3.... etc.**

Fan sound power level......, regenerated noise........, system attenuation......, prediction of room sound pressure level........, Other (please specify).., Don't know...........

12. **How frequently are you able to use manufacturers test data for sound power levels of fans and other appliances, as opposed to using values predicted from formulae? Please indicate frequency of use of manufacturer's data:**

For Fans:

Nearly always......, frequently......, only occasionally......... only rarely........

For other appliances:

nearly always......, frequently......, only occasionally.......... only rarely........

Thank you for completing this questionnaire. If you have any other comments or suggestions about this research please indicate on a separate sheet or contact me by telephone. <u>PLEASE RETURN TO</u>: Dr R J Peters PhD FIOA, Reader in the Built Environment, Nescot, Epsom Centre, Longmead Road, Epsom, Surrey KT19 8BH., A free post envelope is provided. <u>Tel: 081 394 3298</u> <u>Fax: 081 394 3232.</u>

APPENDIX - Analysis of Questionnaire Responses

Q1 Number of predictions carried out

Over 60% of respondents carried out predictions frequently, with several indicating that they carry out many more than 20 per month (hundreds in some cases)

Q2 Basis of prediction methood

CIBSE 41%, ASHRAE 28%, WOODS 47%. Most use a combination of methods modified by their own experience.

Q3 Use of Chart/Graphical or computerised methods

Chart/graph 56%, computerised 69%, with 38% indicating that they use a combination of both.

Q4 Confidence in effectiveness of prediction

Fairly confident 47%, very confident 38%. Several respondents indicated that they thought that the methods appear to overestimate noise levels.

Q5 Frequency of carrying out noise measurement checks on predictions

Hardly ever 35%, sometimes 31%, fairly often 22%, frequently 13%

Q6 Frequency of Occurence of noise problems

Hardly ever 88%, sometimes 12% with the proviso added by many respondents that advice about practice is followed.

Q7 Incorrect installation was indicated as the most frequent source of problems (63%), followed by faulty equipment (38%), followed equally by inaccurate prediction and inappropriate noise level targets. Other causes of problems which were identified include inaccurate or inadequate noise level information, air flow noise problems, poor design, contractual procedures.

Q8 Allowance for margin of error

Usually 50%, sometimes 34%, rarely 16%. The margin allowed varied between 1-2dB and 5-10dB, but most commonly, 3dB. The allowance depends on the sensitivity of the application and on the frequency band.

Q9 Most critical frequency band

125Hz (72%), 63Hz (22%) and 250Hz (22%). In some cases the band depends on the particular application.

Q10 Lowest frequency band predicted with confidence

125Hz (50%), 63Hz (22%), 31Hz (13%). The expected accuracy ranged from 2 to 5dB, but, most commonly $\pm$ 3dB.

Q11 Most critical stages in the predicion process

Fan sound power level was considered to be most important (56%) followed by system attenuation (44%). Duct breakout and structure borne noise were also indicated as being important.

Q12 Use of manufacturers data

For fans: nearly always 56%, frequently 38%

For other appliances: nearly always 31%, frequently 41%, only occasionally 22%.