Proceedings of The Institute of Acoustics

A STUDY OF SOUND ABSORPTION IN A SMALL ENCLOSURE

E.W. Taylor

BBC Research Department, Kingswood Warren, Tadworth, Surrey, KT20 6NP

SUMMARY

Variations in effective absorption which occur when an absorber is moved relative to the standing-wave (room mode) pattern of sound pressure in an enclosure are described, and mechanisms of sound absorption which could account for the observed variations are discussed.

1. INTRODUCTION

The 'reverberation room' method of assessing the sound absorption properties of materials, in which the reverberation times of a room with low inherent sound absorption is measured with and without a sample of the material present, is well known. The resulting frequency-dependent sets of 'absorption coefficients' are used to determine the surface areas of the various materials used in the acoustic treatment of an enclosure (studio, concert hall, etc.) required to give a satisfactory reverberation-time characteristic. This relatively simple technique works reasonably well when the dimensions of the enclosure and patches of absorbing material are large compared with the sound wavelength. Anomalous results may however be obtained when these quantities are comparable. In this case it must be remembered that the reverberant sound energy in a room exists as a set of resonances (room modes or eigentones) each of which is characterised by a standing-wave pattern of sound pressure, and that the process of sound absorption is in some ways akin to the termination of a transmission line by an impedance which is both reactive and resistive, and further complicated by the three-dimensional nature of both the transmission line (the room) and the terminating impedances (the sound absorbers). In the present work the treatment of this subject by Dowell [1] was taken as a starting-point. It was however found that the mechanism of sound absorption implied by Dowell's theoretical relationships did not fully account for sound absorption that actually occurred in practice : this forms the subject of the present paper.

2. THE THEORETICAL RELATIONSHIP

The theoretical relationships between the absorbing-material and room-mode parameters are given in detail elsewhere [1,2,3] and will be summarised very briefly.

Consider a rectangular room having dimensions L_x, L_y, L_z along the directions of the co-ordinate axes x, y and z. For the present, attention is confined to one of the surfaces normal to the z-axis, where a rectangular patch of absorbing material is placed. This patch has its corners at co-ordinates (a_x, a_y), (a_x, b_y), (b_x, b_y) and (b_x, a_y), where b > a. The room is excited at such a frequency that there are n_x, n_y and n_z pressure nodal planes perpendicular to the x, y and z axes respectively. Under these conditions an

A STUDY OF SOUND ABSORPTION IN A SMALL ENCLOSURE

'integration factor' I can be defined, such that:

$$I = \frac{1}{4}(b_x - a_x + A_x)(b_y - a_y + A_y) \qquad \ldots\ldots (1)$$

In Equation 1 the factor A_x has the form

$$A_x = \frac{L_x}{2n_x}\left[\sin\left(\frac{2n_x b_x}{L_x}\right) - \sin\left(\frac{2n_x a_x}{L_x}\right)\right] \qquad \ldots\ldots (2)$$

and A_y is defined in a similar manner. A 'damping factor' D may now be defined, such that:-

$$D = \frac{1}{\epsilon_{n_x}\epsilon_{n_y}\epsilon_{n_z}} \cdot \frac{I}{\xi} \qquad \ldots\ldots (3)$$

Here ξ is the specific acoustic impedance of the surface (assumed real), and the ϵ values take the value unity if the corresponding value of n is zero, and the value $\frac{1}{2}$ otherwise.

Damping factors may also be found for other patches of absorbing material, both on the same room surface as the one considered above, and on the other five room surfaces, in each case using relationships analogous to those shown in Equations 1-3 above. These damping factors may be summed to give an overall damping factor D_o. The reverberation time of the room (T) is then given by:-

$$T = \frac{6V \log_e 10}{cD_o} \qquad \ldots\ldots (4)$$

V and c being the volume of the room and the velocity of sound respectively. Finally, when calculating the sound decay characteristics over a band of frequencies, the whole process of calculation may be repeated for other room mode parameters (i.e. different values of one or more of the quantities n_x, n_y and n_z) whose frequency falls within this band, and the overall decay characteristic ascertained.

It can be seen that the above relationships could in principle be used instead of the Sabine or Eyring formulae to predict reverberation times in a treated room : in this case it would be the values of ξ that are carried over from the reverberation-room tests to the determination of absorber type, size and position required for the treatment. Comparing Equation 4 with the Sabine reverberation-time equation indicates that:-

$$D_o = \frac{S}{4}\alpha_E \qquad \ldots\ldots (5)$$

where α_E is the 'effective' absorption coefficient. The difference between α_E and the 'conventional' absorption coefficient is that it depends on the interrelation between absorber position and room-mode structure. The form of each term in brackets in Equation 1, taking also into account the expansion of the factor A as shown in Equation 2, implies in fact that Dowell's relationships involve the assumption that sound absorption at a point on the surface of an absorber is proportional to the potential energy density (or in other words the square of the sound pressure) at that point. Thus the total absorption is proportional to the integral over the curve relating potential energy density to position on the surface of the room on which the absorber is placed, for that part of the curve corresponding to the position of the absorber. In general the potential energy density may vary in two orthogonal directions over the surfaces and integration is carried out over the complete surface (hence the form of Equation 1 as two terms multiplied together): the process is illustrated in Fig. 1 in which integration along one dimension only is shown. This has direct relevance to the experiments described later in this paper, which were undertaken to test the validity of the theoretical procedure outlined above.

3. TEST PROCEDURE

The experiments were carried out using a small enclosure (0.82m x 0.67m x 0.71m high; volume $0.39m^3$) which had been constructed as a 'reverberation room' in connection with acoustic modelling tests. The walls and base of the enclosure were made of 12mm thick steel plate, in order to minimise losses due to coupling between the structure and the sound field inside the enclosure. The lid of the enclosure was made of 12mm thick acrylic sheet : some difficulty was experienced due to losses caused by vibrational excitation of this component but these were minimised [3] for the particular modes of air vibration used during the tests. A reverberation time of about six seconds was achieved under the test conditions, with no added absorption : this represents an average absorption coefficient over the surfaces of the enclosure of 0.003, which is near the theoretical limit that may be expected from boundary layer considerations [4a]. Sound excitation was introduced through a 18mm hole in one wall, near a corner of the enclosure, using a small external loudspeaker sealed to the wall. Excitation frequencies (208Hz and 257Hz) corresponded to the lowest-order axial modes involving reflection between opposite pairs of vertical walls.

The absorber used in the tests was designed to introduce as much absorbing material as possible, consistent with being reasonably sensitive (at least in theory) to its position relative to the standing-wave pattern of air vibration in the enclosure. This was achieved by making it the full height of the enclosure (0.7m) but only 67mm wide. It was placed with its long dimension vertical (Fig. 2) so that contours of equal sound pressure were parallel with this dimension, and in one of the two alternative positions shown in the Figure. In this way (Fig. 3) its absorption was measured at a position of either maximum or minimum potential energy density in the enclosure. The measurement was carried out either by observing sound decay (reverberation time) directly in the conventional manner, or alternatively by measuring the resonance bandwidth and calculating the corresponding reverberation time.

A STUDY OF SOUND ABSORPTION IN A SMALL ENCLOSURE

The absorber was constructed of high-density mineral wool in the form of a self-supporting slab 31mm thick : this material was chosen as being 'locally reacting' and unlikely to show any resonance. The sides of the slab were covered with adhesive tape and the absorber was attached to the surface of the enclosure with further tape. For most tests this attachment was carried the full length of the absorber, thus effectively sealing the gap between the back of the absorber and the wall of the enclosure : some tests (see Section 5) were however carried out with this seal removed, the absorber then being attached to the wall at each end only.

4. RESULTS OBTAINED USING ONE ABSORBER, SEALED TO WALL

Values of damping factor were obtained for the absorber sealed to the wall at each of the positions shown in Figs. 2 and 3 and for the two lowest-order axial modes discussed in Section 3, by applying Equation 4 to reverberation-time measurements made with the absorber in position and with the absorber absent. In the present context the ratio of damping factors, found when the absorber is placed first in a region of maximum potential energy density and then of minimum potential energy density, is of significance. In one experiment a value of three was found for this ratio, while in another experiment under rather different conditions (see Section 6) the value was four: both these values are averages found over the two excitation frequencies, and for two repeats of each test condition.

Theoretical damping factors can be calculated using Equations 1-3. Again, the ratio of damping factors found in position of maximum and minimum potential energy density are of interest in the present context : this is simply the ratio of the integration factors (Equation 1) for the two absorber positions, or in other words the ratio of the areas under the potential energy density curve bounded by the absorber edge position as shown in Fig. 1. The values for this ratio found for each axial mode used in the tests differ to some extent from each other, since the absorber width is a different fraction (0.08 in one case and 0.1 in the other) of the enclosure dimension along which potential energy density varies with position (see Fig. 3) : on average, however, the ratio value is 150. It can thus be seen that although in qualitative terms the practical effect of moving an absorber from a region of high potential energy density to one of low potential energy density (i.e. the reduction of the damping factor value or effective absorption coefficient) is as predicted by theoretical considerations, in numerical terms the practical effect is far less than is predicted by theory.

Two possible reasons for this disagreement between theory and practice are examined in Sections 5 and 6.

5. RESULTS OBTAINED WITH ONE ABSORBER, NOT SEALED TO WALL

It was stated in Section 1 that the theoretical relationship between effective absorption and position of the absorber relative to the sound field implied that absorption was proportional to potential energy density at the surface of the absorber. Thus in the region of a sound pressure node, where the potential energy density is at a minimum, sound absorption will also be at a minimum. A corollary of this implication is that the kinetic energy component of the sound field plays no part in determining the amount of absorption taking place. Remembering that the kinetic energy density in a sound field is at a maximum where the potential energy density is at a minimum, it can be appreciated that if an absorber is sensitive to both the potential and the kinetic energy sound-field components (i.e. so that one component of the absorption depends on the sound pressure at the absorber and another component depends on the sound pressure-difference), then the ratio of the effective absorption found at positions of maximum and minimum potential energy density will depend on the relative importance of these two components. If the sensitivity to sound pressure difference is significant but still smaller than that due to sound pressure, the absorber would still show reduced absorption at a region of potential energy density minimum, but to a smaller extent than predicted by Equations 1-3. Greater sensitivity to sound pressure-difference than to sound pressure would in fact give rise to a maximum of absorption where theory predicted a minimum.

The possibility of sensitivity to sound pressure-difference was investigated by carrying out measurements with the seal along the long edges of the absorber removed (Fig. 4). This left a gap of a few millimetres between the absorber and the wall. Under these conditions there was little difference in absorption in positions of maximum and minimum potential energy density. In part, this change from the results obtained when the gap was sealed was indeed caused by an increase in absorption at the position of minimum potential energy density (or maximum kinetic energy density) by a factor of 1.4, due presumably to losses in air driven through the gap by the pressure-difference across it (Fig. 5). Interestingly, however, there was also a decrease in absorption (by a factor of two) at the position of maximum potential energy density. This is thought to be caused by the creating of a 'neutral plane' of air movement in the absorber, effectively halving its thickness and reducing its absorption by an amount greater than is compensated for by the effective doubling of the surface area.

6. RESULTS OBTAINED USING 'SHIELDING' ABSORBERS

If a relatively small isolated patch of absorbing material is present in an enclosure with walls of low absorption, energy may flow into it from parts of the sound field not immediately adjacent to it (Fig. 6, left-hand diagram). This effect may be enhanced if a dimension of the absorber is small compared with the sound wavelength : there is an extensive literature on 'diffraction effects' or 'edge effects' in this context. In such circumstances it may be more realistic to ascribe an 'effective dimension' [3] or 'absorber cross-section' [4b] to the absorber, rather than use its actual physical dimensions, when assessing its absorbing properties.

The possibility of sound absorption depending on the potential energy density of the sound field remote from the absorber itself was investigated by measuring the damping factor introduced by an absorber when it was flanked by two 'shielding absorbers' (Fig. 6, right-hand diagram). Under these conditions it could be expected that sound energy flow into the test absorber would only be from immediately in front of the absorber, as shown. As a 'control' experiment, the test absorber was placed on the opposite wall of the enclosure (Fig. 7) so that it was in an equivalent position as far as the standing wave pattern in the enclosure was concerned, but the energy flow was not affected by the shielding absorbers. The 'empty' room reverberation times, required to calculate damping factor values, were measured with only the shielding absorbers present: thus the measured damping factor referred only to the central test absorber. It was found that the effect of the shielding was to reduce the amount of absorption by a factor of nearly five when the test absorber was at a sound pressure minimum, but that shielding had no effect at a sound pressure maximum. The ratio of absorption at positions of maximum and minimum potential energy density increased to a value of about 20 under these conditions (compare with the values shown in Sections 4 and 5). It appears that if an absorber is placed in a region of low potential energy density, sound energy will flow into it from neighbouring regions of higher potential energy density. This effect does not however occur if the potential energy density distribution is uniform. The 'effective dimension' principle is therefore hard to apply, as this quantity changes with the nature of the sound field surrounding the absorber.

It may be noted that the 'control' experiment gave similar results to the condition first tested (see Section 4) with one absorber sealed to the wall of the enclosure.

7. CONCLUSIONS

The practical tests described briefly in this Paper show that theoretical calculations of sound absorption as a function of absorber position in a sound standing wave pattern are not likely to give accurate predictions of absorption, if they are based on the premise that absorption at a point on the surface of an absorber is proportional only to the potential energy density (or square of sound pressure) at that point. Three reasons for the lack of correspondence between theory and practice have been identified. In the first place, a component of sound absorption may be dependent on the kinetic energy density at the absorber : this is at a maximum where potential energy density is at a minimum. Secondly, sound absorption may depend on the potential energy density in regions of the sound field that are not immediately adjacent to the absorber. And thirdly, the amount of absorption may be influenced by the nature of the sound field : for example, the degree to which sound energy flows into the absorber from regions not immediately adjacent to it depends on whether the potential energy density does or does not vary with position.

8. ACKNOWLEDGEMENTS

The author acknowledges with thanks the help given to him by P.H.C. Legate in carrying out the experimental work described in this paper, and the permission of the Director of Engineering, BBC, to publish it.

A STUDY OF SOUND ABSORPTION IN A SMALL ENCLOSURE

9. REFERENCES

1. DOWELL, E.H. Reverberation time, absorption and impedance. J. Acoust.Soc.Am., 64, 1 (July 1978), pp. 181-191.

2. TAYLOR, E.W. A preliminary study of the influence of room mode structure on sound absorption. BBC Research Department Report No. 1983/4.

3. TAYLOR, E.W. Room modes and sound absorption : some practical measurements compared with theoretical predictions. BBC Research Department Report No. 1985/11.

4. KUTTRUFF, H. Room acoustics. Applied Science Publishers, Ltd., London 1973. (a) pp. 128-130. (b) p. 138.

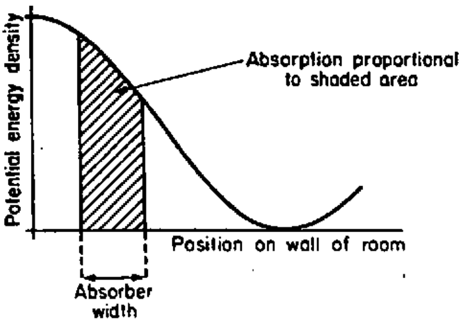

Fig. 1 : Relationship between absorbed energy and potential energy distribution

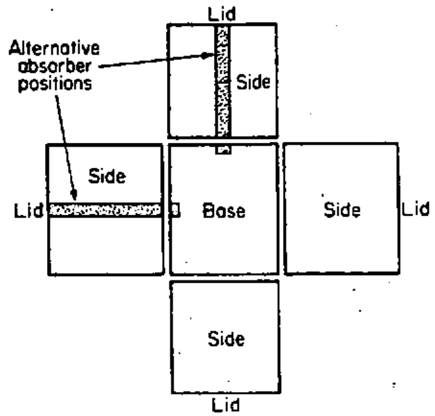

Fig. 2 : Positions of absorber in enclosure

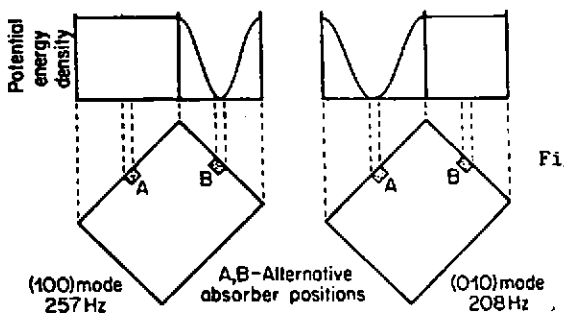

Fig. 3 : Positions of absorber relative to potential energy density distribution

A STUDY OF SOUND ABSORPTION IN A SMALL ENCLOSURE

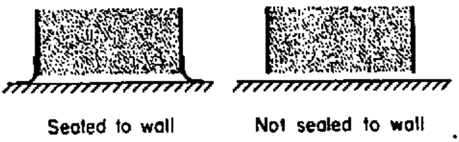

Fig. 4 : Absorber mounting arrangements

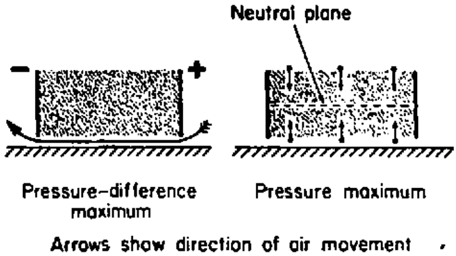

Fig. 5 : Mechanisms of sound absorption

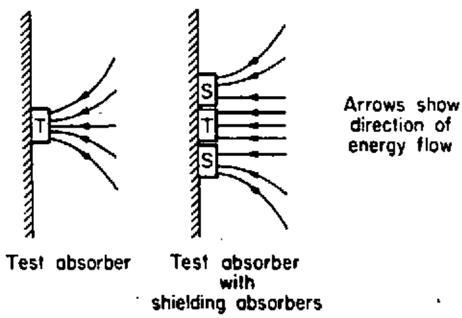

Fig. 6 : Flow of energy into absorber

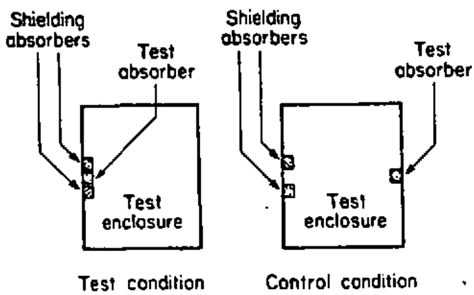

Fig. 7 : Positions of test and shielding absorbers in enclosure