INVESTIGATION OF THE POOR MID-FREQUENCY SOUND REDUCTION CHARACTERISTICS OF PROFILED CLADDING

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1. INTRODUCTION

Profiled metal cladding constructions are an increasingly prevalent feature of modern industrial architecture. Whilst such materials conform to high structural standards, their acoustic characteristics are often poor. Previous work by the authors [1] has shown that large depressions in the single-skin sound reduction (of up to 10dB in magnitude) occur at mid-band frequencies and are most likely caused by certain vibrational modes. When the profiles are incorporated into more common double leaf structures, the dip in transmission loss remains. These features have also been noted during *in situ* measurements [2]. It is considered that this potentially poses a significant environmental noise problem. Present methods of predicting transmission through cladding fail because they are global approaches, i.e. they are unable to account for "localised" effects such as specific vibrational modes. For example, Heckl's method [3] is known to provide a reasonable estimation [2], but will be inaccurate where the described dips occur. Consequently, this paper shall attempt to establish exactly the physical process which precipitates the dips in sound reduction, in order to formulate a more reliable prediction method.

Sound reduction measurements have been undertaken on a large number of profiles with various gauge, pitch, depth, asymmetry and altered stiffness [1]. The magnitude and frequency of the SRI dips were shown to be highly sensitive to relatively small changes in these cladding "profile" parameters, which are defined in figure 1. A "symmetrical" profile has equal crown and valley lengths. A clear trend can be seen when one compares the characteristics of four profiles of the same material, gauge and dimensions apart from the profile "depth" (fig.2). Obvious depressions in the transmission loss occur in the 1K25, 1K, 800 and 630Hz 1:3 octave bands as the depth is increased from 25 to 55mm respectively. One might also observe "dips" in each curve around 200Hz and 2KHz.

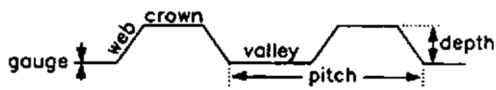


Figure 1 : Definition of cladding profile parameters.

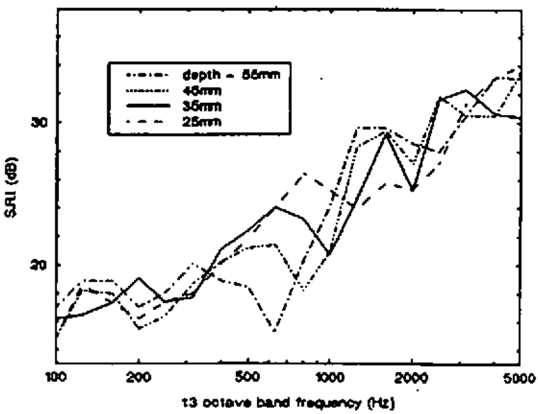


Figure 2 : Variation in the sound reduction of symmetrically profiled steel cladding as depth is varied. Other parameters are constant (gauge = 0.65mm, profile pitch = 250mm).

Similar results were obtained by Cederfeldt [4] who concluded that the decline in transmission loss was partly due to air resonances in the cavities created by profiling. This was based on the observation that the dips were largely negated by filling the profile troughs with a porous absorber. However, by doing this one might be damping any air resonances that may occur or damping the vibrational modes directly as the absorbent was in contact with the cladding surface. In any case, it is not believed by the authors that the observed dips were significantly reduced, but that the whole sound reduction characteristic was simply increased across the frequency spectrum. Cederfeldt also stated that "plate resonances are of great importance". It is postulated that vibrational modes are more likely the sole cause of the "dips".

2. VIBRATION MEASUREMENTS ON CLADDING

2.1 Diffuse Field Excitation

Initial measurements were undertaken on the cladding fully constructed in a transmission suite aperture such that the incident noise field is consistent with sound reduction measurements. The construction is as close to *in situ* practice as possible. An accelerometer was placed at 1cm intervals across the profile (horizontally) and along the profile (vertically). The surface density of cladding is generally much less than $10kgm^{-2}$. One should thereby take great care that the placement of transducers on its surface does not overload the vibrating mass. It is estimated that the frequency domain error will not be more than 5% using a 0.65g accelerometer. In order to normalise the acceleration to input force, the diffuse field pressure is measured in the source room. The transfer function is then analogous to inertance (or "accelerance").

Figure 3 shows some typical results for the symmetrical profile with a 35mm depth. One can see that distinct peaks occur around 1KHz and 2KHz which compare exactly to depressions in the transmission loss (fig.2). This correlation occurs consistently for *all* of the profiles measured, such that a definite link between peaks in vibrational response and the SRI dips has been established.

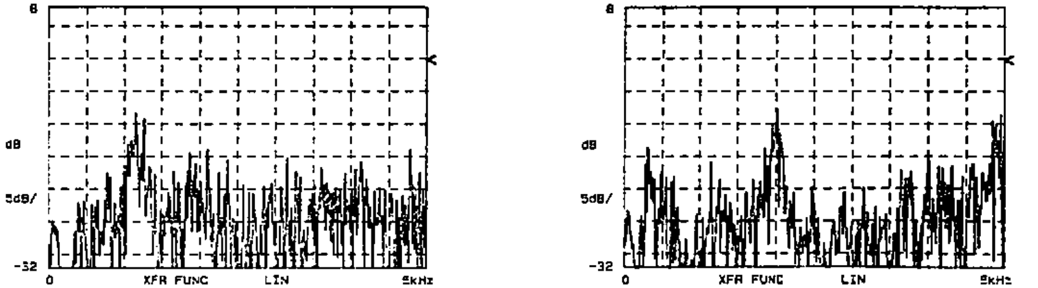


Figure 3 : Inertance-analogous transfer function measured on fully constructed cladding wall (profile depth = 35mm). Accelerometer placed at (a) centre of profile web, (b) centre of profile crown.

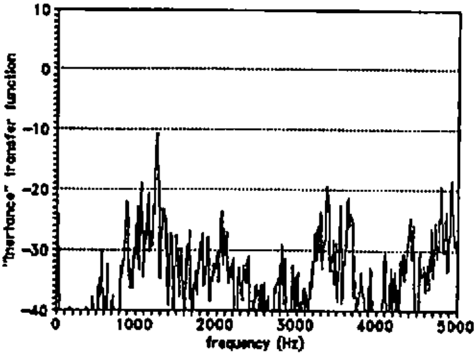
Cederfeldt applied a rudimentary finite element model to idealised profiles [5] in order to demonstrate that the theoretical radiation was high for certain modes. However, this could not be verified by measurements because the profile velocity was only considered in 1:3 octave bands and was not normalised to the force input. Further, a 12.7g transducer was used such that the error could have been as much as 70%.

2.2 Modal Analysis

It is required to obtain the modeshapes associated with the various profiles considered. This demands reliable phase information, i.e. the coherence function between the two signals must approach unity. Diffuse field, random noise measured in the transmission suite is clearly inapplicable. Neither was it found possible to use a microphone or intensity probe placed very close to the wall when the noise source is a discrete sinusoid. Two accelerometers may be placed in close proximity on the profile with one as a phase reference. However, they must remain close and it was considered that this would load the vibrating mass too much. Impulse excitation with an impact hammer was also undertaken, but was found to be incapable of exciting the same response as random noise.

It is evident that acoustic excitation, preferably continuous random noise, should be employed whilst maintaining coherence. This apparent dilemma is resolved by application of maximum length sequence (MLS) analysis [6]. Broadly speaking, the MLS signal is a very long binary sequence which is thereby deterministic, such that it is often referred to as "pseudo-random noise". The signal is entirely consistent and measurements need not be simultaneous for phase information to be retrieved. Analysis was undertaken using a commercial, PC-based system known as MLSSA [7].

A single profiled panel is hung at the end of an anechoic room. A loudspeaker is placed at the opposite end, on a normal from the cladding. The separation is as large as possible (~5m) in order to ensure that the spherical wave is approximately plane at the panel. A microphone is placed on the normal between loudspeaker and cladding such that the incident sound pressure can be measured and used as phase reference. The complex transfer function, $H(\omega)$, is then obtained. Broadly similar peaks in response are observed (fig.4), corresponding to the sound reduction dips (fig.2). The validity of using plane wave excitation may then be appraised by comparing figs.3 and 4. Again, the correlation between peaks in vibration and depressions in transmission loss holds for all of the profiles tested.



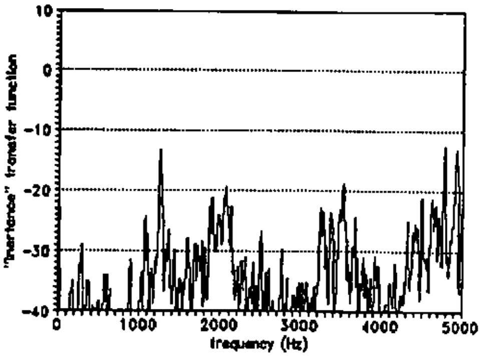


Figure 4 : Transfer function measured on a single profiled panel excited by normal incidence plane waves. Profile dimensions and accelerometer placements (a) and (b) as in figure 3.

The separation between accelerometer placements is 1cm, such that a sufficient "sampling rate" is achieved for measurement of modeshapes up to 5KHz. This may be roughly checked by consideration of a homogeneous plate attributed with the same bending stiffness as the orthotropic panel in each dimension. A single-degree-of-freedom system is assumed about each mode, such that the real part of the transfer function, $H(\omega)$, tends to zero at the modefrequency. One can then plot $\mathrm{Imag}\{H(\omega)\}$ over the profile where this occurs. Figures 5 and 6 demonstrate a selection of modes measured on two panels.

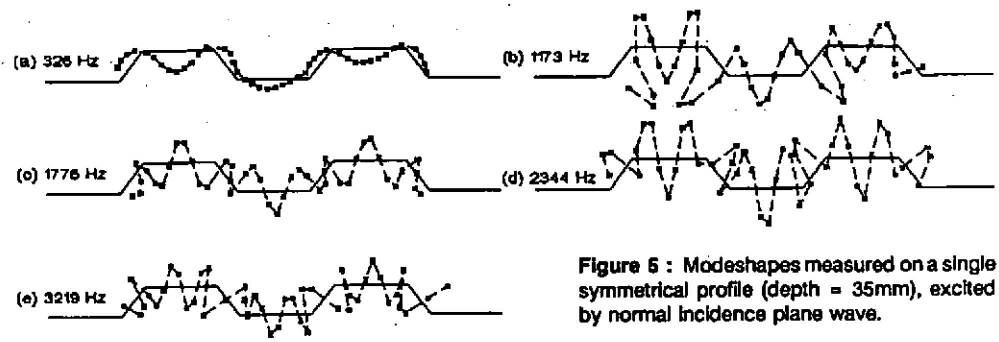


Figure 5 : Modeshapes measured on a single symmetrical profile (depth = 35mm), excited by normal incidence plane wave.

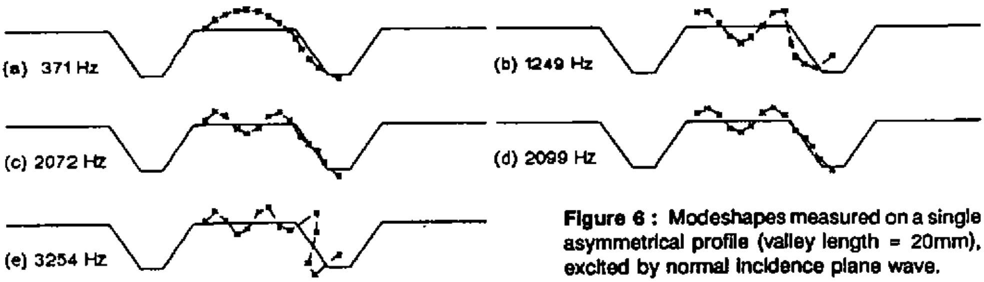


Figure 6 : Modeshapes measured on a single asymmetrical profile (valley length = 20mm), excited by normal incidence plane wave.

Modeshapes were measured on several profiles. It was noted that distinctly similar shapes occurred on all of the measured panels; of which certain modes consistently corresponded to the large peaks in vibrational response and to the depressions in transmission loss. For example, the modeshapes presented in figs.5 a, b and d correspond to the dips observed in the response of the same profile (fig.2, depth = 35mm). Likewise, the modes shown in figs.6 a, b and d are intuitively similar to those in fig.5 and relate to dips in the asymmetrical profile's transmission loss characteristic. Hence, it may be possible to predict the occurrence of dips by "tracking" particular modes. However, one also needs to explain why specific modes cause heightened radiation whereas others do not, e.g. modes (c) and (e) in figs.5 and 6.

It should be noted that measurements taken over 10cm of the flat cross-section of cladding produced a nearly constant level of "inertance" for all of the modes considered. In other words, the bending wavelength in the vertical dimension is far greater than that over the profiled section. This is as much as could be expected given the orthotropic properties of profiled plates, which dictates that the bending stiffness over the flat section may be of the order of 10^4 greater than that over the profiled section. This would seem to suggest that a two-dimensional approach may be valid.

SOUND REDUCTION OF PROFILED CLADDING

3. FINITE ELEMENT ANALYSIS OF CLADDING

3.1 Method

The finite element method provides an obvious means of synthesizing the modes observed thus far. One problem associated with this particular application is that relatively high order modes are required. Essentially, the higher the mode order scrutinized, the more elements are needed in order to avoid large errors in matrix computation. Plainly, the process should be reduced to as rudimentary a level as possible. It was therefore useful to keep the problem two-dimensional. Accordingly, simple beam finite elements were applied with their motion restricted to one plane. Further, one must take care to fashion the bulk modulus correctly when using a 2-D beam model to simulate a 3-D plate structure.

Observation of modenumbers m>20 generally required as many as 800 finite elements. It should be recognised that for such high order modes, the error may become large and unpredictable. One should therefore attempt to eliminate the number of "useless" modes located by the model; e.g. global modes dependant on the number of profile pitches or fixing conditions, and modes which are non-continuous. Several FE models were applied in order to assess the variation due to selected boundary conditions. A solution was to approximate an "infinite" profile. This was achieved by "constraining" the profile end-sections to have the same motion as nodes in structurally similar areas of the cladding (fig.7). Such a model yielded only symmetrical modeshapes, i.e. those which are repeated over finite profile sections. Comparison with larger models (up to six profile pitches) proved the "infinite" approach to be the most consistent. This is useful because one is unsure of the real fixing conditions of profiled cladding within a full wall construction. Every profile used in the sound reduction tests [1] was modelled in this manner. Symmetrical profiles were also analyzed over three profile pitches with both simply held and free ends. This served to confirm that there were no conspicuous omissions of significant local natural modes by the "infinite" approximation.

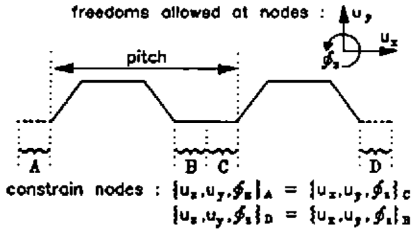
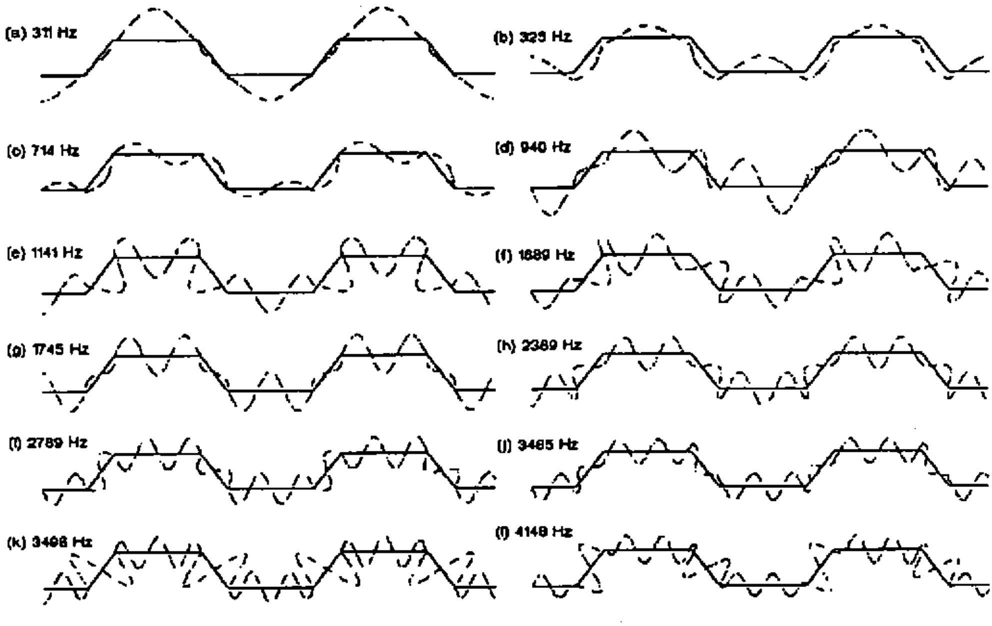


Figure 7 : Approximation of an infinite profile using finite element constraints.

3.2 Results

All continuous, natural modes found by FE analysis (between 300Hz and 5KHz) are presented in figure 8, for the symmetrical panel with 35mm profile depth. One can immediately note the excellent correlation with measured modes (fig.5). The "infinite" model is also able to reproduce all of the measured mode shapes. It should be borne in mind that the "measured" frequencies will be prone to error because of the simplified SDOF system on which they are based; whereas the FE modes are natural and will not include residual effects of adjacent modes, but are subject to computational errors which generally increase in relation to the mode order. There is also an uncertainty in physical constants. Hence, the predicted modefrequency accuracy observed is remarkable. Similar results were achieved for *all* of the profiles tested.

As has been noted, certain modeshapes were seen to correspond to dips in the transmission loss. One can compare the modes of figs.5 a, b, d to figs.8 b, e, h respectively. Intuitively similar modes were observed on the other profiles and consistently appeared about the same frequency as the depressions in sound reduction : fig.9 demonstrates this relation, where each point represents the occurrence of a particular mode on a different profile. For example, as one varies the profile depth (fig.2) the modefrequency of the general mode shape shown in fig.9b corresponds to the SRI dips observed in the 1K25, 1K, 800 and 630Hz 1:3 octave bands. This shows that the two dimensional finite element model is capable of predicting the frequency at which transmission loss dips occur by "tracking" modes which are known to correspond to dips.



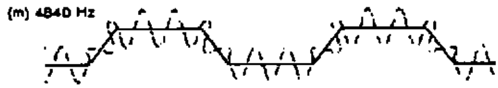
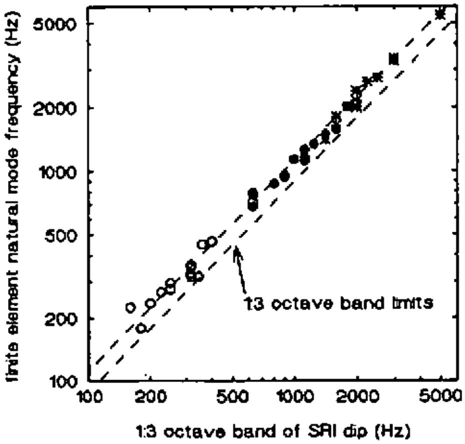


Figure 8 : Natural modes on symmetrical cladding (profile depth = 35mm), produced by "infinite" finite element model.



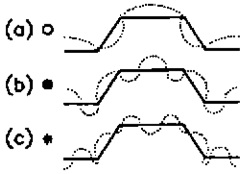


Figure 9 : Scatter plot showing relationship between FE natural modes and occurrence of dips in the SRI, for three generalised mode shapes.

Several other deductions were made from the finite element results. Firstly, it is clear that the frequency at which one particular modeshape occurs on a certain profile is precisely proportional to the metal gauge. Accordingly, one need only predict the transmission loss dip for a profile of arbitrary gauge. One may also note the similarity of modes on various asymmetrical profiles where similar profile sections occur. Hence, some control of the dip magnitude is possible by varying the degree of asymmetry, c.f.[1].

3.3 Empirical Prediction of Modes

Finite element analysis has been shown to yield a dependable prediction of the depressions in transmission loss. However, it is a fairly long-winded procedure requiring skilled operators. It is thereby considered necessary to apply the results in order to define a simpler method which may be used by any design engineer, for example. The use of simple beam elements in the FE analysis might lead one to contemplate applying elementary bar theory to predict given modeshapes. This was found to be possible by representing the crown and web lengths separately as bars with pivoted ends, then setting the modenumber equal to the number of half-bending wavelengths on the particular section of profile. If the resulting frequencies are simply averaged, one can achieve a very good prediction of the actual modefrequency. However, this will only work for modes with small displacements at the bends in the cladding, corresponding to a pivot.

This crude technique led to an empirical prediction of all modes, by back-substituting natural modefrequencies calculated by FE analysis into the bar equations, in order to obtain a qualitative "real" modenumber. If one assumes that the ratio of crown to web length (ℓ_C/ℓ_W) is equal to the ratio of the real modenumbers for each length, it is possible to plot ℓ_C/ℓ_W against either modenumber in order to obtain a simple polynomial relationship. This may be done separately for each mode associated with a sound reduction dip, yielding empirical charts which may be used to predict modefrequencies on any symmetrical, trapezoidal profile without resorting to finite elements. One might note that the length ratio ℓ_C/ℓ_W is seen to be the most significant factor in determining the frequency of a particular mode (it has already been demonstrated that modefrequency is exactly proportional to metal gauge).

4. THEORETICAL CONSIDERATION OF RADIATION FROM VIBRATIONAL MODES

It is clear that some formulation is required to describe sound radiation, such that one is able distinguish efficient vibrating modes. The required descriptor for the acoustic output of a particular mode is the radiation ratio, σ, which relates the radiated sound power to the magnitude of structural vibration :

$$\sigma = \frac{\overline{W}}{\rho_o c S \langle \overline{v^2} \rangle} \qquad (1)$$

where $\overline{W}$ is the time-averaged power radiated by the source of surface area S. The "average mean square velocity" $\langle \overline{v^2} \rangle$ is defined as the spatial average of the time-averaged squared velocity.

The most theoretically rigid way of modelling sound radiation from vibrating surfaces is by use of the Helmholtz-Kirchhoff integral equation. In practice, one must break the profile shape down into discrete sections to which velocities are assigned, enabling the calculation of surface pressure. The model used was three dimensional [8] which necessitated a large number of elements. However, computation limits restricted the effective radiating area and the modeshape itself was insufficiently detailed. For this reason, it was decided to begin development of a two-dimensional boundary integral solution. However, it is possible to formulate a much simpler method which does not involve such excessive computation periods.

A "low frequency" model may be appropriate because the frequencies considered are well below the critical frequency related to the profiled cross-section of the cladding. Consequently, the simplest possible manner of representing the vibrating profile is to replace it with a single simple source. The source strength Q is then defined by the mode shape :

$$Q e^{j\omega t} = \int_S u.\hat{n}\, dS \qquad where \quad u = U_0 e^{j(\omega t + \phi)} \tag{2}$$

where $u.\hat{n}$ represents the normal surface velocity, with magnitude U_0. Hence, the steady state source strength can be gained from $Q = S_i U_0$, where U_0 is simply attributed positive or negative phase and S_i is the area associated with each source. In other words, instead of replacing each half bending wavelength on the panel with a point source, one can calculate the sum of the vector volume velocity over the profile which will yield a residual or "uncancelled" value. In practice, modes with a high level of radiation will be those for which "inter-cell cancellation" is not complete because of the modeshape, which is dictated by the dimensions of the cladding profile.

For a simple source radiating omnidirectionally into a half-space, one can use the well-known elementary relationships between pressure, intensity and power substituted into eqn.(1) to yield a simple expression for the radiation ratio :

$$\sigma = \frac{k^2 Q^2}{4\pi S \langle \overline{v^2} \rangle} \tag{3}$$

where k is the air wavenumber and Q now represents the summed vector volume velocity over the total surface area S of the source. This approach is not wholly consistent : the modeshape data exists in two dimensions, whereas the theory is three-dimensional. Nevertheless, comparison between the efficiency of modes may be achieved by regarding only one profile pitch in each case. S is then the total surface *length* of the profile cross-section. The volume velocity sum may be calculated from both measured and FE modeshape data, from which one computes a *comparative* value of radiation ratio.

Figure 10 shows the residual volume velocities and radiation ratio for the symmetrical panel with 35mm profile depth. The volume velocity is clearly the dominant term. However, what is most interesting is that the modes which are identified as exhibiting relatively high radiation, using this procedure, are precisely those which have been previously selected as corresponding closely to depressions in transmission loss. For instance, the modes shown in figs. 8 b, e, h and m can be matched to SRI dips. The same modes correspond to the peaks in radiation at 325, 1141, 2388 and 4839 Hz respectively (fig.10). Again, this result is matched for *every* profile tested. For example, one can compare the peaks in the radiation response of a 55mm deep panel (fig.11a) with the SRI dips (fig.2). Note that the same modeshapes are responsible in the asymmetrical profile measured (figs.6 and 11b). The procedure described was also applied to mode shapes "measured" on certain profiles using the SDOF approximation (figs.5,6). The results were virtually identical to those derived from finite element synthesized shapes. All of this suggests that sub-critical inter-cell cancellation plays a major role in determining the sound reduction of a profiled panel.

The introduction of profile stiffeners is known to cause significant deterioration in the overall sound reduction of cladding [1]. Using FE analysis and the simple radiation model, it is seen that the additional bends in the plate metal enforce more irregular modeshapes; i.e. adjacent "cells" are of uneven strength such that near-field cancellation is minimised.

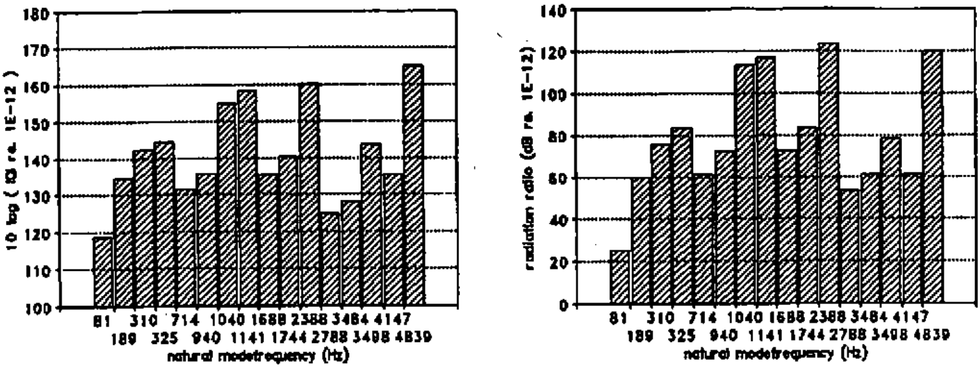


Figure 10 : Magnitude of summed vector volume velocity (a) and the corresponding comparative radiation ratio (b) of the symmetrical profile (pitch = 250mm, depth = 35mm, gauge = 0.65mm) approximated using the single simple source model on FE mode shape data.

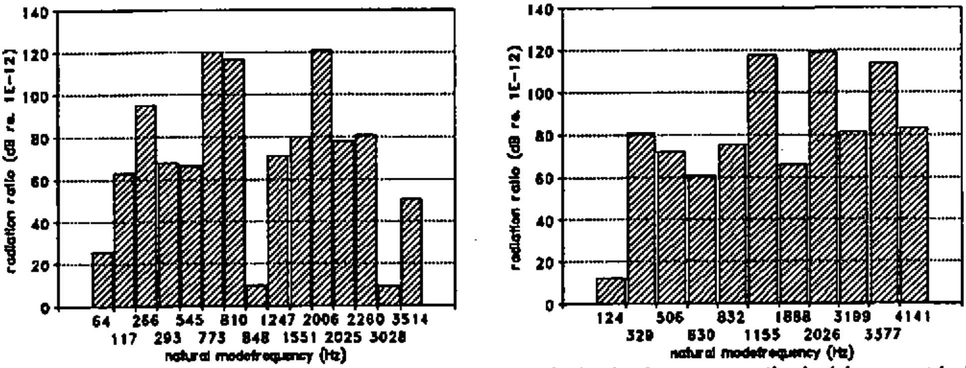


Figure 11 : Comparative radiation ratio calculated using the single simple source method : (a) symmetrical profile (as fig.10 except depth = 55mm), (b) asymmetrical profile with 20mm valley (as in fig.6).

One may also observe that the magnitude of radiation predicted for the "efficient modes" consistently varies in line with the observations made in sound reduction tests (compare the radiation ratio at peak in figs.10b and 11a to the magnitude of transmission loss dips in fig.2). This correlation was observed to hold for all modes as any of the cladding parameters were varied. However, characterising profiled cladding with one point source is rather a large approximation. It should be noted that the assumption of low frequency is not fully met for higher order modes because the air wavelength becomes comparable or even smaller than the section of profile considered; e.g. at 5KHz, $\lambda \approx$ 7cm compared to a typical profile pitch of 25cm. In this instance one might not expect full cancellation between non-adjacent cells, as is inherently assumed in the theory. More rigorous analysis (e.g. 2-D integral equation solutions) will be carried out in the near future.

5. CONCLUSION

Large depressions in sound transmission are known to occur over the whole frequency range which typically characterizes industrial noise. This paper has established that the "dips" are related to specific modal vibrations. Hence, current prediction techniques are manifestly unsuitable because they are "global" approaches unable to account for localised vibrational effects. The occurrence of the modes which cause sound transmission dips may be accurately predicted by employing a two-dimensional finite element model. An empirical formulation of this method was shown to be useful in forecasting the frequency at which this occurs. Rudimentary evaluation of the mechanical to acoustic coupling of such modes has shown that near-field volume velocity cancellation normally results in little far-field radiation. However, profiling the metal panels causes certain mode shapes for which a large amount of residual energy exists, resulting in efficient sub-critical radiation and "dips" in the observed sound reduction. However, more rigid theoretical analysis is required before a direct prediction of sound transmission can be made from these results.

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7. ACKNOWLEDGEMENTS

This work is funded by and published with the permission of H.M. Department of the Environment, directed by the Building Research Establishment, Watford. The authors also wish to express their thanks to British Steel Corporation, Strip Products for their help with samples and measurement.