Institute of Acoustics

AN EXAMINATION OF THE PROPERTIES OF FUNCTIONAL ABSORBERS

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1. Introduction

The term "functional absorber" has come to mean the use of panels or sheets of porous material arranged to project from the surfaces of the treated room so that both sides are exposed to the sound field. The technique is now often used for industrial noise control, the absorbing panels being suspended from the ceiling of the treated area : this has the twin advantages of being relatively inexpensive and using an otherwise unoccupied space. To obtain the greatest degree of sound absorption, and therefore noise level reduction, a "saturation" approach is usually adopted, as much treatment as possible being installed to maximize the reduction of sound pressure level (spl). This approach would clearly not be satisfactory if the functional absorber technique was to be applied to the treatment of sound studios, and the present work was undertaken to obtain a more detailed insight into the use of this method of treatment.

It is evident that many variables are involved which could affect the absorption characteristics, among them being panel size, thickness, flow resistance, surface density, pattern of arrangement and spacing from room surface, and room size. In the present investigation, described in detail elsewhere[1], only one size of panel was considered, the principal purpose being to test the effect of the pattern of arrangement of the panels in the room. Measurements were made using the standard "reverberation room" technique, in a room of volume $106m^3$. The area of absorbing material used in the calculation of absorption coefficient from the reverberation times of the empty and treated room was taken simply as the product of the dimensions of one absorbing panel (see below) and the number of panels used, even though both sides of the panel were exposed to the sound field.

The basic absorbing element was a mineral wool blanket 1.2m long, 0.6m wide and 30mm thick, supported round its periphery by a light wooden frame provided with hardboard side-cheeks. Tests were carried out with sixteen such panels arranged in various configurations[1] in the reverberation room, the panels being positioned on a 1.2m grid giving an "absorber density" of one per $1.44m^2$ of surface covered (0.7 per m^2). Additional tests using a smaller number of absorbers were also carried out.

The results of the tests outlined above are described in Sections 2-4 of this paper. Tests were also carried out with the mineral wool blanket supported on each side by a sheet of welded steel grid having a mesh of 12mm (½ inch). It was found that the presence of the steel grid increased (on average) the low and mid-frequency absorption. The results obtained with the steel grids in place are described in Section 6 and discussed in Section 7.

2. General characteristics

Fig. 1 shows the experimental relationship of absorption coefficient as a function of frequency (one-third octave band centres) for two typical absorber arrangements. Eight absorbers were used in one test (see Fig. 2(a), panels labelled "A" removed) and seven in the other (Fig. 2(b), panels labelled "C" removed). These two results illustrate the general sound absorption characteristics of the functional absorbers used in the tests described in this paper. For frequencies up to about 250 Hz, the general trend is for absorption to increase with frequency, but considerable irregular changes of absorption coefficient between adjacent one-third octave bands are also present. The frequencies at which maxima and minima of absorption occur depend on the arrangement of absorbers in the room, but not in any obviously systematic manner. Between about 250 Hz and 1 kHz absorption continues to rise with frequency, but in a more continuous manner without the irregularities present at lower frequencies. Above about 1 kHz the absorption coefficient is effectively independent of frequency.

In this paper no attempt is made to take account of the individual variations of absorption coefficient at low frequencies. For this reason the absorption characteristics are discussed in terms of smoothed curves, so as to show the trend of absorption coefficient with frequency more clearly. These curves include results taken using a number of similar test conditions, not all of which are illustrated in this paper.

3. The effect of absorber position in the reverberation room

Sixteen absorbing panels were arranged in three basic arrangements. In the first case, four groups, each containing four panels, were placed on three walls and the floor of the reverberation room (Fig. 2(a)). In the second case one large group of panels was used, standing on the floor of the room: Fig. 2(b) illustrates one of these arrangements*. In the third case (Fig. 2(c)) two closely-spaced groups were used, also standing on the floor. The smoothed absorption characteristics obtained under these conditons are shown in Fig. 3. It can be seen that for any given frequency the absorption coefficient is reduced for the conditions shown successively in Fig. 2(a), (b) and (c), indicating that mutual screening of one absorbing panel by another is a significant factor. However, the factor by which the absorption coefficient is reduced, on going from one of these conditions to another, is less than the factor by which the absorber density (see Section 1) is increased: thus to obtain the greatest overall absorption in (say) a television studio, it would be appropriate to provide a large number of close-spaced functional panels (i.e. as shown in Fig. 2 (c)) rather than fewer panels more widely spaced.

4. The effect of the number of absorbers in the room

Tests were carried out using the basic panel arrangements shown in Figs. 2(a) and 2(b), but with some of the panels removed. In the case of the panels arranged in four groups (Fig. 2(a)) the panels labelled "A" were first removed to form four groups of two panels each, and then those labelled "B" were also removed so that only one absorber was present on each room surface (no significant difference was found between these two test conditions). In the case of the panels arranged in one large group (Fig. 2(b)), the panels labelled "C" were removed to form a group of seven absorbers.* The smoothed result of these tests are shown in Fig. 4. The principal effect of reducing the number of panels in the room is to increase the value of absorption coefficient, especially at high frequencies. This can be attributed to the mutual screening between panels, as discussed in Section 3. In the case of the panels arranged in groups, no significant change in absorption coefficient with number of panels present was observed below 315 Hz. This may partly be a statistical effect, as the "scatter" in individual absorption coefficient values at any one frequency was considerable (see Section 2). However, it may also indicate that the sound absorption of an individual panel, at these lower frequencies, was more strongly influenced

* Two alternative arrangements were also tried but showed no significant differences.

by its position in the room (i.e. in relation to the room mode pattern) than by the presence of other nearby panels. This may reflect the different mechanisms of sound absorption present at low and high frequencies, as discussed in Section 6.

A further factor which was observed during these tests was the effect of changing the spacing between panels arranged in a single row. Ten panels could be accommodated in the row along the longer floor centre-line, almost touching each other. The absorption coefficient at frequencies of 800 Hz and above, measured under this condition, was significantly lower than when only seven panels were equi-spaced along the same line with a gap between each one. This can be explained in terms of an absorber extracting energy from parts of the sound field which are not immediately in front of it, or, in other words, that the lines of power flow into the absorber are not necessarily normal to its surface[2]. Absorbers placed closer together then have less sound energy available to absorb and the apparent absorption coefficient is therefore reduced.

5. The effect of room volume

Fig. 5 compares results obtained during the present series of tests with those obtained by Cops[3] and Orlowski[4] under similar but not identical conditions. In all cases the absorbing panels were arranged in one large group. In the present results the absorber density was 0.7 absorber per square metre of surface covered. In the case of the other two authors' work the results have been interpolated to the same absorber density using their published data which refers to densities of 0.5, 1.9 and 2.0 absorbers per m^2. These interpolated values have also been smoothed to show general trends more readily, as discussed in Section 2. The absorbing panels used in the three cases were only very approximately comparable, as they differed in terms of their size, thickness and material used for their construction: in fact, Orlowski used a modelling technique on a 16:1 (linear) scale. The most significant difference in the present context, however, is the volume of the reverberation room used for the tests, this being $106m^3$ for the present work, $193m^3$ for the work carried out by Cops, and $1597m^3$ (scaled) for the model work undertaken by Orlowski. It can be seen from Fig. 5 that there is a strong correlation between increase of volume and increase of absorption coefficient for a given frequency : this observation is in agreement with work carried out by Harwood, Randall and Lansdowne[5].

It may also be noted that the results obtained by Cops and Orlowski at three different absorber densities indicate an increase in the absorption coefficient of an absorbing panel as the number of such panels in the room is reduced, in agreement with the results of the present work discussed in Section 4.

6. The effect of increase of surface density

Tests on nine absorber configurations were carried out with the mineral wool absorbing blanket supported with grids of welded steel rods on each side of the blanket, thus increasing its surface density. The tests were repeated without such supports in place. Details of the test conditions are given in the full account of this work[1]. The mean differences in absorption coefficient obtained when the steel grids were present (α_W) and absent (α_O) are plotted in Fig. 6, together with limit lines corresponding to twice the standard error of the mean. These limit lines indicate the range of values of absorption coefficient difference within which there is a 95% probability that the mean value of another nine sets of measurements would fall, assuming that factors such as the size and shape of the absorbers and the geometry of the room in which they are situated are similar. It can be seen from Fig. 6 that for frequencies between 80 Hz and 500 Hz the mean values of absorption coefficient obtained with the steel grids in place were statistically significantly greater, on average, than those formed without using the grids. In making the measurements it was found that arranging that the mineral wool layer was in contact with the supporting grids over its entire surface (for example by pulling the grids together at several points using a fastening passing through the mineral wool) significantly increased the sound absorption. The increase in absorption is roughly 20% of the value obtained without the use of the steel grids and is perhaps not of great significance in acoustic terms: its main interest lies in the insight it gives into the mechanism of sound absorption at these frequencies, as discussed in Section 7.

Measurements were also carried out with and without the steel grid supports with the absorbers lying flat on the floor. In this case there was no significant difference between the results obtained on the two occasions.

7. Discussion

Sound energy in a reverberant enclosure consists of two components: potential energy and kinetic energy. For any one particular room mode, potential sound energy occurs at regions of maximum sound pressure, where sound pressure changes very little with position in the sound field, and the air velocity is low. Kinetic sound energy, on the other hand, occurs at regions of maximum air velocity, where the change in sound pressure with position in the sound field is greatest, while the sound pressure itself is small.

In the case of porous material backed by a hard surface, sound pressure at the free surface gives rise to air movement through the pores, where viscous losses cause sound absorption. Absorption is therefore greatest near a sound pressure maximum[6, 7]. Absorption rises as the sound wavelength decreases in relation to the thickness of material, and porous materials are therefore more absorptive under these conditions at higher frequencies. If both sides are exposed to the sound field, a "neutral plane" of zero sound pressure will form at the midpoint of the material, which will then behave as a panel of half the actual thickness and twice the area. On the other hand, a sheet of porous material placed with its face normal to the velocity vector at a sound pressure minimum, where air velocity is greatest, will absorb sound because of air flow through the material (this assumes that the flow resistance of the material is low enough so as not to be the controlling factor in determining the air velocity). When placed parallel to the velocity vector, however, no air will flow through it and no absorption will occur. These latter effects are likely to be significant at low frequencies, where the absorber thickness is very small compared with sound wavelength.

Bearing the factors discussed above in mind, the effect on the absorption coefficient of a panel of porous material situated in a region of high air velocity, when its surface density is increased by loading it with a relatively massive supporting structure, may be examined by replacing mechanical parameters with electrical equivalents. Fig. 7(a) shows the equivalent circuit of the absorber without extra loading, M_A representing the surface density of the material and R its flow resistance. The air volume velocity U divides into two components U_M and U_R "flowing" though M_A and R respectively as shown in the phasor diagram (Fig. 7(b)), and causing a sound pressure Δp to appear across the absorber. Assuming that the nature of the porous material is such that the impedance of M_A is low compared with that of R at the frequency in question, U_R is small compared with U_M and only a small amount of power is dissipated. When the surface density is increased, however, (Fig. 8(a)), the value of the mass component is increased (shown by the "extra" surface density M_L in series with M_A) and the partition of velocity in the two branches of the circuit changes, so that the component "passing through" R increases (Fig. 8(b)). The power dissipated in R, and, therefore, the absorption coefficient, thus increases. In physical terms, a lightweight porous blanket will tend to vibrate under these conditions, diminishing the velocity of the air passing through it and, therefore, reducing the sound absorption. As the surface density of the blanket is increased, there will be less tendency for it to vibrate, and sound absorption will be enhanced by the resulting increase in the velocity of air passing through it.

In the case of porous material backed by a hard surface, the conditions which exist at a region of maximum sound pressure should be considered. Here (Fig. 9) the sound pressure p forces air into the absorber against the compliance C of the entrapped air. The mass of the porous blanket (M_A) is swamped with the very large mass of the floor itself (M_S): the sound velocity components U_M and U_R are therefore not significantly affected by an increase of surface density (M_L), and no increase in absorption occurs. The same behaviour might be expected to occur at high frequencies with both sides of the material exposed to the sound field, because of the formation of a "neutral plane" of sound pressure as discussed earlier. In this case, however, the effective absorber thickness would be halved, leading to an increase in the frequency at which maximum absorption is achieved. The tests carried out with the absorbing panels lying flat on the floor of the reverberation room confirmed these predictions.

The general behaviour of functional absorbers may be explained in terms of different mechanisms of sound absorption occurring in the three frequency bands discussed in Section 2. Below about 250 Hz the absorbers appear to be sensitive, at least to some extent, to the kinetic component of the sound field, as shown by the increase in absorption which is obtained when the strengthening panels of steel rods are added. This accounts for the extreme variability of the absorption coefficient values obtained in individual frequency bands using different absorber configurations, as these configurations are likely to react in different ways to the mode structure present in the reverberation room. Above 1 kHz the absorbers are sensitive to the potential component of the sound field, behaving as porous panels of half the thickness and twice the surface area (compared with their actual dimensions) backed by a massive rigid surface. The two octaves between 250 Hz and 1 kHz may be regarded as a transitional region between these two modes of operation.

8. Conclusions

Functional absorbers show, in general, a rising characteristic of absorption coefficient against frequency up to a certain value (about 1 kHz in the case of the panels used in the present tests), above which the absorption coefficient remains constant. For frequencies below about 250 Hz a significant proportion of the overall sound absorption appears to be due to the velocity (kinetic) component of the sound field. In this case, increasing the surface density of the absorber without altering its thickness or porosity gives rise to an increase of absorption coefficient. For frequencies above about 1 kHz absorption is due mainly to the potential (pressure) component of the sound field and the absorber behaves as if it had twice its actual surface area and half its thickness, backed by a massive rigid surface. The frequency range 250 Hz - 1 kHz may be regarded as a region of transition between these two mechanisms of sound absorption.

The absorption coefficient of an individual absorbing panel depends considerably on the arrangement of the absorbers in the room: in general, the absorption coefficient rises as the spacing between the panels is increased. There is some evidence to suggest, in addition, that for a given absorber size and spacing, the absorption coefficient of an individual absorber increases with the volume of the treated room.

9. Acknowledgments

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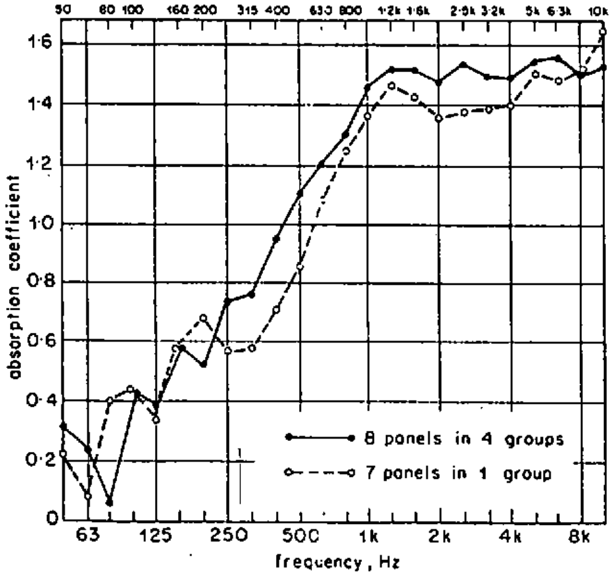

Fig. 1 - Typical absorption characteristics obtained using the functional absorber technique.

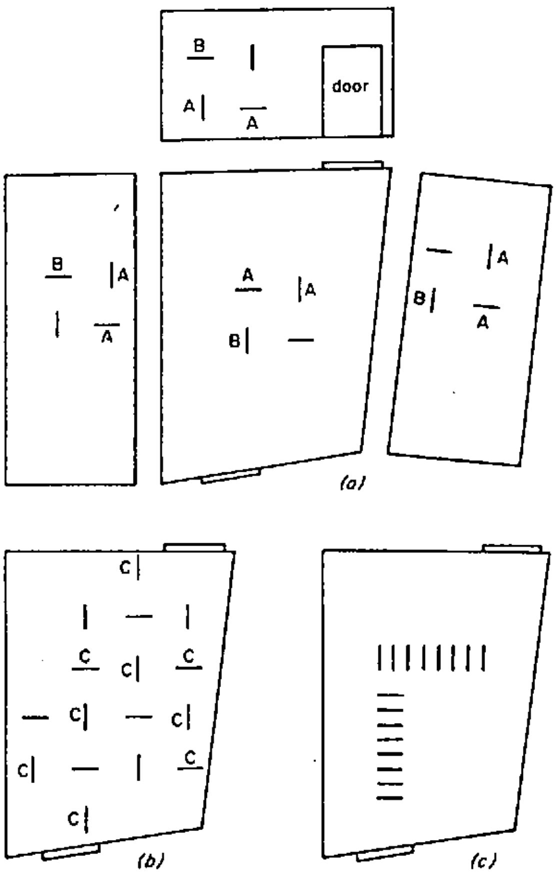

Fig. 2 - Arrangements of absorbers in reverberation room.

(a) Four groups on three walls and floor
(b) One group on floor
(c) Two close-spaced groups on floor

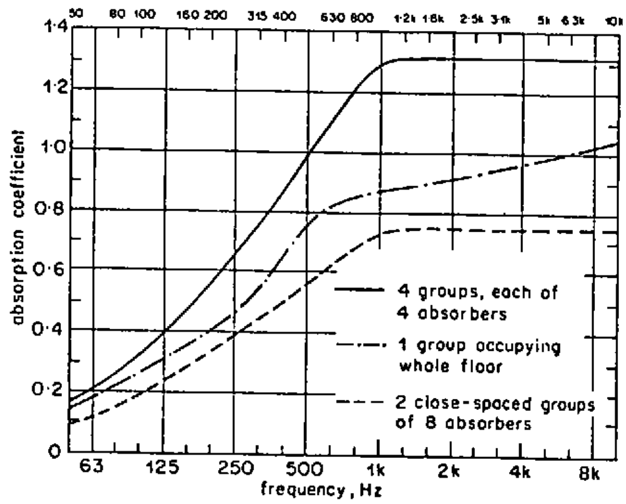

Fig. 3 - Comparison of smoothed absorption characteristics using 16 absorbing panels.

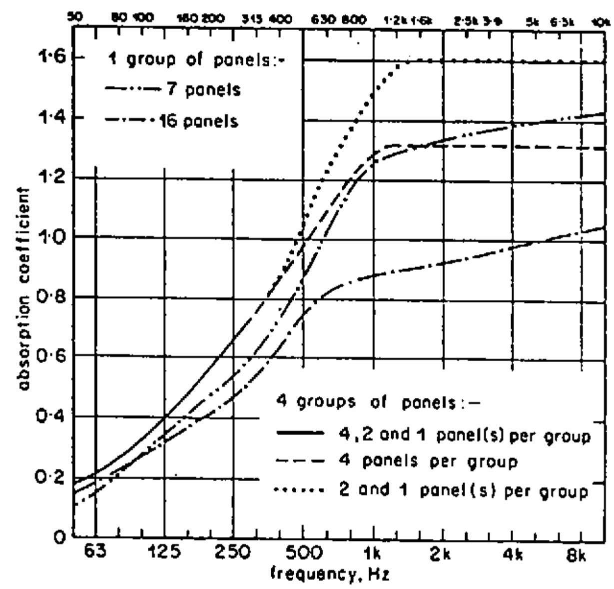

Fig. 4 - Smoothed absorption characteristics showing effect of number of panels.

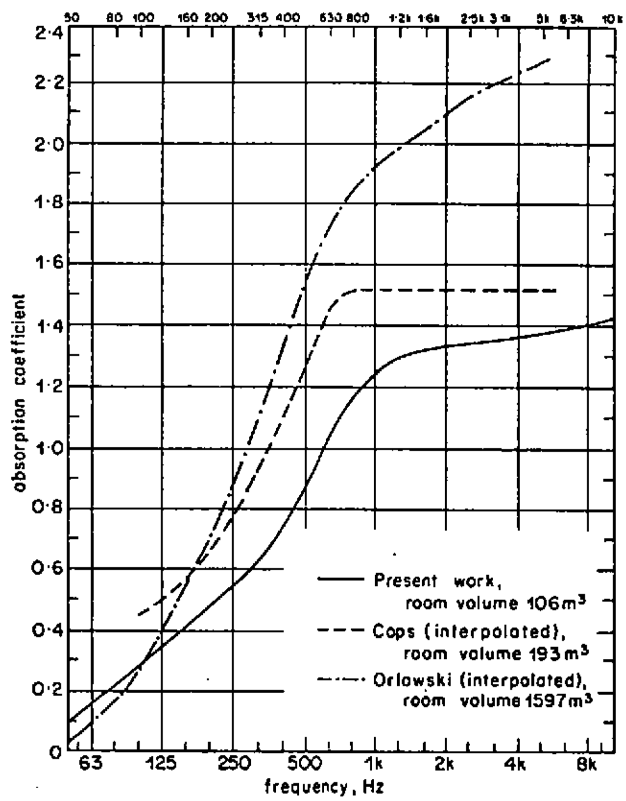

Fig. 5 - Comparison of present results with those obtained by Cops and Orlowski.

Absorber density : 0.7 absorber per m² of surface covered

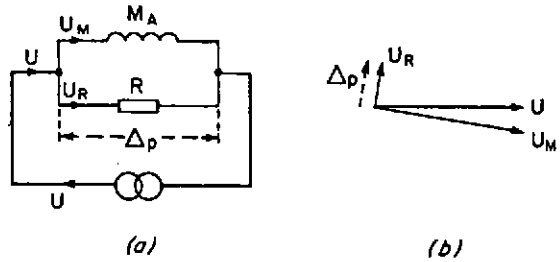

Fig. 7 - Electrical analogue of porous absorber in kinetic sound field

(a) Equivalent circuit
(b) Phasor diagram

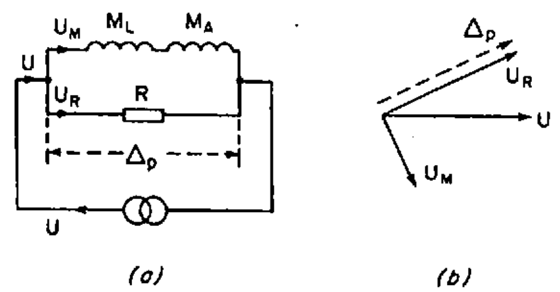

Fig. 8 - Effect of increasing surface density of porous absorber in kinetic sound field

(a) Equivalent circuit
(b) Phasor diagram

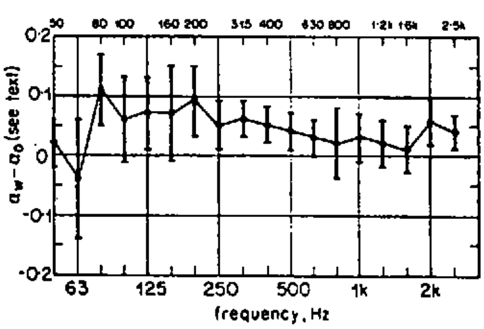

Fig. 6 - Mean change in absorption coefficient caused by presence of steel grid supports

Bar lines show ± twice standard error of mean values.

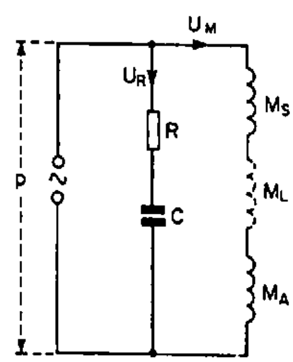

Fig. 9 - Electrical analogue of porous absorber in potential sound field, backed by hard surface