The design of a new wideband modular sound absorber

R. Walker, B.Sc.(Eng.)
P.H.C. Legate
THE DESIGN OF A NEW WIDEBAND MODULAR SOUND ABSORBER

R. Walker, B.Sc.(Eng.), P.H.C. Legate

SUMMARY

The use is discussed of a modular system of acoustic treatment for use in a wide range of types of studios, control rooms and other areas in which the control of the acoustic environment is essential. Although the system at present in use is generally satisfactory, several defects are identified. The design of a new module intended to absorb sound over the entire audio frequency range and to overcome these defects is discussed and the results of some measurements are presented.

The final design for the new module is described and its performance is compared with that of the combination of the earlier modules most commonly used in small and medium sized rooms. The proposed new module is far from ideal and will probably prove to be more expensive to manufacture. However, it does offer significant advantages in some circumstances. Also, because fewer of the new modules would be required to achieve a given result, the total cost of using the new module may not be very different from that of using the older modules.
THE DESIGN OF A NEW WIDEBAND MODULAR SOUND ABSORBER

<table>
<thead>
<tr>
<th>Section</th>
<th>Title</th>
<th>Page</th>
</tr>
</thead>
<tbody>
<tr>
<td>1.</td>
<td>Introduction</td>
<td>1</td>
</tr>
<tr>
<td>2.</td>
<td>Design principles of the new module</td>
<td>2</td>
</tr>
<tr>
<td>2.1.</td>
<td>Size</td>
<td>2</td>
</tr>
<tr>
<td>2.2.</td>
<td>Required performance</td>
<td>2</td>
</tr>
<tr>
<td>2.3.</td>
<td>Method of achieving the required performance</td>
<td>2</td>
</tr>
<tr>
<td>3.</td>
<td>Theoretical considerations</td>
<td>3</td>
</tr>
<tr>
<td>3.1.</td>
<td>General</td>
<td>3</td>
</tr>
<tr>
<td>3.2.</td>
<td>Low-frequency equivalent circuit</td>
<td>4</td>
</tr>
<tr>
<td>3.3.</td>
<td>Methods of increasing the loss resistance</td>
<td>4</td>
</tr>
<tr>
<td>3.4.</td>
<td>High frequency equivalent circuit</td>
<td>5</td>
</tr>
<tr>
<td>4.</td>
<td>Experimental results</td>
<td>6</td>
</tr>
<tr>
<td>4.1.</td>
<td>The measurement of absorption coefficient</td>
<td>6</td>
</tr>
<tr>
<td>4.2.</td>
<td>Measurements on standard A2 and A3 modules</td>
<td>6</td>
</tr>
<tr>
<td>4.3.</td>
<td>Confirmation of the proposed low frequency component</td>
<td>6</td>
</tr>
<tr>
<td>4.4.</td>
<td>Measurements on the first prototype</td>
<td>6</td>
</tr>
<tr>
<td>4.5.</td>
<td>Measurements with additional low frequency damping</td>
<td>7</td>
</tr>
<tr>
<td>4.6.</td>
<td>Effect of reducing the mineral wool layer thickness</td>
<td>8</td>
</tr>
<tr>
<td>5.</td>
<td>Discussion of results</td>
<td>8</td>
</tr>
<tr>
<td>6.</td>
<td>Conclusions</td>
<td>9</td>
</tr>
<tr>
<td>7.</td>
<td>References</td>
<td>9</td>
</tr>
<tr>
<td>Appendix 1</td>
<td>Calculations of component values</td>
<td>10</td>
</tr>
<tr>
<td>Appendix 2</td>
<td>Calculation of damping coefficients</td>
<td>11</td>
</tr>
</tbody>
</table>
THE DESIGN OF A NEW WIDEBAND MODULAR SOUND ABSORBER

R. Walker, B.Sc.(Eng.), P.H.C. Legate

1. Introduction

In broadcasting studios and control rooms (cubicles), materials which absorb sound energy must be added to the otherwise adequate interior of the room in order to control the acoustic environment. Such material, whose prime or only function is to absorb sound energy, is usually known by the generic description “acoustic treatment”. In order to minimise the quantity of acoustic treatment which must be installed, one of the main characteristics of “good” acoustic treatment is that the coefficient of sound energy absorption should be as high as possible within the working frequency range. Values close to or even apparently exceeding unity are common. Although many materials satisfy this requirement, it has been common practice in the BBC, since 1968 when the components were first designed, to use a modular system of acoustic treatment. This system comprises a number of different standard modules in three different sizes giving a range of absorption characteristics from which suitable choices can be made for the treatment of individual studios and control rooms. A complete description of the different modules is given in Ref. 1.

There are three main advantages of fixed module sizes. The first is that the architect designing the room can provide the mounting frames before the detail of the treatment is finalised. The second is that the treatment can be prefabricated, usually at a lower cost by a subcontractor away from the building site, thus saving both time and money. The third advantage is that subsequent changes, either immediately after construction if the treatment was initially incorrect or during later refurbishment, can easily be made.

In practice, for the acoustic treatment of the majority of small and medium sized rooms, two modules only are in common use. These are the “A3” and the “A2” modules which have 580 mm square fronts and are 184 mm deep. The A2 module is a relatively narrow-band, low-frequency sound absorber which uses the front-panel as an acoustic mass element resonating with the compliance of the airspace enclosed behind. The A3 module consists essentially of a mineral-wool layer in front of an airspace, the resulting absorption coefficient is high for all frequencies above about 315 Hz. To compensate for the increased absorption of the air above about 4 kHz and the almost unavoidable presence of fabrics and other materials which have a rising absorption characteristic above 1 or 2 kHz, the A3 module has a low-pass filter in the form of a perforated hardboard front face with about 20% open to total area ratio. These two modules are complementary to each other and when used in equal numbers provide a high and reasonably uniform absorption coefficient over a significant part of the audio frequency range. Together they provide a sufficiently flexible method of providing acoustic treatment for a wide range of room types.

However, despite the fact that these two modules have been used together for many years, they have three main defects. The first is that the two modules are not truly complementary and together they have an excess of sound absorption around 400 Hz. This gives rise to lower reverberation times around 400 Hz than at higher and lower frequencies in areas treated with these two modules.

The second defect is that as each of the modules is effective only over part of the audio frequency range the total surface area of treatment required is about twice that which would be required if one module was effective over the whole audio frequency range. In some small areas it is difficult to find sufficient free wall or ceiling space for the required quantity of acoustic treatment. The third defect is that the type A2 absorber, because it is effective only at low frequencies, is an efficient reflector of high-frequency sound energy. The front surface of the A2 is essentially unperforated and flat; the resulting high frequency reflections are specular rather than diffuse. This causes great difficulties, especially in control rooms equipped for stereo where the flat surfaces of the observation window, equipment bays, control desk, free wall surface as well as the front panels of the low-frequency acoustic treatment, all combine to produce a great deal of specular reflection.

It was therefore thought that the development of a new type of A-size module was justifiable. This new module should combine the best features of both the A2 and A3 modules, but without the excessive absorption around 400 Hz. Because it will be an efficient sound absorber at high fre-
quencies, it will inevitably have a non-reflecting front surface.

2. Design principles of the new module

2.1. Size

Experience has shown that the A-size module is a satisfactory and convenient size for the acoustic treatment of a large range of relatively small and medium sized rooms. Larger modules, whilst effective and economic for large rooms are difficult to fit into the available spaces in smaller rooms without needing a large proportion of individual, specially designed units to fill in the small spaces. Modules much smaller than the A-size are uneconomic because larger numbers would be required, thus increasing the manufacturing and fitting costs per unit area of acoustic treatment.

One minor difficulty with the A-size module in very small rooms is the depth, which at 184 mm can reduce the available floor space significantly. In this respect, the B-size module, which is approximately 130 mm deep, is better. However, it was thought to be unrealistic to expect to obtain from the new absorber the desired absorption coefficient characteristic using a unit only 130 mm deep.

The external size of the new module was thus determined to be 580 mm square and 184 mm deep.

2.2. Required performance

Fig. 1 shows the measured absorption coefficient characteristics for typical sets of A2 and A3 modules. Also shown in Fig. 1 is the characteristic (c) which is the arithmetic average of the two measured characteristics (a) and (b). This would be the average absorption characteristic for an area of acoustic treatment comprising equal proportions of A2 and A3 modules. In such a combination, because each module is fully absorbent over less than the whole audio frequency range, the general level of the characteristic is lower than it would be if all of the modules were effective over the whole audio frequency range. Therefore, the target characteristic for the new module should be as Fig. 1(c) but raised in level, ideally by a factor approaching 2:1. Because of incidental structural and surface absorptions, the A2 and A3 modules do not have absorption coefficients of zero outside their intended working ranges. Thus, in practice, a factor of 2:1 will not be achieved.

2.3. Method of achieving the required performance

The design of acoustic treatment which is effective at medium and high audio frequencies is comparatively simple. A suitable thickness of mineral wool or other acoustically resistive element of the appropriate density either with or without a backing airspace, is generally satisfactory. With normal materials, the total thickness of the absorbing layer and the backing airspace sets the lowest frequency at which high values of absorption coefficient are obtained. In contrast, the provision of highly effective low-frequency absorption in a compact form is more difficult. Accordingly, the first decision to be made in the design of a new module was the low-frequency absorption mechanism. Apart from the obviously impracticable use of a mineral wool layer with a very deep airspace ($\approx$ 1 m total) which would, in any case, conflict with the size restrictions of Section 2.2, two principal methods of achieving high absorption coefficients at low frequencies are available. One of these is the use of Helmholtz resonators. Although one studio in the BBC was successfully treated with such resonators$^{3,4,5}$, in general, because they are effective only over narrow frequency ranges, several different types are required even to cover the octave of frequency from 100 Hz to 200 Hz. It may also be necessary to tune the absorbers individually after installation. In contrast, membrane low frequency absorbers are usually effective over a wider frequency range and are relatively much less critical in construction. They also have a long history of successful application in the acoustic treatment of broadcasting studios and control rooms$^{6}$.

A membrane absorber consists of a flat panel enclosing an airspace. The superficial mass of the panel and the compliance of the enclosed airspace

---

*Fig. 1—Absorption characteristics of current "A2" and "A3" modules*
Form a system resonating at the frequency at which the absorber is designed to have a high absorption coefficient. By a suitable choice of damping coefficient, the absorption can be made high over a range of frequencies. It was proposed that the new absorber should consist of a standard A-size module with an internal resonant panel recessed by a sufficient distance from the front. The depth of the recess should be sufficient to accommodate a mineral-wool layer thick enough to give a lower cut-off frequency of absorption coefficient which, when added to the absorption of the resonant panel, would give a reasonably uniform absorption coefficient over the whole audio frequency range. A conflict arises between the spaces behind and in front of the resonant panel. The enclosed space behind the panel should be large (and the mass of the panel low) in order to minimise the acoustic impedance at resonance. The space in front of the panel, filled with mineral wool, is also required to be large as stated above. The final compromise chosen for the experimental tests is shown in Fig. 2. It is similar in principle to the absorber type C21, but only 184 mm deep instead of 215 mm and like the A3 module, has a perforated front cover (20% open to total area ratio) to limit the extreme high-frequency absorption.

3. Theoretical considerations

3.1. General

It is often convenient and helpful to represent an arrangement of acoustic components as an electrical analogue. The network-solving techniques which are a familiar part of electrical circuit theory can then be applied to the equivalent of the acoustic

---

**Fig. 2**—Part section of proposed new modular absorber (all dimensions in mm)

**Fig. 3**—Electrical analogue of proposed new module, showing all main components
network representing the proposed acoustic absorber. Many sets of equivalence relationships are possible; the one which will be used here equates electrical current to acoustic volume velocity and electrical voltage to acoustic pressure. With this convention, mass is represented by inductance, compliance by capacitance and energy loss by resistance. Fig. 3 shows a simplified equivalent circuit of the proposed new absorber module. At high frequencies, that is where the absorber dimensions are comparable with or larger than the wavelength of the incident sound energy, this model of the complete absorber is only valid for normally incident sound energy. The components shown represent the principal features of the absorber; all of the stray and second order components except the side panel resonances have been neglected. \( M_{P1} \) and \( M_{H1} \) represent the mass of the front panel and the holes therein; \( R_{H1} \) represents the loss resistance of the front panel holes. \( R_{MW} \) represents the loss resistance of mineral wool layer. \( M_{p2} \) represents the mass of the internal, unperforated panel and \( C_A \) represents the compliance of the sealed airspace enclosed behind the internal panel. \( R_{p2} \) represents the inherent total loss of the resonant system formed by \( M_{p2} \) and \( C_A \); it has been shown in that position purely for convenience and will later be neglected. \( v \) represents the incident acoustic energy and \( Z_R(= R_r + jX_r) \) the complex radiation impedance. \( Z_{sp} \) represents the complex impedance of the most significant stray component, the resonance of the side panels from which the box is made up.

### 3.2. Low-frequency equivalent circuit

Fig. 4 shows a simplified equivalent circuit for frequencies near to the resonant frequency of \( M_{p2} \) and \( C_A \).

All of the mass components including the reactive part of the radiation impedance have been lumped together and are represented by \( M_{p2} \). Appendix 1 gives the calculations for each component and the simplifications. The resonant frequency of the equivalent circuit of Fig. 4 is 98.7 Hz. (Appendix 1). At this frequency, all of the reactive components cancel each other leaving only the loss and radiation resistances. The absorption coefficient can be defined as the ratio of the acoustic power absorbed to the acoustic power which would be absorbed by an ideal absorber, that is, one which is matched for maximum power transfer. Using the terminology of Fig. 4, the absorption coefficient, \( \alpha \), can be expressed at the resonant frequency, by

\[
\alpha = \frac{4 \cdot R_r \cdot R_{p2}}{(R_r + R_{p2})^2}
\]

\[ \cdots (1) \]

The numerical value of \( R_{p2} \) is equal to \((1/Q) \times (\text{reactive impedance of } M_{p2} \text{ or } C_A \text{ at resonance})\) where \( Q \) is the usual “quality factor” for a resonant circuit. With the normal methods of construction, that is a sealed box of 9 mm plywood sides, a 6 mm hardboard rear panel and a 3 mm hardboard front panel, the value of \( Q \) is 30 – 40. Therefore,

\[
6480/40 \leq R_{p2} \leq 6480/30
\]

or

\[
162 \leq R_{p2} \leq 216
\]

Substituting these values in Equation 1 gives an absorption coefficient at the resonant frequency of 0.965 to 0.998.

This shows that the basic structure is slightly underdamped and also that the absorption coefficient at the resonant frequency is not strongly dependent on the value of the damping coefficient. Apart from slightly increasing the peak absorption coefficient there is a second reason for introducing additional damping (lowering the \( Q \)). With high values of \( Q \), the frequency range over which the absorption coefficient has a high value is comparatively narrow. Lower values of \( Q \) would increase the effective bandwidth without catastrophically reducing the maximum absorption coefficient.

### 3.3. Methods of increasing the loss resistance

A number of methods of increasing the loss resistance \( R_{p2} \) to lower the \( Q \) are possible. The most obvious method is to use a material for the front panel which has a higher inherent loss than the 3 mm hardboard. This method has been employed in the past using bitumen roofing felt either alone or as part of a composite, bonded to a sheet of hardboard. An alternative method using the same principle is found in a commercial, low-frequency absorber available at the time of writing, using a modern high-loss polymer sheet. The "roofing felt" modules were sometimes satisfactory, but frequently suffered from inconsistent bonding and long-term changes in the material characteristics. The modern,
polymer types were unknown at the time of the experimental work described in this report.

A second method of increasing the loss-resistance which has also been used in the past is to fix a sheet of acoustically resistive material (such as mineral wool) close to, but not touching, the vibrating front panel. Using this method, it is difficult accurately to maintain the very close spacing required for a significant damping effect, especially with non-rigid materials like mineral wool.

The method which was finally devised and used for the experimental work is illustrated by the low-frequency equivalent circuit shown in Fig. 5. It is

![Fig. 5—Low frequency electrical analogue, showing additional damping components](image)

![Fig. 6—Electrical analogue of proposed new module, simplified for high frequencies](image)

The modification shown in Fig. 5 to the basic circuit of Fig. 4 can be implemented by a hole of a correct size to give the required mass component, covered by a layer of an appropriate fabric to give the required resistance component. This hole may be placed anywhere on the surface of the box provided that it communicates with the airspace enclosed behind the internal panel. It is most conveniently drilled in the side panel. For a 9 mm thick side panel, a mass component equivalent to 4.5 times the mass of the internal panel is provided by a circular hole, 38 mm in diameter. Two such holes give a mass component equal to 2.25 times the panel mass (Appendix 1).

### 3.4. High frequency equivalent circuit

The complete equivalent circuit shown in Fig. 3 can be simplified at high frequencies to that shown in Fig. 6. At frequencies above about 400 Hz, the mass component of the radiation impedance is negligibly small. Also $M_{P1} >> M_{H1}$ and $M_{P1}$ can, therefore, also be neglected. The mass reactance of the internal panel $M_{P2}$ is also very large so that it and the compliance of the enclosed airspace, $C_A$, can be omitted. Also, the numerical value of $R_{H1}$ is small in comparison with the remaining circuit impedances at all frequencies.

At high frequencies, the mineral wool layer is thick enough to behave as a distributed network which can only be analysed using transmission line theory. In Appendix 3, an expression is derived for the input impedance, $Z_I$, of this transmission line.

$$Z_I = Z_0 \cdot \frac{\sinh 2al - j \sin 2bl}{\cosh 2al - \cos 2bl}$$

where,

$$Z_0 = (L/C - jR/\omega C)^{1/2}$$

$$a = (((\omega^2 LC)^2 + (\omega CR)^2)^{1/2} - \omega^2 LC)/2)^{1/2}$$

$$b = (((\omega^2 LC)^2 + (\omega CR)^2)^{1/2} + \omega^2 LC)/2)^{1/2}$$
and \( R, L \) and \( C \) are the mineral wool parameters expressed as resistance, mass and compliance per unit length. \( \omega \) is the angular frequency of the incident sound wave, and \( l \) is the length of the transmission line, i.e. the thickness of the mineral wool.

Using these expressions for the input impedance of the transmission line, the normal incidence absorption coefficient for the circuit of Fig. 6 can be calculated at a number of different frequencies. The results of these calculations for different thicknesses of mineral wool are given in Appendix 3. Also given is a comparison between the theoretical and measured results.

4. Experimental results

4.1. The measurement of absorption coefficient

The only practicable method of measuring the absorption coefficient of materials under realistic conditions is to compare the rates of sound energy decay (reverberation times) in a room before and after the installation of the test sample. For the greatest accuracy, the absorption of the empty room should be low and the quantity of treatment installed should be large enough to produce a significant change in the average absorption coefficient of the wall surfaces. Also, to produce results which are representative of the type of application of the acoustic treatment, the sound energy field should be diffuse. At each frequency, the absorption coefficient can be calculated from the reverberation times of the room with and without the acoustic treatment together with the room and sample dimensions. In the BBC Research Department, measurements are normally made in a reverberation room of approximately 100 m\(^3\) which has an "empty" reverberation time of about two to three seconds. The test sample area is about 10 m\(^2\) and is divided into four approximately equal areas placed on three walls and the floor of the room. For the measurement of the A-size modular absorbers, this requires 24 modules split into four groups of six modules each. This requirement for 24 modules was a constraint which, because of the work involved in manufacture and mounting, set a practical limit to the number of modifications which could be measured.

4.2. Measurements on standard A2 and A3 modules

As a point of reference and because the modular absorbers are not exactly reproducible, the performance of a set of standard modules was measured twice. For the first measurement, the set of boxes was constructed as A3 modules with high density mineral wool and 20\% open area front covers. For the second measurement, the same set of boxes was constructed as A2 modules with low density mineral wool and 0.5\% open-area front covers. The results of these measurements are shown in Fig. 1 and are representative of the normal performance of A2 and A3 modules.

4.3. Confirmation of the proposed low frequency component

Fig. 1 shows that the peak absorption coefficient for an A2 modular absorber which is 184 mm deep occurs at a frequency of 100–125 Hz. In Section 3.2 it was shown that an absorber with an enclosed airspace 110 mm deep and an unperforated front cover should have a resonant frequency of about 99 Hz, the lower airspace compliance being offset by the increased front panel mass. A set of modular absorbers with a recessed, unperforated front panel of 3 mm thick hardboard and enclosed airspace 110 mm deep was constructed. Fig. 7 shows the measured absorption coefficient of this set with no mineral wool layers. It shows that the peak absorption coefficient occurred at a frequency of 100–125 Hz and that the general shape of the characteristic was similar to that of the A2 (Fig. 1(a)) at low frequencies. The absolute level of the peak was lower because no additional damping had been introduced at that stage and the inherent damping coefficient was too small to give the maximum absorption.

4.4. Measurements on the first prototype new absorber

A set of modular absorbers was constructed, using the same boxes as in Sections 4.1 to 4.3 above,
as shown in Fig. 2. The mineral wool layer shown in Fig. 2 consisted of one layer of 30 mm thick high density (\(\approx 150 \text{ kg/m}^3\)) material and one layer of 30 mm thick low density (\(\approx 50 \text{ kg/m}^3\)) material with the higher density material nearer to the front panel. Measurements were made of the absorption coefficients as functions of frequency of this set of modules both with and without the 20% perforated front cover. Fig. 8 shows the results obtained. At frequencies above about 250 Hz the performance of the modules with the perforated front cover (Fig. 8(b)), is similar to that of the A3 modules shown in Fig. 1. At lower frequencies, the absorption coefficient was significantly less than the target given in Section 2.2.

4.5. Measurements with additional low frequency damping

Fig. 8 shows that the low frequency performance of the prototype modules was inadequate, probably due to the low inherent damping coefficient at and near to the frequency of the internal panel resonance. The damping was increased as proposed in Section 3.3 above, first by drilling one 38 mm diameter hole in the side panels of the boxes, between the enclosed rear airspace and the outside. Fig. 9(b) shows the measured performance of this set of modules, compared with that of the undrilled modules, Fig. 9(a). Although the difference between these two sets of results at frequencies around 100 Hz is of the same order as the experimental errors, there is slight evidence of an increase in both the peak low frequency absorption coefficient and in the frequency at which this peak occurs. Both of these effects were predicted theoretically in Section 3.4.

Further measurements were made with two 38 mm diameter holes drilled in the side panels of the boxes. For these measurements, additional damping of the low frequency resonance was added by one layer of lightweight fabric (10 rayls flow

---

**Fig. 8**—Absorption characteristic of proposed new module showing the effect of the perforated front panel

**Fig. 9**—Absorption characteristic of proposed new module, showing the effect of increased damping of the internal panel

**Fig. 10**—Absorption characteristic of proposed new module, showing the effect of further increased damping of the internal panel
resistance. The results are shown in Fig. 10. In changing from no additional damping, Fig. 10(a), to the same modules with the addition of two 38 mm diameter holes in the side panel, Fig. 10(b), the peak low-frequency absorption coefficient was increased significantly. However, the frequency range over which this high value of absorption coefficient applied was still very narrow. Adding a layer of fabric of 10 rays flow resistance, Fig. 10(c), resulted in a lower peak level, corresponding to greater than optimum damping. The increase in the frequency range over which this high value applied was not as large as was expected.

4.6. Effect of reducing the mineral wool layer thickness

One measurement was carried out on a set of modular absorbers with only 30 mm thick, high density mineral wool instead of the 60 mm thick layer of high and low densities used for the previous tests. For this test the additional damping consisting of two 38 mm diameter holes with two layers of fabric over each was also included. Fig. 11 shows the results obtained.

It is evident that the lowest frequency at which the mineral wool layer had a high absorption coefficient was too high. This resulted in a range of frequencies around 250 Hz over which neither the low frequency panel nor the mineral wool layer was absorbing the sound energy effectively.

5. Discussion of results

Fig. 12(a) shows the measured performance of the proposed new modular absorber. The module which gave these results is shown in Fig. 2 and comprises a standard A-size box with an internal unperforated panel of 3 mm hardboard, spaced so as to enclose an airspace 110 mm deep. This internal panel is supported in position by being lightly pinned to an internal beading which is continuous around the perimeter of the panel and which is glued and pinned to the internal side surfaces of the box. The space in front of this panel (62 mm deep) is filled with two layers of 30 mm thick mineral wool, the front one being high-density (≥150 kg m⁻³) and the second one low-density (≥50 kg m⁻³). The whole assembly is enclosed by a perforated 3 mm hardboard front panel of 20% open-area ratio and a 6 mm hardboard rear panel. Two holes of 38 mm diameter each were drilled in the side panels in such a position as to join the enclosed rear airspace and the outside of the module. Each of these holes was covered by two layers of fabric with a flow resistance of 10 rays. Fig. 12(b) shows the absorption coefficient characteristic which would be obtained if an area of treatment made up of equal areas of A2 and A3 modules were to be measured under the same conditions and the results multiplied by 1.6. It was obtained by taking the arithmetic average of the two characteristics of Fig. 1(a) and Fig. 1(b) in the same way as for Fig. 1(c) and multiplying the result at each frequency by a factor of 1.6. Comparison of Figs. 12(a) and 12(b) shows that, at all frequencies except 50 Hz, an area of the proposed new modular absorber absorbs more sound energy than the same total area of treatment made up from equal numbers of A2 and A3 modules. For frequencies above about 500 Hz, the absorption coefficient characteristic of the proposed new modular absorber is similar in shape and approximately 1.6 times larger at all frequencies than that for the combined A2/A3 treatment.
At lower frequencies, the proposed new module shows an irregular absorption coefficient characteristic with a severe relative loss in performance around 160–200 Hz compared with the A2/A3 modules.

Fig. 11 shows that with only one layer of 30 mm thick high-density mineral wool the loss of performance extends to about 300 Hz and is much worse in both level and in the frequency range over which it occurs.

It is evident that a 30 mm thick layer of mineral wool is not sufficient to provide a wide frequency range over which the absorption coefficient has a high value and that even 60 mm is insufficient. Appendix 3 shows that little additional benefit would be obtained at frequencies around 160–200 Hz by increasing the thickness to 90 mm. In fact, the theoretical results indicated that the performance would be slightly worse in this frequency range with a 90 mm thick mineral wool layer.

Fig. 13—Calculated reverberation time of small studio

---

The assumptions made were that the walls of the studio were constructed of plastered and pointed 225 mm brickwork, the floor was of timber boards on joists with a carpet tile covering and the ceiling was of plastered wood-wool slab. The structural absorption was small in comparison with that of the carpet and the acoustic treatment. When treated with sufficient new modules (30 m²) to reduce the mid and high frequency reverberation time to about 0.30 secs, the reverberation time characteristic of Fig. 13(a) was obtained. The increase in reverberation time at 160 Hz is a direct result of the low efficiency of the new module at this frequency. This irregular characteristic at low frequencies would probably be unacceptable. By removing some of the new modules and replacing them by a greater quantity of A2 modules, the irregularity of the characteristic can be reduced at the cost of a slight increase in the total quantity of acoustic treatment required. The characteristic shown in Fig. 13(b) was obtained by “installing” 25 m² of the new module and 8 m² of standard A2 modules. The irregularity of the characteristic has been reduced and the result would probably be acceptable. To achieve the same results with A2 and A3 modules would require the installation of approximately 45–50 m² of acoustic treatment.

6. Conclusions

The range of the system of standard size modules for the acoustic treatment of studios and control rooms has been extended by the design of a new A-size module. Although the new module is far from being perfect, it offers a method of reducing significantly the total quantity of acoustic treatment required to obtain a given result; in a small talks studio a saving of over 30%, in area can be expected. It also offers the very important potential advantage of significantly reducing the problems caused by the reflections from the front surfaces of earlier types of low frequency acoustic treatment.

The construction of the new module is more complex than previous modules, but it contains no critical features, such as sensitive glue joints or materials, which are difficult to control in production. It may be more expensive to use than an equivalent area of acoustic treatment using A2 and A3 modules, although the total costs are beyond the scope of this work, requiring information on the costs of installation as well as module unit costs.

If the advantages offered are considered to be of sufficient value then a full-scale trial including a total costing, will have to be carried out in a practical working environment before the new module can be adopted for general use.

7. References


**Appendix 1**

**Calculations of component values**

For the circuit shown in Fig. 3:

1. Acoustic mass of the air contained in one hole, $M$, is given by:

$$M = (\rho_o/\pi a^2)(t + 1.7a(1 - a/b))$$

where $\rho_o =$ standard air density (1.18 kg/m³)
\[a = \text{hole radius (1.5 \times 10^{-3} m)}\]
\[t = \text{panel thickness (3.2 \times 10^{-3} m)}\]
\[b = \text{hole spacing (6 \times 10^{-3} m)}\]

Therefore, $M = 853.5$ kg m⁻⁴ for one hole.

The panel has 8100 holes, therefore,

$$M_{H1} = 0.105 \text{ kg m}^{-4}$$

2. Acoustic resistance of a hole, $R$, is given by:

$$R = \sqrt{\omega}.(\rho_o/\pi a^2)(\sqrt{2\mu.(t/a)} + 2(1 - A_h/A_b))$$

where
\[\omega = \text{the angular frequency of the sound energy}\]
\[\mu = \text{the kinematic viscosity of air (1.56 \times 10^{-5} m^2 s^{-1})}\]
\[A_h = \text{the area of one hole}\]

3. $A_b =$ the area of the panel around one hole

$$R_{H1} = 3737 \sqrt{\omega}/8100 = 0.461 \sqrt{\omega} \text{ ohms}$$

3. Acoustic mass of internal panel

$$M_{p2} = \text{mass per m}^2/(\text{area})^2$$

therefore,

$$M_{p2} = 1.07/(0.58)^4 = 9.51 \text{ kg m}^{-4}$$

4. Acoustic mass of front panel, $M_{p1} = M_{p2} \times 0.8$ for 20% perforation

therefore,

$$M_{p1} = 7.608 \text{ kg m}^{-4}$$

This is very much larger than the parallel mass of the holes, $M_{H1}$, and can therefore be neglected.

5. Compliance = volume $/\rho_o \cdot c^2$ if the dimension in the direction of propagation is small in comparison with wavelength of the sound energy.

$$\text{Volume} = (0.110)(0.58 - 0.018)^2 \text{ for 110 mm deep airspace, external width and height of 0.58 m and 9 mm side panel thickness.}$$

Therefore

$$C_A = 2.488 \times 10^{-7} \text{ m}^5/\text{N}$$

6. The resonant frequency, $f_R$, of the internal panel mass and the enclosed airspace compliance is given by

$$f_R = 1/(2\pi \sqrt{M_{p2} C_A})$$

$$f_R = 103.5 \text{ Hz}$$

7. To calculate the low frequency resonant frequency, bearing in mind Fig. 4 and taking into account the additional mass component of the radiation impedance: Radiation impedance at 105 Hz (from Ref. 2) is

$$Z_R = R_R + j \cdot \omega M_R$$

$$= 236 + j \cdot 564 \text{ acoustic ohms}$$

Effective total mass, $M'_{p2}$, is given by

$$M'_{p2} = M_{p2} + M_{H1} + M_R$$

$$= 10.44 \text{ kg m}^{-4}$$
Resonant frequency, \( f'_R \), is given by

\[
 f'_R = \frac{1}{2\pi \sqrt{M_{H2} C_A}} \\
 f'_R = 98.74 \text{ Hz}
\]

An explicit solution to this calculation is not possible because the expression for the radiation impedance mass contains a Bessel function. However, the iterative solution above gives a sufficiently accurate answer for the present purposes.

8. For the circuit shown in Fig. 5, the acoustic mass component of the 38 mm diameter hole, \( M_{H2} \), calculated in the same way as \( M_{H1} \) above is,

\[
 M_{H2} = 42.97 \text{ kg m}^{-4}
\]

**Appendix 2**

**Calculation of damping coefficients**

For the simplified, low frequency equivalent circuit shown in Fig. 5, the addition of the parallel mass and loss resistance, \( M_{H2} \) and \( R_{H2} \), increases both the resonant frequency and the loss component. At the resonant frequency (\( \approx 99 \text{ Hz} \)) the reactance of \( C_A \) is approximately \(-j \cdot 6480 \) acoustic ohms. The total circuit mass reactance is equal in magnitude and in opposing phase.

The circuit can thus be simplified to a source resistance equal to \( R_R \), 236 acoustic ohms, and an equivalent load resistance equal to 6480/Q acoustic ohms, where Q is the normal "quality factor" for a simple resonant system.

A table can be calculated showing the resonant frequency, Q-factor and peak absorption coefficient for the circuit of Fig. 5 for given values of \( M_{H2} \) and \( M_{H1} \). Table 1 shows such a table for two different values of \( M_{H2} \). The corresponding fabric resistivity can be calculated by multiplying the value of \( R_{H2} \) by the area of the hole.

The absorption coefficient, \( \alpha \), can be calculated from:

\[
 \alpha = 4 \cdot \frac{R_R \cdot R_L}{(R_R + R_L)^2 + (X_R + X_L)^2}
\]

where

- \( R_R \) is the radiation resistance
- \( R_L \) is the equivalent absorber resistance
- \( X_R \) is the mass reactance of the radiation impedance
- \( X_L \) is the reactance of the load

At resonance, \( X_R = X_L = 0 \).

<table>
<thead>
<tr>
<th>Loss Resistance ( R_{H2} ) ohms</th>
<th>( M_{H2} = 42.97 \text{ kg m}^{-4} )</th>
</tr>
</thead>
<tbody>
<tr>
<td>1 hole, 38 mm diameter</td>
<td></td>
</tr>
<tr>
<td>Resonant Frequency</td>
<td>Q-Factor</td>
</tr>
<tr>
<td>10(^3)</td>
<td>108.7</td>
</tr>
<tr>
<td>10 \times 10^{3}</td>
<td>107.9</td>
</tr>
<tr>
<td>12 \times 10^{3}</td>
<td>107.6</td>
</tr>
<tr>
<td>14 \times 10^{3}</td>
<td>107.2</td>
</tr>
<tr>
<td>16 \times 10^{3}</td>
<td>106.9</td>
</tr>
<tr>
<td>18 \times 10^{3}</td>
<td>106.5</td>
</tr>
<tr>
<td>20 \times 10^{3}</td>
<td>106.1</td>
</tr>
<tr>
<td>22 \times 10^{3}</td>
<td>105.7</td>
</tr>
<tr>
<td>24 \times 10^{3}</td>
<td>105.3</td>
</tr>
<tr>
<td>26 \times 10^{3}</td>
<td>105.0</td>
</tr>
<tr>
<td>28 \times 10^{3}</td>
<td>104.6</td>
</tr>
<tr>
<td>30 \times 10^{3}</td>
<td>104.3</td>
</tr>
<tr>
<td>32 \times 10^{3}</td>
<td>103.9</td>
</tr>
<tr>
<td>40 \times 10^{3}</td>
<td>102.9</td>
</tr>
<tr>
<td>10^{5}</td>
<td>100.0</td>
</tr>
<tr>
<td>10^{6}</td>
<td>99.2</td>
</tr>
</tbody>
</table>

| \( M_{H2} = 21.48 \text{ kg m}^{-4} \) |
|----------------------------------|----------------------------------|
| 2 holes each 38 mm diameter      |                                  |
| Resonant Frequency | Q-Factor | Absorption Coefficient |
| 117.0 | 58.2 | 0.852 |
| 113.3 | 7.23 | 0.683 |
| 112.1 | 6.58 | 0.647 |
| 110.8 | 6.22 | 0.625 |
| 109.6 | 6.05 | 0.615 |
| 108.5 | 6.00 | 0.612 |
| 107.4 | 6.05 | 0.615 |
| 106.5 | 6.16 | 0.622 |
| 105.7 | 6.32 | 0.631 |
| 105.0 | 6.51 | 0.643 |
| 104.4 | 6.73 | 0.656 |
| 103.9 | 6.97 | 0.669 |
| 103.4 | 7.24 | 0.684 |
| 102.1 | 8.4 | 0.74 |
| 99.7 | 18.76 | 0.975 |
| 99.2 | 183.6 | 0.433 |
Appendix 3
High frequency equivalent circuit

For a normally incident plane wave, the layer of mineral wool at the front of the new module can be considered as a square section transmission line of sides 0.562 m with elementary series resistance, $R$, inductance (mass), $L$, parallel capacitance (compliance), $C$, and conductance, $G$.

From Ref. 7 the attenuation constant, $A$, for the transmission line is given by

$$A = ((R + j\omega L)(G + k\omega C))^{1/2} \quad \ldots \text{A3.1}$$

In a properly sealed unit, the shunt conductance, $G$, is zero.

Therefore, putting $G = 0$, Eq. A3.1 can be written as

$$A = a + jb \quad \ldots \text{A3.2}$$

where

$$a = (\omega^2 L C) + (\omega CR)^{3/2} - \omega^2 LC)/2\quad \ldots \text{A3.3}$$

and

$$b = (\omega^2 L C)^{1/2} + (\omega CR)^{3/2} + \omega^2 LC)/2\quad \ldots \text{A3.3}$$

The characteristic impedance is given by

$$Z_0 = ((R + j\omega L)/j\omega C)^{1/2}$$

$$= (R/ j\omega C + L/C)^{1/2} \quad \ldots \text{A3.3}$$

Also from Ref. 7, it can be shown that the input impedance, $Z_i$, of a transmission line of length $l$ and characteristic impedance $Z_0$ terminated by an infinite impedance is given by

$$Z_i = Z_0 \cosh Al / \sinh Al \quad \ldots \text{A3.4}$$

where the attenuation constant, $A$, is as given in Eq. A3.2.

This can be separated into real and reactive components by expanding $\cosh Al$ and $\sinh Al$ to give

$$Z_i = Z_0 \left( e^{a+jbl} + e^{-(a+jbl)} \right) / \left( e^{a+jbl} - e^{-(a+jbl)} \right)$$

Further expansion and rearrangement eventually gives

$$Z_i = Z_0 \frac{\sinh 2al - j \sin 2bl}{\cosh 2al - \cos 2bl} \quad \ldots \text{A3.5}$$

$^\dagger$ If $a = 0$, this expression simplifies to

$$Z_i = Z_0 \left( -1 \right) \left( \cot bl \right)$$

which is the normal expression for the input impedance of a lossless transmission line with open-circuit termination.

where $a$, $b$ and $Z_0$ are as defined above and $l$ is the thickness of the mineral wool layer.

This expression can be used in the circuit shown in Fig. 6 to calculate the total impedance of the absorber at each frequency and hence the absorption coefficient for normally incident plane waves. First, however, numerical values must be obtained for the elementary transmission line parameters $R$, $L$ and $C$.

Approximate values for $R$, $L$ and $C$ can be derived as follows:

1. Elementary compliance, $C$

The compliance of a volume of mineral wool, assumed to be rigid, is very nearly equal to the compliance of the same volume of air, the proportion of the volume occupied by the fibres of the mineral wool being small even for high density mineral wool ($\approx 4 \%$ for 150 kg/m$^3$). Therefore the compliance per unit length, $C$, is given by

$$C = \text{volume}/\rho c^2$$

$$C = 2.263 \text{ m}^3 \cdot N^{-1} \text{ per metre.}$$

2. Elementary mass, $L$

The mass component of the elementary transmission line is difficult to calculate directly. It requires a knowledge of the relative size of the fibres and the air spaces between the fibres, as well as the individual fibre shapes and the material parameters such as density and Young's modulus. However, it is known that thick layers of these materials will absorb nearly all of the incident sound at high frequencies. This can only be so if the characteristic impedance of the material at high frequencies is nearly equal to the acoustic free space impedance in both magnitude and phase.

Thus, the characteristic impedance of the mineral wool, $Z_0$, at high frequencies is assumed to be 407/0$^\circ$ mks rays for unit area or approximately 1200 mks rays for the size of module being considered.

Now, from Equation A3.3.

$$Z_0 = \sqrt{L/C} \quad \text{at high frequencies}$$

$^\dagger$ If the amplitude of the air motion is comparable with or larger than the spaces between the fibres of the mineral wool then there will be significant cyclical transfers of heat between the air and the fibres. The sound wave propagation will then approximate to isothermal conditions rather than the adiabatic conditions of free space. In that case, the effective volume will be $\gamma$ time greater ($= 1.4$ for air) than the physical volume. It is assumed here that the sound pressure level is sufficiently low for this effect not to occur.
Fig. 14—Calculated normal incidence absorption characteristics of 150 kg m$^{-3}$ mineral wool with rigid backing and 20% perforated hardboard front cover.

Fig. 15—Calculated normal incidence absorption characteristics of 150 kg m$^{-3}$ mineral wool with rigid backing.
Therefore,

\[ L = Z_0^2, \quad C = 3.258 \text{ kg m}^{-4} \text{ per metre} \]

3. Elementary resistance, \( R \)

A wide range of resistance values have been measured for mineral wools, even for different samples cut from the same piece. A representative value of 2000 mks rays is typical for a 30 mm thick sample of high density mineral wool.

For the elementary transmission line this is equivalent to 2000/(Area \times 30 \text{ mm}) rays/metre.

Therefore \( R = 211,000 \) rays per metre.

Using these approximate derived values and equations A3.5, A3.3 and A3.2, the input impedance of the transmission line and hence the absorption coefficient can be calculated at any frequency. Figs. 14 and 15 show the calculated absorption coefficients as functions of frequency for different thicknesses of the mineral wool layer, both with and without the 20% open-area ratio front cover.

The shapes of these characteristics show that even for these comparatively simple structures and for normally incident plane waves, the results are surprisingly complicated functions of frequency. Clearly, the application of thick layers of mineral wool for sound absorption is not as simple as it might at first appear – even without the perforated front cover.

Figs. 14 and 15 also show some results measured for normal incidence for both a 30 mm thick and a 10 mm thick mineral wool layer. The agreement between the theoretical and the measured results is not good, although the same trends appear in both the theoretical and the measured results. This shows that, whilst the theory is not numerically correct and it is hardly surprising considering the initial assumptions, the general principles are confirmed by the measurements.

The characteristics shown in Figs. 16, 17 illustrate the comparison between the theoretical results calculated for normal-incidence plane waves and the diffuse-field results obtained by measurements on the completed modules in the reverberation room. The two measured conditions of 30 mm thick and 60 mm thick mineral wool layers are shown (these are the same results as plotted in Figs. 11 and 10). Comparison was only valid at frequencies above about 500 Hz because, below that frequency, the module is too small for the radiation impedance to be purely resistive. Some allowance has to be made for the inherent absorption of the box material. The comparisons shown in Figs. 16 and 17 were obtained by adding 0.6 to the theoretical absorption coefficients at all frequencies. This was an arbitrary and probably too large a figure to be accounted for entirely by the box surface absorption. It was deliberately chosen to illustrate the close fit of the shapes of the measured and theoretical absorption coefficient characteristics.

Thus, the simple, normal-incidence theory seems to predict the shape of the high-frequency absorption-coefficient characteristic fairly well, but not the absolute level of the characteristic.

![Figure 16](image1.png)

Fig. 16—Comparison of measured and calculated results for high frequencies for complete module.

(30 mm thick mineral wool layer)

![Figure 17](image2.png)

Fig. 17—Comparison of measured and calculated results for high frequencies for proposed new module.

(60 mm thick mineral wool layer)