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SOUND POWER DISTRIBUTION IN BRANCHED PIPING SYSTEMS

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1. INTRODUCTION

As noted by Neil Spring in the January 1985 edition of the Acoustics Bulletin, the procedures used to calculate the distribution of sound power in a building ventilation system, and hence to specify the necessary noise control components, are based upon a mixture of simplistic theory, empirical data and experience. This combination, in the hands of an experienced designer, will usually produce acceptable results. However, this state of affairs is generally unsatisfactory; to the designer, because he has no reliable means of evaluating the uncertainties in his calculations; to the component manufacturer, because he does not possess sufficiently complete knowledge to develop and optimise his designs to improve his competitive position; and to the acoustician, because he does not feel confident in the validity of the assumptions and approximations employed, some of which seem not to be consistent with physical principles.

Much of the uncertainty surrounding the calculation procedure stems from an inability to evaluate the processes of sound energy transmission and dissipation in complex duct assemblies, and to check the operational performance of individual noise control components. A case of particular difficulty is that of the reactive/resistive plenum which may be installed to control low frequency noise: no reliable theoretical analysis, and little generalised experimental data, are available to the designer.

The advent of instrumentation for the direct measurement of sound intensity holds out some promise of future improvements in our understanding of branched duct acoustics, especially if probes can be designed for measurement in turbulent, low speed flow ($M < 0.1$). This paper reports the results of some initial attempts to measure sound intensity distribution in very simple small-scale models of branched ducts, and of studies of simple attenuators.

2. OBJECTIVES

The overall purpose of the work, which is by no means complete at the time of reporting, is to use intensity measurement to investigate the validity of some of the assumptions made in predicting the sound transmission and dissipation behaviour of branched piping systems which carry low speed air flow. The aspect of particular interest is the assumption regarding the division of sound power at branch points; other questions concern the influence of acoustic load on injected power and the implication of the end-correction.

3. ADVANTAGES AFFORDED BY INTENSITY MEASUREMENT

Conventional techniques for evaluating the in-duct sound power generated by air-moving devices, and for determining the effectiveness of transmission control devices, require the provision of special purpose duct terminations which facilitate the interpretation of sound pressure measurements in terms of transmitted sound power. The two most commonly used terminations are the anechoic terminal chamber and the reverberation chamber: both such constructions are bulky and expensive. Verification of the performance of individual components in any particular operational ventilation system is of uncertain reliability, because the ideal terminal conditions do not exist: hence it is rarely attempted, and little in-situ data exists.

In principle, in-duct sound power may be evaluated at any point in a complex duct system, under any terminal condition, from axial intensity measurements, thereby eliminating the need for special terminal constructions, and introducing the possibility of in-situ evaluation of individual component performance. At frequencies above the lowest cut-off frequency of a duct section, the intensity field has to be sampled over a cross section, in order to estimate the total sound power. A method exists for determining plane wave intensity in ducts carrying mean flow [1,2], but so far extension to non-plane fields has not been achieved. This is, in principle, a serious shortcoming for practical application to low speed ventilation systems, in which the lowest cut-off frequency in the larger main ducts may be less than 200 Hz. However, research is in hand to evaluate the practical significance of this constraint. For the moment, higher order mode transmission tests must be restricted to static (zero flow) conditions.

As an example of the advantage afforded by intensity measurement we may cite the possibility of investigating the interaction between in-line dissipative and reactive elements, by direct evaluation of the dependence of the sound power dissipated in the former element on the presence, and form, of the latter. Similarly, the dependence of fan sound power on acoustic load may be directly evaluated. It must be carefully noted that the measurement of acoustic power entering and leaving a duct component is not a direct measure of transmission loss; what is measured is the nett power entering and leaving a device, and hence the nett power dissipated (and/or generated) by the device.

4. EXPERIMENTAL ARRANGEMENT

The experimental arrangement consisted of a heavy wooden rectangular box of dimensions 1 × 0.5 × 0.5 m, into which air was blown by a small centrifugal fan, and onto which could be connected heavy gauge circular section plastic pipes of 106 mm internal diameter (lowest cut-off frequency at 20°C = 1895 Hz). The box was lined with absorbent plastic foam, and contained an 8" loudspeaker mounted against one side wall: a bypass hatch could be opened in the box to control the air flow speed in the pipe without altering the fan inlet, or pipe outlet conditions. A pair of phase matched Brüel & Kjaer 4134 microphones could be mounted flush to the inner surface of the pipe wall with separation distances of 40 or 15 mm. The loudspeaker could be driven by a PRBS signal from a Solartron 1200 signal analyser, which was also used to analyse the microphone signals. The signal generation and analysis procedure was controlled by a microcomputer. The overall system had an inter-channel phase mis-match of less than 0.1° in the range 30-4000 Hz.

The basic procedure was to evaluate the axial sound intensity spectrum using a corrected form of the Chung and Blaser equation [1,2], which accounts for mean flow in the plane wave case ($f < 1895$ Hz):

$$I_f = \frac{S_{11}}{4\rho c \sin^2\left[ks/(1 - M^2)\right]} \left\{ (1 + M)^2 \left| \exp\left[j\left(\frac{ks}{1 - M}\right)\right] - H_{12} \right|^2 \right.$$

$$\left. - (1 - M)^2 \left| H_{12} - \exp\left[-j\left(\frac{ks}{1 + M}\right)\right] \right|^2 \right\}. \qquad (1)$$

Measurements were also made at frequencies above the cut-off frequency, but without flow, in which the microphone pair was located at four different radial positions on a pipe cross section in order, approximately, to estimate the total axial sound power flow. Spectral intensities, initially computed with resolutions of 4 or 10 Hz, were combined into 100 Hz bands for ease of presentation of the data. Repeatability was found generally to be better than ±0.5 dB, and overall error is considered not likely to exceed ±1 dB.

5. EXPERIMENTAL CONFIGURATIONS AND TEST RESULTS

5.1 Sound power injected into a straight pipe with various terminations

Various lengths of straight pipe, having various terminal conditions, were attached to the box, and the relative amounts of sound power injected into the pipe by the fan alone were determined. Figure 1 presents the results of these tests. There is clearly an effect at lower frequencies of acoustic load (pipe entry impedance) on injected power, which is not solely determined by the exit (radiation) impedance, as seen from the difference between the long and short pipe results. However, the virtual disappearance of the differences at frequencies above 1000 Hz shows that the resonant behaviour of the pipes, created by end reflection, which diminishes at frequencies approaching cut-off, does affect injected power: the longer uniform pipe, with the higher longitudinal acoustic modal density, accepts more power than the short pipe, which has resonances at 170 Hz intervals. The narrow band intensity spectrum in the conically terminated pipe exhibited only very broad harmonic spectral peaks, unlike those in the straight pipes which were strongly resonant. The results do not necessarily imply that the fan sound power output changed with pipe entry impedance because the intervening plenum box absorbed an unknown fraction of its output.

5.2 Attenuation by a right-angle bend

Zero-flow intensities were measured upstream and downstream of a right-angle bend. The inner bend radius was zero (sharp corner) and the outer radius was one pipe diameter. In addition to a frequency-independent difference of 1 dB, attributable on the basis of earlier measurements entirely to straight-pipe dissipation, narrow dissipative attenuation peaks of 7-10 dB appeared at frequencies just above the straight duct cut-off frequencies (1895, 3142, 3945, 4326 Hz). It is not known if flow affects these peaks, because intensity could not be measured in flow above 1895 Hz.

5.3 Attenuation by an expansion chamber

A simple expansion chamber of area expansion ratio 4.7:1 and length 360 mm, made of 12 mm thick dense paper board, was inserted into the pipe at 1 m from entry.

Differences between sound power entering and leaving the chamber, in the case of zero flow, are plotted in Figure 2, for open pipe and anechoic termination conditions. Unexpectedly, the empty chamber with open pipe termination produced dissipative attenuation of between 2 and 12 dB, the peak attenuation corresponding to the classical insertion loss peaks. By contrast the lower frequency attenuation of the anechoically terminated chamber averaged only 2 dB. There is no evidence of a resonance at the expansion chamber cut-off around 875 Hz, which suggests that only longitudinal, and not transverse, acoustic resonance creates dissipation. An influence of end reflection (terminal impedance) is clearly confirmed by the convergence of the curves at frequencies approaching cut-off. Mean flow speeds in the pipe of up to 10.6 ms^{-1} did not significantly affect the open pipe result, except at frequencies below 500 Hz where aerodynamic noise generation was observed. The mechanism of the observed dissipation is assumed to be vorticity generation at the ports.

The expansion chamber was lined with 50 mm thick absorbent plastic foam and the tests were repeated with anechoic and open pipe terminations. The results, shown in Figure 3, again indicate that the dissipative attenuation is influenced by the termination condition. The upstream sound power entering the expansion chamber was quite unaffected by the termination. The influence of the end reflection on power dissipation by the absorber clearly illustrates the rôle of end reflection, at least in a simple series system; it enhances dissipative attenuator performance!

5.4 Division of sound power at a branch take-off

The division of sound power between a straight pipe and a right-angle branch take-off was investigated for various termination and flow conditions. The branch was of the same diameter as the main pipe, and the intersection was sharp edged. Flow effects could only be investigated up to 1895 Hz. A configuration of particular interest was that in which the main pipe was anechoically terminated to form a "dead-leg" attenuator. The results are presented in Figure 4. In the case with both pipes open, the sum of the powers in the two legs downstream of the junction was not more than 1 dB lower than the measured incident power, a difference mainly attributable to the straight pipe dissipative losses mentioned in section 5.2: hence it is concluded that, in this case, dissipation at the junction is negligible. The sound power division is clearly strongly affected by the proximity of duct cut-off frequencies. The "dead-leg" attenuator is seen significantly to reduce the sound power transmitted round the bend, even at low frequencies, although here its effect is clearly a function of the impedance of the downstream leg, as demonstrated by the effect of terminating the latter anechoically. Mean flow of speeds up to 10 ms^{-1} had little influence on the results below 1895 Hz.

6. CONCLUSIONS

(1) Dissipative mechanisms may contribute significantly to the attenuation produced by ostensibly purely reactive devices.

(2) End reflections enhance energy dissipation by resistive attenuators.

(3) The division of sound power at a side branch take-off does not seem to be related to the division of volume flow, or to the area ratio: it is much

more a function of the higher order mode cut-off frequencies of the ducts.

(4) Because more sound energy travels straight on in the main duct than into the side branch, a "dead-leg" attenuator which communicates acoustically with a duct through a porous, aerodynamically smooth bend panel could provide useful attenuation at a bend without affecting the bend pressure loss, or generating flow noise. Two, or more, such attenuators, placed in series at bends of successively smaller dimension, could significantly reduce the necessary length of in-duct splitter attenuators, which produce pressure loss and aerodynamic noise. (Note that the model frequencies are 5 to 10 times the equivalent full scale frequencies.)

(5) Further work needs to be done on branch take-offs of smaller cross-sectional area than the main duct.

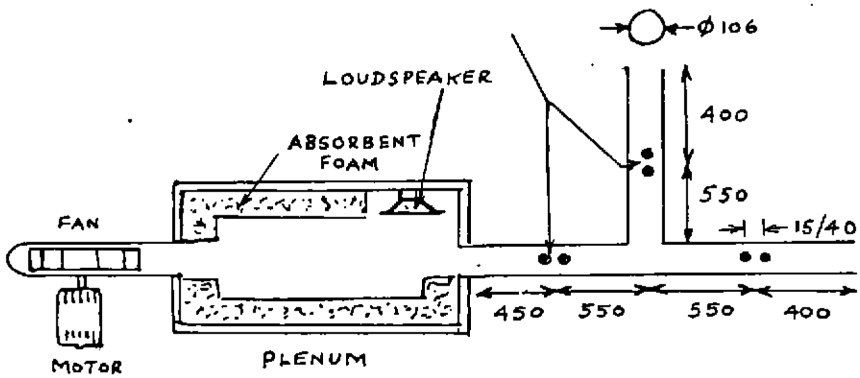
APPENDIX

The ratio of nett incident to transmitted powers in a system containing an attenuator of transmission coefficient τ, followed by a termination of sound power reflection coefficient α_r, is $W_i/W_t = (1 - \tau^2\alpha_r)/\tau(1 - \alpha_r) \sim [\tau(1 - \alpha_r)]^{-1}$ if $\tau << 1$.

REFERENCES

[1] J.Y. Chung and D.A. Blaser, "Transfer function method of measuring acoustic intensity in a duct system with flow", J.A.S.A., Vol. 68, No. 6, 1570-1577 (1980).

[2] M.S. Vasudevan and F.J. Fahy, "Comment upon 'Transfer function method of measuring acoustic energy in a duct system with flow' (Chung and Blaser, J.A.S.A. 68(6), 1570 (1980))", Letter to the Editor of J.A.S.A. - submitted for publication.



Sketch of experimental arrangement for open branch test.

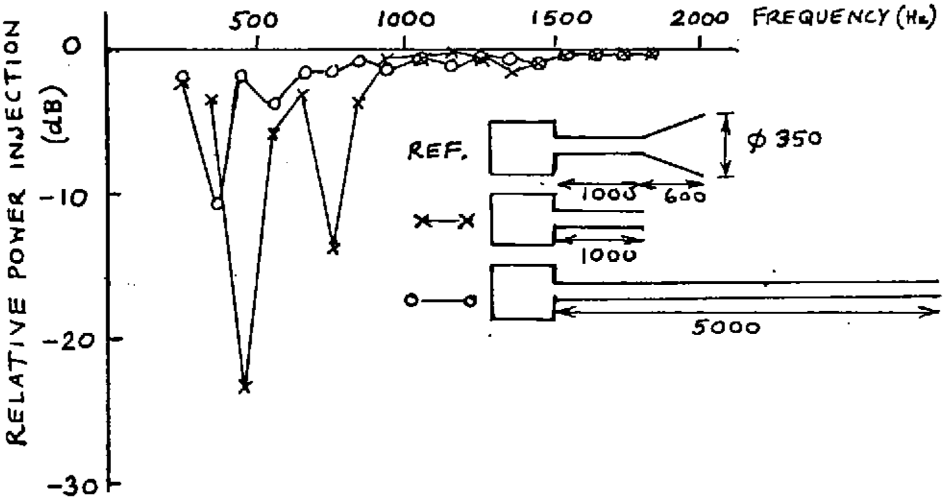


Figure 1. Relative sound power injected into pipes with various terminations.

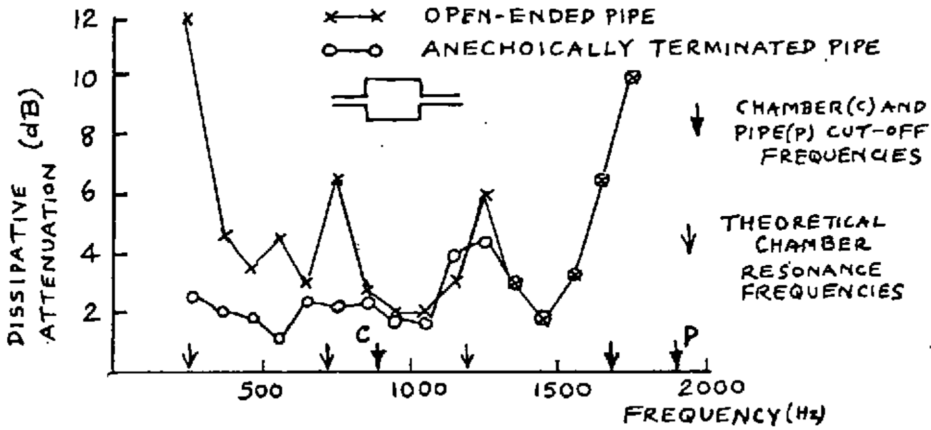


Figure 2. Dissipative attenuation of an unlined expansion chamber.

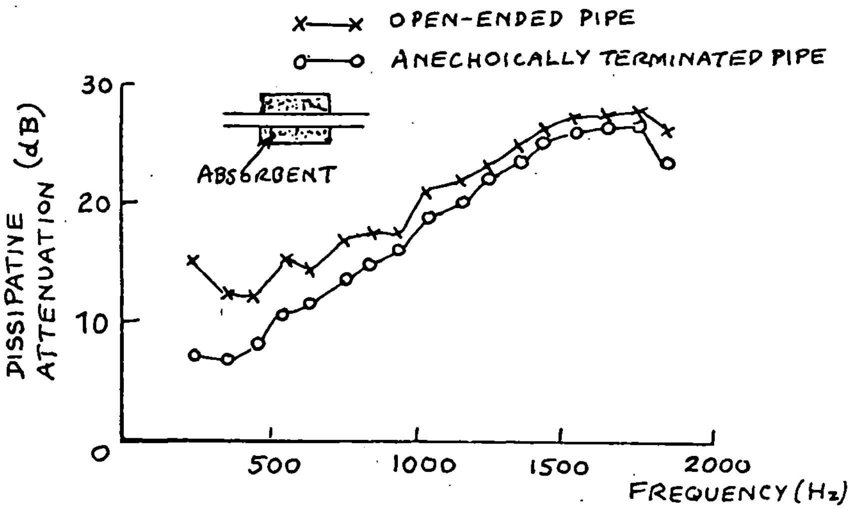


Figure 3. Dissipative attenuation of an absorbent expansion chamber.

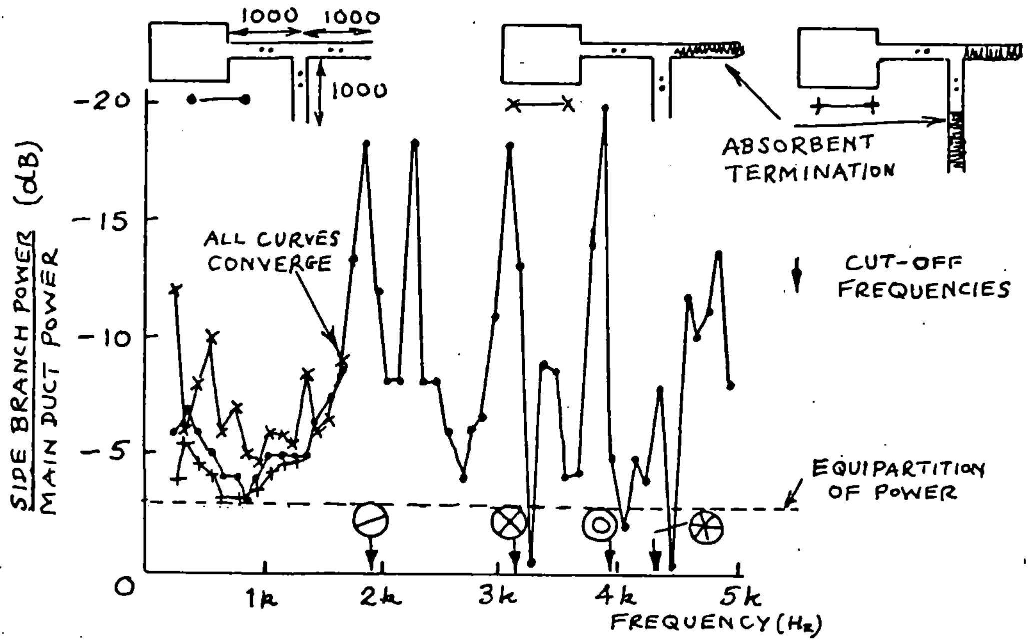


Figure 4. Ratio of side branch sound power to main duct incident sound power.