A NUMERICAL INVESTIGATION INTO AN EMPIRCALLY DESIGNED ABSORBER USED IN RECORDING STUDIO CONTROL ROOMS

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1. INTRODUCTION

The investigation described in this paper forms part of a larger research project at the ISVR into an empirically designed absorber used primarily in recording studio control rooms.
The absorber comprises large chipboard panels covered with mineral wool and freely suspended from the ceiling structure such that they make a 50° angle to the normal of the absorbent walls of the room. The centres of adjacent panels, which are parallel with each other, is around 0.4 metres. The absorber is applied to all room boundaries apart from the monitor wall and floor, both of which are reflective. Due to the efficiency of the absorber in dissipating low frequency sound energy, it is commonly referred to as a *Bass Trap*.
Previous publications by the authors have concluded that flexural vibrations in the panels are not responsible for low frequency absorption [1], and that the side wall *Bass Trap* introduces significant attenuation, in a narrow frequency band, as sound propagates down the room at grazing incidence to the side walls [2]. The subject of this investigation however, is the absorption characteristic of the rear wall of a *Bass Trap* room.
One possible mechanism in particular is considered, and is investigated using a Finite Element model. The post-processing capabilities of the FE software (*Sysnoise*) are used to yield results for pressure and velocity distribution, whilst further analysis performed on exported data allows for absorption coefficient and surface normal acoustic impedance to be found.

2. 'HORN EFFECT' HYPOTHESIS

The possible mechanism of absorption explained in this section, is the brainchild of Professor F J Fahy, who suggested the idea to the authors after a number of conversations on the subject. The hypothesis is based on the acoustic properties of conical horns, and the principle of reciprocity.
To start with, some basic theory about horns is given, using the 2-D model illustrated in *Figure 1.* Horns have long been used to increase the radiation efficiency of direct radiating loudspeakers, by matching the acoustic impedance of the loudspeaker to that of the throat of the horn[1]. Interestingly, even small changes to the flare rate have been shown have a marked effect on the radiation efficiency when the wavelength of sound is many times greater than the dimensions of the horn [3].
In *Figure 1* a conical horn is illustrated, with a small direct radiating loudspeaker placed at the throat. The loudspeaker is assumed to act as a piston, enabling its behaviour to be characterised as a periodic velocity source, u_1. Due to the horn, the velocity source gives rise to a large pressure p_2 at the mouth. Applying the principle of reciprocity to the system, a velocity source at the mouth u_2 will cause a large pressure p_1 at the horn's throat. The significance of this elementary argument will become clear from the geometry of the rear wall *Bass Trap*, illustrated in *Figure 2.*

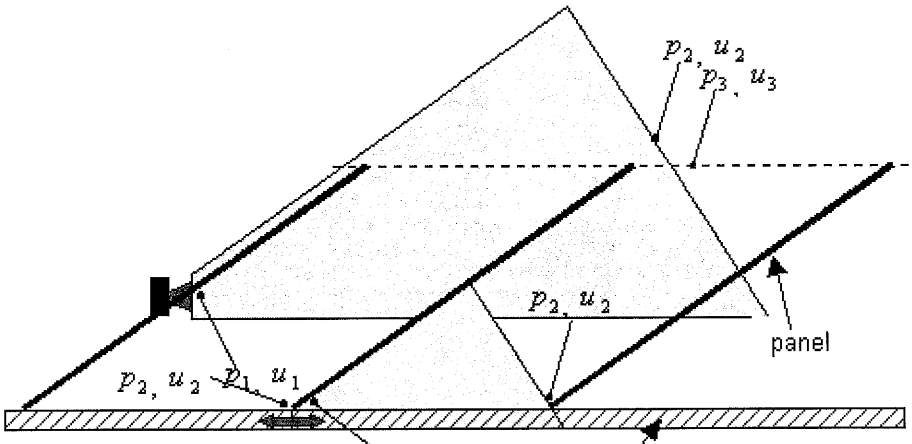

Figure 1 – Conical horn with pressure and velocity shown at the throat and mouth

Figure 2 – Back wall 'horn effect' model (red arrow indicates anticipated velocity)

In this diagram, the conical horn of the previous figure can be seen as a shaded area, together with the aforementioned pressures and velocities. Applying the same argument as before, a velocity source u_1 will cause a large pressure at the mouth p_2 and a similarly large pressure p_3 at the open end of the *Bass Trap* channel. One would expect little difference in magnitude between p_2 and p_3 as there is no horn flare in this section of the channel. Using reciprocity again, it can be seen that a velocity source at the open end of the channel u_3 will give rise to a large pressure p_1 but a far smaller pressure p_2. It is the periodicity of the rear wall *Bass Trap* that makes this

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evident either side of the panel, as the magnitude of p_1 will be larger than p_2. A pressure difference such as this, concentrated in a small area will result in high particle velocity **through** the porous outer layer of the absorbent wall (shown as the red arrow), it is reasonable therefore to expect significant absorption. Generally, low frequency absorption in rooms is not easily achieved through porous absorption, as the high particle velocity necessary occurs ¼ wavelength from a rigid boundary – this may represent a number of metres. The hypothesis presented here however could prove to be far more space efficient. From the explanation given it is clear that any investigation must be able to account for particle velocity motion through an absorbent layer

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3. NUMERICAL INVESTIGATION USING ABSORBENT ELEMENTS

3.1 Description of the Absorbent Element FE Model

Figure 3 describes the Finite Element mesh used in the numerical investigation. As the interaction between adjacent channels is key to the *horn effect*, the FE model uses four channels, although only the acoustic behaviour of the middle two is analysed. By having the two outer 'dummy' channels, the two in the middle should be characteristic of the large array employed in a typical rear wall *Bass Trap*.

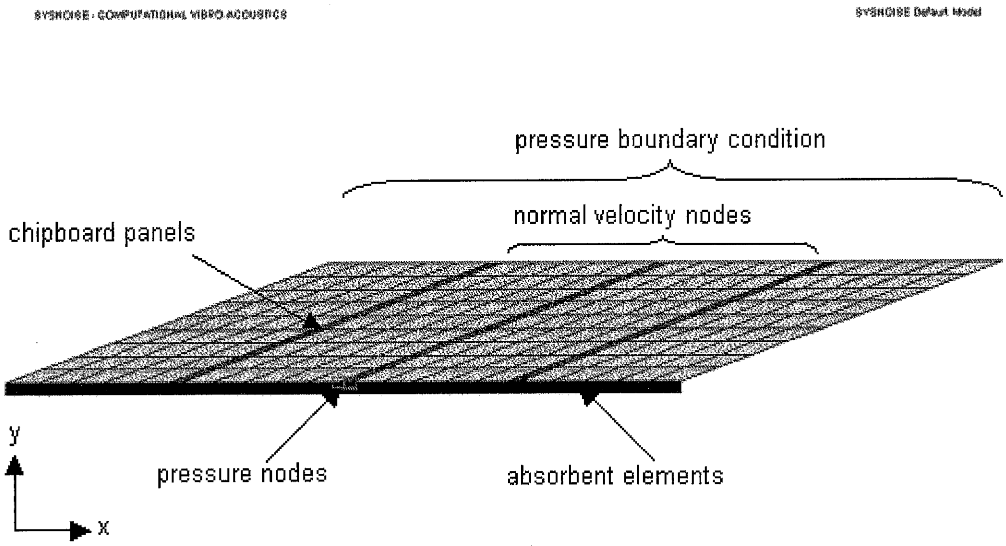

Figure 3 – Finite Element model of the back wall absorber

A pressure boundary condition is specified across the open end of the channels, with the normal velocity of only the middle nodes used in the calculation of the rear wall absorption coefficient. The chipboard panels used in the *Bass Trap* are characterised by assigning the appropriate density and sound speed (650 kg m^{-3} and 3700 m s^{-1}) to the relevant elements. Given that a large pressure difference is anticipated at the wall between one side of the panel and the other, the pressures at two nodes in this region were calculated – these are highlighted in the figure.

The blue shading in *Figure 3* shows the elements used to describe the porous outer layer of the absorbent wall. In addition to the aforementioned material properties, any element in a FE mesh can be assigned a porosity, structure factor, and resistivity. Between these five variables, it is possible to model many porous materials, thus allowing the propagation of sound through them to be analysed.

The material properties assigned to the absorbent elements in the FE model were chosen to characterise a typical medium density porous material, not the cotton waste felt used in the full scale design. By choosing a medium value for resistivity, and typical values of structure factor [4] and porosity [5], the colour maps of pressure and particle velocity within the absorbent elements could be more clearly seen. Clarity in the results allowing ratification of the hypothesis, was deemed more important than making the model specific to a particular design. The sound speed and density was that of air. The table below summarises the quantities specified.

Sound speed	Density	Porosity	Structure factor	Resistivity
340 m s^{-1}	1.205 kg m^{-3}	0.95	2	2 x 10^4 Pa s m^{-2}

3.2 Pressure, Velocity, and Absorption Coefficient Results

[illegible] ny [illegible] om *Sysnoise* for analysis and manipulation in *MatLab*, or by using the post-processing capabilities of *Sysnoise* itself. The results are presented according to the order in which the hypothesis was explained. Firstly, a colour map of pressure is shown at a particular frequency of interest. By this we mean a graphical illustration of the pressure magnitude (red end of the spectrum describing large values) within the rear wall *Bass Trap* FE mesh. This result will show whether or not the expected difference in pressure at the junction of wall and panel is present. The corresponding particle velocity colour map will follow, which it is hoped will illustrate an increased particle velocity within the absorbent elements. Once the colour maps have been presented, the absorption coefficient at the open end of the channel, and the magnitude of the pressure difference between the aforementioned nodes will be plotted against frequency. The pressure difference is shown normalised to the pressure boundary condition p_0 at the open end of the channel, and calculated according to $p_{diff} = \left| \frac{p_1 - p_2}{p_0} \right|$, where p_1 and p_2 are the complex pressures either side of the panel. Finally a plot of the normalised reactance and resistance at the open end of the channel will be shown. It is anticipated that, between the four figures, there will be sufficient evidence to prove or disprove the hypothesis described in *Section 2*. Although the analysis was repeated for three different angles of panel, for the sake of brevity only the 50° orientation is included here.

3.2.1 50° Channel Rear Wall *Bass Trap*

In *Figure 4* the magnitude of the pressure is illustrated by the different colours. The absolute values, barely visible on the scale to the right of the figure, will be presented later – it is the contrast that is the important feature of this colour map. As mentioned earlier, discussion will be restricted to the centre two channels. The reason for displaying the colour map at 230 Hz, is because a peak in the absorption coefficient is evident at this frequency – this result will be shown in *Figure 6*.
The figure clearly shows that a large pressure (shown in red) occurs at the throat of the horn in the third channel from the left, and contrasts with the lower pressure magnitude everywhere else. In addition, the pressure on the opposite side of the panel is the lowest (extreme blue end of the spectrum) in the colour map. These two observations therefore prove the first part of the hypothesis. The following result (*Figure 5*) will show whether or not this pressure difference gives rise to a high particle velocity in the absorbent.

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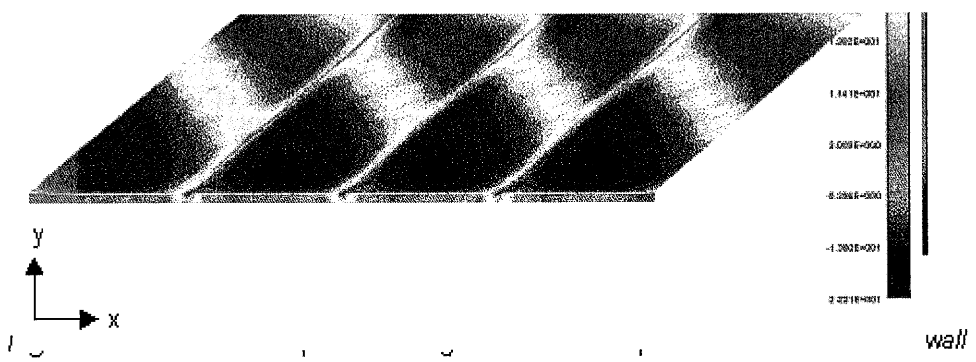

Figure 5 – FE calculated x-vector velocity colour map at 236 Hz for 40° back wall

The colour map for the particle velocity is on a dB scale as the large dynamic range would make a linear representation unclear. Velocity is a vector quantity, thus only one direction at a time can be illustrated using a colour map. As the hypothesis hinges on the oscillatory motion of velocity through the absorbent, it is the x-component of the vector that is presented. As before, it is the contrast between colours that is of primary importance. Concentrating the discussion on the particle velocity within the absorbent layer, it can be seen that there is a significant increase in the region near the panel. Indeed, immediately behind the panel, the amplitude of the particle velocity is about 40 dB greater than other parts of the absorbent. The premise that the pressure difference identified in *Figure 4* would give rise to this, is thus proved correct. It is important to note that velocity normal to the absorbent will also yield significant absorption, and the colour map in *Figure 5* is not wholly representative. However, whilst it is unlikely that the x-direction velocity motion in this region of the absorbent accounts for the total absorption of the rear wall *Bass Trap*, it is a feature that will undoubtedly prove beneficial.

In order to test this hypothesis, the plots of absorption coefficient and surface normal impedance to re ne

position where the high particle velocity was noted. As a comparison, the FE mesh is altered by extending the panel to the wall through the absorbent layer, thus impeding any velocity motion. This change to the model will remove any absorption due to the *horn effect*. Alongside this dual trace plot will be the pressure difference either side of the panel, as it is hoped the comparison between the two will indicate any connection between the *horn effect* and the total absorption of the *Bass Trap*. These results are shown in *Figure 6*.

With no absorbent gap, the absorption coefficient exhibits three peaks, the second being the frequency to which the colour maps relate, though the general trend is a smooth rise in absorption with frequency. The exact frequencies of these peaks approximate to odd integer multiples of ¼ wavelength, though the rhomboidal channel means that the frequencies do not tie in precisely. From the green trace we can conclude that a ¼ wavelength resonance is primarily responsible for the absorption. With an absorbent gap, the absorption is increased at all frequencies above 150 Hz.

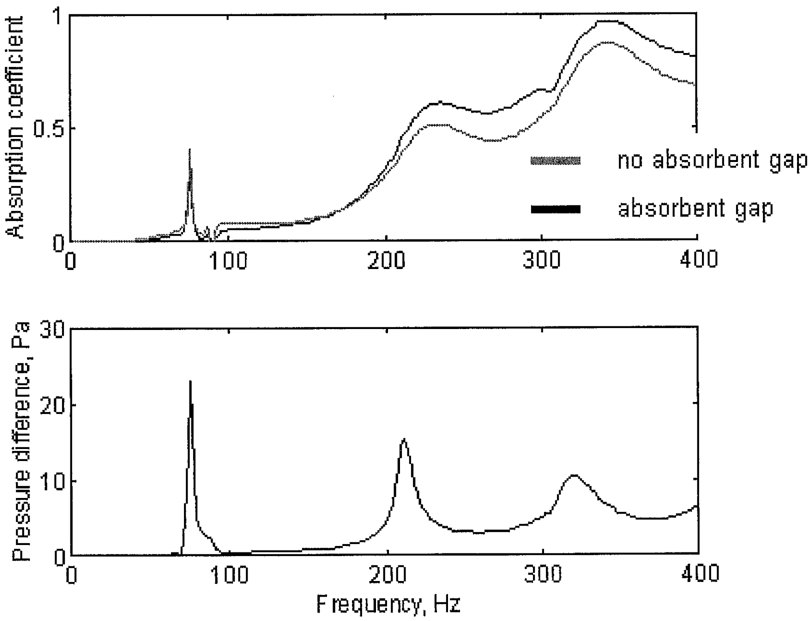

Figure 6 – FE calculated absorption and normalised pressure difference for back wall panels at 40°

The pressure difference either side of the panel, at the absorbent, is larger at this angle than at others, with the peaks at 210 and 320 Hz increasing the absorption even at the ¼ wavelength resonance. Given that the frequencies of maximum pressure difference are slightly lower at other angles of panel, it would suggest that changing the geometry of the *Bass Trap* influences the frequency at which the *horn effect* is most effective, in the same way that changing the flare rate of a horn influences its radiation efficiency. If therefore, the rear wall *Bass Trap* used different depth, angle, and spacing of panel, the *horn effect* could be positioned at different frequencies, enabling the *Bass Trap*, as a whole, to be more effective over a wider frequency range.

Figure 6 has served to give great insight into the mechanism of absorption of the rear wall *Bass Trap*, and the significance of the *horn effect*. The primary effect of resonance at odd integer

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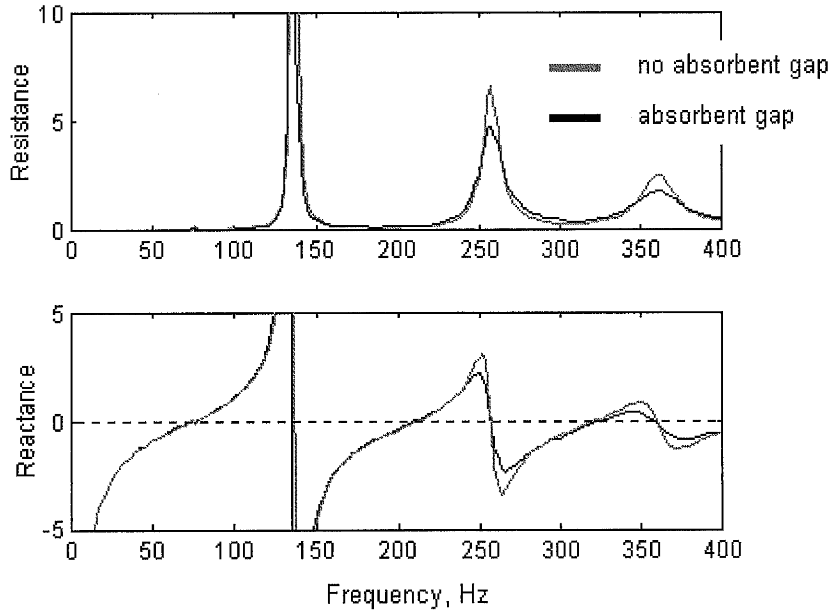

Figure 7 – Surface normal impedance (normalised to $\rho_0 c_0$) for back wall panels at 40°

Further insight is given in the final result – the normalised surface normal resistance and reactance at the open end of the channels, shown in *Figure 7*.

As with the absorption coefficient, both traces are fairly similar. Each exhibit strong ½ wavelength resonances at 140 and 260 Hz where the resistance in the system maximises and the reactance transitions between a large positive to large negative value. At these frequencies, the assertion that the *horn effect* weakens the resonance, is shown to be true, as variation in magnitude of both parts of the impedance is less. Apart from at the ½ wavelength resonances, the presence of an absorbent gap increases the acoustic resistance slightly. As the frequency increases, the impedance of the rear wall *Bass Trap* appears to tend to a slightly negative reactance, and the resistance to a value close to the characteristic impedance of air.

4. CONCLUSIONS

The investigation described has been primarily concerned with a hypothesis that makes the connection between the radiation efficiency of conical horns, and porous absorption in the rear wall *Bass Trap*.

Through the use of absorbent elements as part of a FE model, every stage in the 'horn' effect hypothesis was ratified, from the large pressure difference, to the associated particle velocity giving rise to additional absorption. The absorption afforded is, as mentioned, only supplemental to the primary mechanism which can be clearly seen at the more obtuse angles as a ¼ wavelength

absorption away from resonance. Additional aspects of the *Bass Trap* design, neglected in the FE model, will also serve to make it a highly effective means of absorption. The 4 cm thick felt covering both sides of all the angled panels will lower the Q of the resonances by adding a significant amount of acoustic resistance to the system. In the impedance plots presented, it was seen that the resonances were weakened by the presence of an absorbent gap. This effect will be even more accentuated in the real life design, as in practise the panels cannot be placed hard against the multi-layer wall, and thus a small air gap between the two is present. The degree to which this also weakens the 'horn' effect will require further investigation, though it seems likely that the magnitude of pressure difference, particle velocity and thus absorption will all be less.

Although the 'horn' effect was investigated here in the context of the *Bass Trap*, it would undoubtedly warrant consideration in its own right. The research in this chapter has shown it to be a useful acoustic phenomenon, and one that could be extended into possibly a stand-alone absorber. If a modular unit incorporating at least two horn-like geometries (though not necessarily conical), and a suitable porous material (50 kg m^{-3} mineral wool), were placed in a position of high pressure (the corner of a room would be ideal), then it might be possible to reduce low frequency reverberation without having to use the full *Bass Trap* design.

References

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[3] Holland, K., 1992: 'A Study of the Physical Properties of Mid Range Loudspeaker Horns and Their Relationship to Perceived Sound Quality', PhD thesis, ISVR, University of Southampton

[4] Cremer, L., & Muller, H., 1982: *Principles and Applications of Room Acoustics, Vol. 1*, (London: Applied Science), p. 349.

[5] Cremer, L., & Muller, H., 1982: *Principles and Applications of Room Acoustics, Vol. 2*, (London: Applied Science), p. 132.

[1] The narrow end of a horn is known as the throat, and the wider end, the mouth.