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STIFFNESS CONTROL OF LOW FREQUENCY NOISE BREAKOUT FROM RECTANGULAR DUCTS

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Introduction

Previous work by the author (1,2,3) has shown that low frequency noise "breakout" from the walls of rectangular ducts may be successfully modelled by assuming the existence of a coupled acoustic/structural wave system, in the fluid contained in the duct and in the duct walls, with uniform acoustic pressure fluctuations on a duct cross section. In references (1) and (3), the mechanisms of the wave transmission and of the external acoustic radiation were examined. In reference (2), the behaviour of external lagging on duct walls, as a noise control treatment, was studied, and it was concluded that, to be effective, lagging must be carefully designed, and that it is anyway an expensive and inconvenient means of noise reduction.

The present article presents an alternative approach to the reduction of breakout. In typical curves of duct wall transmission loss (TL) versus frequency, a series of dips appears; these are associated with transverse wall resonances. The dip at the lowest frequency is caused by the fundamental duct wall resonance, and below this, the TL rises at about 9dB per halving of frequency. In this region the TL is "stiffness controlled". In typical air conditioning ducts, the fundamental frequency (denoted here by f_1) may vary between about 5 Hz and 200 Hz, depending on the size and wall material of the duct; usually, f_1 would be in the range 30 - 100 Hz. Clearly, the stiffness controlled region would normally be at such low frequencies as to be of little practical significance, and duct wall resonances - which are the most troublesome aspect of breakout - would occur in the critical frequency region of 50-300 Hz, where fan noise (for example) is most energetic, and where duct silencers are least effective. If, however, one can substantially increase the ratio (g/m) - where g and m represent the flexural rigidity and mass per unit area, respectively, of the duct walls - then f_1 rises to much higher frequencies, and the region of stiffness control can be utilized, both to increase the low frequency TL generally, and to remove low frequency wall resonances. This is the method adopted in the present piece of work.

Theory

The theoretical approach is essentially that reported in references (1) to (3): the duct wall response to the internal pressure field is calculated on the basis of thin plate theory, due regard being taken of boundary conditions at the duct corners. Then the external radiation is estimated, on the assumption that the duct radiates like a line source of finite length. One finds that, below f_1, the TL may be approximated by a "quasi-static" expression which, for a square duct, is:

$$TL = 10\log\left[1800E^2h^6/(1-\sigma^2)^2\rho_0^2 c_x \omega^3 a^6\right],$$

where E, h and σ are the Young's modulus, thickness and Poisson's ratio of the duct walls, ρ_0 is the density of the fluid inside and outside the duct, c_x is the axial phase speed of the wave system, ω is the radian frequency and a is the width of the duct. (The TL is defined as the logarithmic ratio between the acoustic power in the duct, and the radiated sound power per unit length of duct). The ω^3 factor gives the -9dB per octave slope. An equivalent expression exists for rectangular ducts. These formulae are useful for engineering purposes, since one may easily estimate the TL in the stiffness controlled region.

A feature which is not of great concern in the case of ordinary sheet metal ductwork, but does become of more interest in the case of ducts with a high g/m ratio, is that there is generally more than one way in which the acoustic and structural waves can couple together (the number of combinations depends on frequency: below f_1, there is only one; between f_1 and the next duct resonance, there are two; between this region and the following resonance, three, and so on). These are distinguished by having differing axial wavenumbers, and also different distributions of energy flow between the acoustic wave in the duct and the structural wave in the walls. In most (though not all) cases, the predominant energy path is either acoustic or structural. Accordingly, one may label these wave systems "A modes" or "S modes", respectively. The S modes tend to be relatively "leaky", in that they easily transmit sound through the duct walls, whilst the A modes tend to transmit much less energy. The mode existing below f_1 is, fortunately, an A mode, and the quasi-static TL formulae apply. Above f_1, estimating the TL is complicated in the case of stiff walled ducts. Both A and S modes exist, but only one mode - an A mode - tends to be strongly excited (though to what extent is difficult to ascertain accurately). The nub of the matter here would appear to be that, in the case of stiff walled ducts, the TL of the strongly excited A mode is so high that any small energy sharing with an S mode is apt to be very noticeable, whilst with relatively limp walled ducts, the A mode does not have a very high TL in the first place, so that A mode transmission tends to predominate. These arguments, it must be pointed out, are based largely on theoretical considerations and have not yet been fully investigated experimentally.

Measurements and Comparison with Theory

In order to test the idea of stiffness control of the duct wall TL at low frequencies, a series of ducts was constructed, and the TL of each was measured by injecting an acoustic wave into one end of the duct and measuring the radiated sound power by the reverberant room method. Anechoic terminations were incorporated in all three ducts. The first duct was constructed of expanded polystyrene sheet, 38mm thick. The duct was 240mm square. Although the Young's modulus of this material is low, the specific gravity is extremely low and the thickness relatively large. Thus the g/m ratio was high, and this gave f_1 as 560 Hz, much higher than that for comparable metal ducting. Figure 1 shows the measured and predicted TL (which are in good agreement). Although the TL below f_1 is modest (because, although g/m was large, g itself - which controls the TL below f_1 - was actually little higher than that typical of ordinary sheet metal duct wall material), the fact that low frequency wall resonances are absent means that the worst feature of breakout has been eliminated. Figure 2 shows the TL of a 203mm square 18 gauge steel duct, for comparison; one observes the resonance at 175 Hz, which would certainly cause problems in practice. Expanded polystyrene has obvious practical disadvantages (mainly to do with fire protection requirements) despite its low cost and very light weight (1/16th of that for the equivalent 18 gauge galvanised steel ducting!), so other materials were tested. One suitable duct wall construction involved the use of a sandwich material consisting of two 0.91mm Aluminium sheets, resin bonded to a resinated paper honeycomb core; the laminate was 13mm thick. It was little more than half the weight of the equivalent steel sheet, and was extremely rigid. Some problems were encountered with the corner joints (the sheet could not simply be bent), which could not be made of comparable rigidity to the sheet itself. In the event, the corners behaved as if they were "pin jointed". Figure 3 shows predicted results (for pin jointed corners) and measurements on a 216mm square duct. Although the quantitative agreement leaves something to be desired (because of other "non-idealities" in the duct's construction), the principle is seen to work very well, and a very substantial TL is achieved at low frequencies. (A further duct was constructed from a similar sandwich material, 25mm thick. This gave comparable TL figures; they were not higher, because some delamination of the sheet had occured - this is one of the difficulties in working with sandwich materials of this type).

During the tests, a duct was constructed of a sandwich material consisting of 22 gauge galvanised steel facing sheets with a rigid polystyrene foam core. This proved disastrously unsatisfactory, since a large dip appeared in the TL curve at about 350 Hz, almost certainly caused by a resonance <u>within</u> the wall material itself. The foam was much less rigid than the honeycomb core, in compression normal to its plane, and this tended to lower the resonance frequencies of the sandwich material to a region where they could prove troublesome.

Discussion

Clearly, the stiffness control method for reducing low frequency breakout can be successfully applied, at least in a laboratory situation; further tests are needed on its implementation in practical ventilating duct work, or similar systems. Great improvements in wall TL can be achieved, and in principle, at least, the problems associated with breakout in noise sensitive areas could be completely eliminated. The most promising material studied so far is the honeycomb sandwich, but this brings with it certain difficulties in duct construction. These, however, are mainly associated with the "one off" methods which necessarily were adopted during the tests. No doubt more efficient means of fabrication can easily be developed. More development work is also required on duct wall materials, and further investigation into the relative roles of A and S modes in determining the net TL of a given duct configuration needs to be carried out.

References

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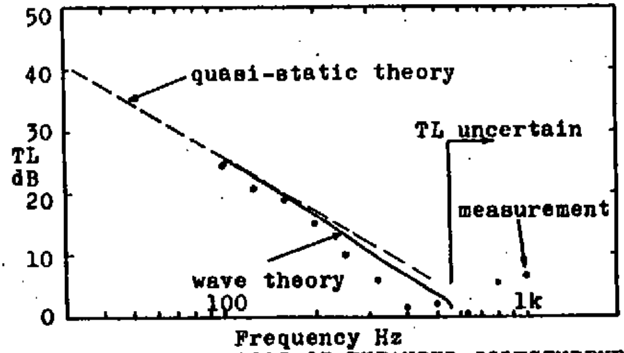

FIGURE 1 TRANSMISSION LOSS OF EXPANDED POLYSTYRENE DUCT

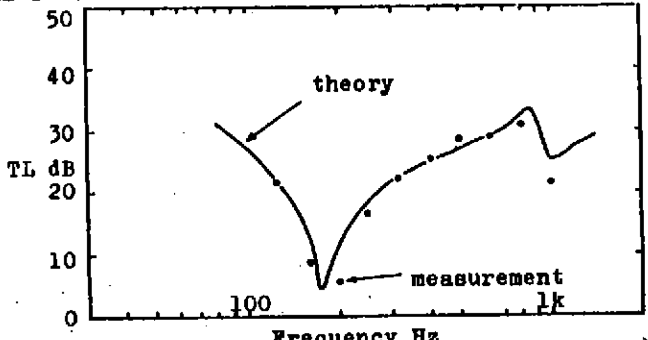

FIGURE 2 TRANSMISSION LOSS OF 18 GAUGE STEEL DUCT

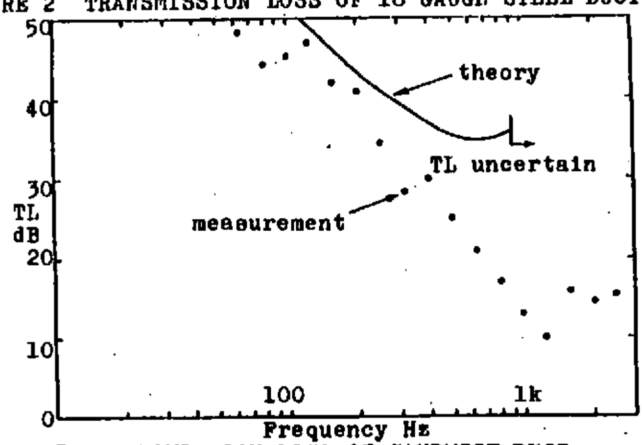

FIGURE 3 TRANSMISSION LOSS OF SANDWICH DUCT