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TRANSMISSION LOSS PROPERTIES OF HONEYCOMB/CONVENTIONAL METAL PANELS

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ABSTRACT

Honeycomb panels are being more widely used in helicopter structures, but they are known to have poor acoustic properties in terms of transmission loss and radiation efficiency. This is because, unlike conventional aluminium skin/stringers panels, the coincidence frequency of honeycomb panels falls within the frequency range of interest. By suitable choice of panel parameters i.e. core thickness, core shear modulus, skin thickness, etc., it is possible to optimise on the honeycomb panel design so that the coincidence frequencies are shifted to higher values and the acoustic properties return to those associated with conventional panels.

1. INTRODUCTION

Honeycomb is being increasingly used on helicopter structures because of its low weight, high stiffness properties leading to considerable savings in production costs. Recent helicopter designs at Westland Helicopters Ltd. (WHL) have incorporated honeycomb panels for the cabin roof structure and from general noise design principles this is beneficial since a flat continuous roof can be formed between the gearbox noise source and the receiver. Unfortunately, however, honeycomb materials have undesirable acoustic properties compared with conventional skin/stringer panels and this is particularly important in the cabin roof area, since they are subjected to both acoustic (airborne) and vibrational (structure borne) excitation from the gearbox.

The acoustic disadvantages of honeycomb materials are a low transmission loss and a high radiation efficiency. This is basically due to the increased thickness of the panel giving a low coincidence frequency so that the characteristic troughs and peaks of the transmission loss and radiation efficiency curves respectively fall in the middle of the frequency range of interest.

Optimisation of the honeycomb parameters to shift the coincidence frequency back to high values is possible and recent studies at WHL and the Royal Aircraft Establishment, Farnborough, (RAE) have concentrated on designing honeycomb panels for helicopter cabin structures which have similar acoustic properties to conventional skin/stringer structures. RAE have concentrated on radiation efficiency studies whilst WHL have been concerned with transmission loss. In both cases, suitably optimised honeycomb panels have been manufactured and tested to provide experimental data for comparison with theory.

It is also possible to improve the acoustic properties of honeycomb panels by keeping the coincidence frequency low and applying damping to increase the transmission loss in the frequency range above coincidence. This work is being studied by other investigators, but is briefly mentioned in this paper because of its related interest.

This paper examines the coincidence frequency theory for honeycomb panels and its effect on transmission loss. The optimisation process is discussed and comparisons of measured transmission loss data are made between honeycomb and conventional panels. Radiation efficiency is also briefly mentioned but this aspect is covered more fully by RAE in reference 1.

2. EFFECT OF COINCIDENCE FREQUENCY ON TRANSMISSION LOSS AND RADIATION EFFICIENCY

Transmission loss and radiation efficiency are parameters which give a measure of the response of panels to acoustic and vibrational excitation respectively. Transmission loss determines the noise reduction through the panel when it acts as a barrier between the noise source and the receiver. Radiation efficiency is a measure of the noise radiated by the panel when it is excited vibrationally. Since a panel radiates noise whether it is excited acoustically or vibrationally, transmission loss and radiation efficiency are related to each other.

The speed of bending waves in a panel increase with increasing frequency until at a particular frequency the speed is equal to the speed of sound in air. At this coincidence frequency the bending wave couples very efficiently with the surrounding air so that the panel becomes almost transparent to sound and the transmission loss curve falls well below the mass law relationship. In addition, the panel becomes a good radiator of sound because above the coincidence frequency the bending waves become 'acoustically fast' and the panel radiates from its whole area instead of just the edges or corners. Idealised transmission loss and radiation efficiency curves are given in figures 1 and 2 respectively. These show that the position of the coincidence frequency is very important since it divides the TL curve into the mass controlled and damping controlled regions (fig. 1) and divides the radiation efficiency curve into perimeter controlled and area controlled regions (fig. 2).

3. COINCIDENCE FREQUENCY FORMULAE

The speed of bending waves in a panel is given by

$$C_B = \omega^{\frac{1}{2}} \left(\frac{B}{M}\right)^{\frac{1}{4}} \qquad - (1)$$

where ω = angular frequency
B = bending stiffness
M = mass/unit area

and for an isotropic plate B is independent of frequency and given by

$$B = \frac{E}{(1-\sigma^2)} \frac{a^3}{12} \qquad - (2)$$

where E = Young's modulus
σ = Poisson's ratio
a = thickness of plate

Combining these two equations at $C_B = C_o$ = speed of sound in air, gives the well known coincidence frequency formula

$$f_c \simeq \frac{C_o^2}{2\pi} \sqrt{\frac{M}{B}} = \frac{C_o^2}{2\pi} \sqrt{\frac{12M}{Ea^3}} \qquad - (3)$$

which varies with the thickness of the panel.

For honeycomb panels, however, (figure 3(a)), the bending stiffness varies with frequency because both bending and shearing of the core layer take place. At low frequencies the core (of thickness b) merely acts as a spacer between the two skins (each of thickness a) which bend as shown in figure 3(b) so that the coincidence frequency becomes

$$f_c \approx \frac{Co^2}{2\pi b}\sqrt{\frac{2M_T}{aE}} \qquad - (4)$$

where M_T is total mass/unit area and E is Young's modulus of skin. Shearing of the core must also be considered, however, (Figure 3(c)), leading to a shift in the coincidence frequency to higher values.

Before proceeding with the theory, however, it should be noted that this paper is only concerned with the flexural (antisymmetric) mode of vibration (figure 3(d)). References 2 - 7 show that dilatational (symmetric) bending (figure 3(e)) is important in sandwich panels consisting of soft cores and heavy faces (e.g. hardboard skins and foam cores) giving additional coincidence frequencies and resonant frequencies in the range of interest. It is not known how important these modes are in honeycomb panels of the thicknesses considered here and is an area requiring further study.

The theory for the flexural modes is pursued in reference 8 which derives an equation for the flexural wave speed c_B in sandwich panels as

$$M_T Gb \left(\frac{c_B}{C_o}\right)^6 + \frac{\omega^2 M_T}{C_o^2}(B_C + 2B_1)\left(\frac{c_B}{C_o}\right)^4 - \frac{B_c Gb\omega^2}{C_o^4}\left(\frac{c_B}{C_o}\right)^2 - \frac{2B_1 B_c \omega^4}{6} = 0 \qquad - (5)$$

where B_c = composite bending stiffness of panel = $\dfrac{E_1 b^2 a}{2(1-\sigma^2)}$

B_1 = skin bending stiffness = $\dfrac{E_1 a^3}{12(1-\sigma^2)}$

G = shear modulus of core

Solving equation (5) shows that the total bending stiffness varies with frequency between two limits (figure 4). At lowfrequenciesthe core acts as a spacer and couples the stresses in the two skins giving the maximum value

$$B_T = B_c = \frac{E_1}{(1-\sigma^2)}\frac{b^2 a}{2} \qquad - (6)$$

At intermediate frequencies B_T decreases ($\propto \frac{1}{\omega^2}$ approx) and is controlled by the propagation of shear waves in the core. At high frequencies B_T is the sum of the two skin stiffnesses bending independently

i.e. $$B_T = 2B_1 = \frac{E}{(1-\sigma^2)}\frac{a^3}{6} \qquad - (7)$$

These upper and lower limits of the bending stiffness, therefore, control the response of honeycomb panels.

Solving equation (5) for $c_B = C_o$ gives the coincidence frequency of a honeycomb panel. This is a complicated expression given by

$$f_c = \frac{C_o}{2\pi}\left\{ \frac{2C_o^2\, Gb\, M_T}{-\left\{C_o^2 M_T(Bc+2B_1)-B_c\, Gb\right\} \pm \left\{\left[C_o^2 M_T(Bc+2B_1)-B_c\, Gb\right]^2+8c_o^2 M_T GbB_1B_c\right\}^{\frac{1}{2}}} \right\}^{\frac{1}{2}} \quad -(8)$$

which can generally be simplified for $2B_1 \ll Bc$. The important term in the denominator is $C_o^2 M_T(Bc+2B_1)-B_c Gb$ which for $2B_1 \ll Bc$ becomes $-B_c\left(1-\frac{M_T C_o^2}{Gb}\right)$ and thus the coincidence frequency varies with the parameters of the honeycomb panel according to whether $\frac{M_T C_o^2}{Gb}$ is greater or less than 1. If $\frac{M_T C_o^2}{Gb}$ is > 1 then a high coincidence frequency is obtained and if $\frac{M_T C_o^2}{Gb} < 1$ then a low coincidence frequency is obtained. Thus, it is possible to optimise on the design of the honeycomb panel to shift the coincidence frequency to higher values outside the gearbox excitation range.

4. OPTIMISATION OF HONEYCOMB PARAMETERS

The term $\frac{M_T C_o^2}{Gb}$ can be rewritten as $\frac{\rho_c C_o^2}{G}\left(1+\frac{M_{2s}}{M_c}\right)$ where

ρ_c = density of core

M_{2s} = mass/unit area of 2 skins

M_c = mass/unit area of core

and thus for $\frac{M_T C_o^2}{Gb} > 1$ then $\frac{M_{2s}}{M_c} > \left(\frac{G}{\rho_c C_o^2} - 1\right)$. At this point it is interesting to examine some typical honeycomb panel materials as used in the helicopter airframe construction in table 1. For a nomex core, $\left(\frac{G}{\rho_c C_o^2} - 1\right) \simeq 8$ which means that $\frac{M_{2s}}{M_c}$ must be near to the value of 8, which is quite possible by suitable choice of parameters. For aluminium honeycomb core, $\left(\frac{G}{\rho_c C_o^2} - 1\right)$ is typically 43 and a value of $\frac{M_{2s}}{M_c}$ of 43 is not possible. Thus optimisation can only be carried out on a nomex cored panel. In a similar manner the choice of skin materials can be considered. For high coincidence frequency, M_{2s} must be as high as possible which means choosing a high density skin material such as aluminium in preference to other possibilities such as fibreglass, carbon or kevlar.

Considering, therefore, a panel constructed of aluminium skins and nomex core, the optimisation possibilities are shown in figure 5. The figure is divided into two parts; the first part shows the variation of $\frac{M_T C_o^2}{Gb}$ with core thickness and core shear modulus for a constant skin thickness of 0.3 mm and the second part then gives the additional variation with skin thickness for a given core shear modulus and thickness. Figure 5 only shows values of $\frac{M_T C_o^2}{Gb}$ up to 1.0 but values above 1.0 are readily obtainable with suitable design of the panel parameters.

Nomex cored panels often considered for helicopter structures have the following parameters:-

Skin thickness 0.3 mm
Core thickness 12 mm
Core density 32 kg/m^3
Core shear modulus 35 x 10^6 $newton/m^2$ (longitudinal)

Reference to fig 5 shows that such a panel only has a $\frac{M_T c_o^2}{Gb}$ value of 0.6 and hence a low coincidence frequency of approx.1000Hz. Values of $\frac{M_T c_o^2}{Gb} > 1$ are obtained with a very low core thickness, a very low core shear modulus and a high skin thickness. Such panels have obvious disadvantages from the structural design point of view in terms of increased panel weight (from the high skin thickness) and a core with low structural integrity. Compromises have to be made between the structural and acoustic properties and a suitably optimised panel is considered to be:-

Skin thickness 0.5 mm
Core thickness 10 mm
Core density 24 kg/m^3
Core shear modulus 26 x 10^6 $newton/m^2$ (longitudinal)

In fact, figure 5 shows that a 0.4 mm skin would probably be sufficient but a 0.5 mm skin has been chosen for conservative reasons.

One point to notice in the optimisation theory is that the core shear modulus decreases as the core density decreases but the theory assumes that the core is isotropic, when, in fact, the core is othrotropic giving different shear moduli in different directions. Reference 9 studies the effect of orthotropy on the transmission loss of plywood panels and shows that the coincidence dip broadens out over a fairly wide frequency band. It seems reasonable to assume that a similar effect will occur with honeycomb panels with the width of the trough dependent on the variation in core shear modulus. Since,however, the longitudinal shear modulus is generally greater than the lateral value then, providing the former is optimised (i.e. made as low as possible) then the latter also becomes optimised since both the longitudinal and lateral shear moduli decrease in a similar manner with core density.

5. TRANSMISSION LOSS THEORY

The transmission loss (TL) of a panel can be calculated using the Statistical Energy Analysis method in which the energy flow between coupled systems is considered, taking into account panel stiffness and damping. The theory, derived by Crocker and Price (10) and Heron (11), will not be reproduced in this paper but can be found summarised in reference 12. The final formula is

$$TL = -10 \log_{10}\left[\left(\frac{\rho c_o}{\omega M}\right)^2 4\sqrt{10}\right] - 10 \log_{10}\left[\left(\frac{\rho c_o}{\omega M}\right)^2 \frac{8\pi^2 c_o^2}{A\omega} \; e_r^2 \frac{n_p}{\eta_{PT}}\right] \quad - (9)$$

where ρ_0 = density of air
M = mass/unit area of panel
e_r = radiation efficiency ratio of panel
ω = angular frequency
A = area of panel
n_p = model density of panel
η_{PT} = total damping loss factor of panel

The purpose of showing this equation is to comment on the terms affecting the transmission loss. The first term in equation (9) represents the non resonant transmission loss or well known mass controlled contribution and the second term is the resonant controlled contribution. For conventional skin/stringer panels the transmission loss over the frequency range of interest is almost entirely mass controlled and the resonant part only contributes near or above the coincidence frequency dip. Obviously, as the coincidence frequency decreases the resonant controlled transmission loss becomes more important and increases with increasing damping, η_{PT}, so that the coincidence dip is smoothed out. Thus for honeycomb panels with low coincidence frequencies, the damping of the panel is also important (see section 6).

The other parameter of interest in equation (9) is the modal density n_p of the panel. For a flat metal panel the modal density is independent of frequency, but for a honeycomb panel it varies with frequency (13) according to the expression

$$n_p(\omega) = \frac{A}{4\pi S h_1^2} \frac{\Omega^2}{\omega} \left(1 + \frac{\left[\Omega^2 + 2(1-M^2)S^2\right]}{\left[\Omega^4 + 4(1-M^2)S^2\Omega^2\right]^{\frac{1}{2}}} \right) \qquad - (10)$$

$$\text{for } S = \frac{G h_1}{E h_2} \text{ and } \Omega^2 = \frac{4\pi^2(\rho_1 h_1 + \rho_2 h_2)\omega^2 h_1^2}{E h_2} \qquad - (11)$$

where h_1 = half thickness of core
h_2 = total thickness of skins
ρ_1 = density of core
ρ_2 = density of skins

This formula is an approximation since it does not take into account the bending stiffness of the two skins with the result that the model density increases continuously with frequency. It is clear, however, that the above formula is once again dependent on the core thickness, core shear modulus and skin thickness of the honeycomb panel and as shown in figure 6 the modal density decreases (and hence TL increases) with increasing core thickness and core shear modulus. Thus the model density effect counteracts to a certain extent the benefits gained by the optimisation process described in section 3. The significance of this will become clearer when the experimental results are compared in section 8.

6. EFFECT OF DAMPING

As mentioned in the previous section, damping is important in the region near and above coincidence frequency. For example, the SEA theory for transmission loss shows that a factor of 10 increase in damping will give a 10 dB increase in the resonant component of the transmission loss with a corresponding change in the total transmission loss around coincidence. Cremer (14) predicts a 9 dB/octave increase in transmission loss of a homogenous plate with frequency well above coincidence (compared with 6dB/octave below coincidence) with the formula

$$\mathrm{TL} = 20 \log \frac{\omega M}{2\rho c_o} + 10 \log \left(\frac{\omega}{\omega_c} d\right) - 3 \text{ dB} \qquad - (12)$$

where d is the internal damping loss factor and ω_c is the coincidence frequency. These equations lead to another form of optimisation of honeycomb panels in which the coincidence frequency is kept below the frequency range of interest (instead of above it) and damping is then applied to give a high transmission loss above coincidence. Such a procedure is described by Mead (15) and Meier (16). In Mead's theoretical work (15) aluminium honeycomb core panels are considered (since they have a lower coincidence frequency than nomex) and consideration is given to reducing the thickness of the skins so that damping can be added in the form of unconstrained or constrained damping layers and even constructing a damping layer between two sandwich plates. In this way damping loss factors of about 0.1 can be obtained. In Meier's work (16) the top skin is replaced completely by a damping layer and the transmission loss of the complete panel rises above mass law by choosing a suitable combination of panel parameters. The mass law equation for field coincidence is

$$\mathrm{TL} = 20 \log \frac{\omega M}{2\rho c_o} - 6 \text{ dB} \qquad - (13)$$

and it is found that, by equating the right hand sides of equations (12) and(13), there is a transition frequency $\omega_t = \frac{\omega_c}{2d}$ where the transmission loss of the plate reverts back to mass law. Thus for a low coincidence frequency and a high value of damping, a high transmission loss can be obtained over a wide frequency range. Typical theoretical data taken from Meier's paper is given in figure 7, and, although this applies to heavier panels than considered in this paper, it shows how the shear modulus should exceed approximately 2×10^8 N/m^2 i.e. an aluminium honeycomb type core is required. This is in contrast to section 4 where a nomex cored panel was required to improve the transmission loss.

Unfortunately, in the WHL studies damping has not yet been applied to an aluminium honeycomb core panel and so the above theory cannot be verified. Damping has, however, been applied to a nomex cored panel (with low f_c) and the results are described in section 8.

7. EXPERIMENTAL STUDY

In order to verify the optimisation theory, experimental studies have been carried out at WHL and RAE. RAE have concentrated on testing panels of varying skin thicknesses but with constant core parameters whilst WHL have mainly varied the core parameters. The range of WHL test panels (conventional and honeycomb) are shown in table 2. The 3 conventional aluminium panels (panels 1 - 3) were included for comparative purposes and show the effect of going from a plain aluminium sheet to one with stringers and frames added (i.e. a typical helicopter panel) and finally changing the single layer skin to a 2 layer bonded skin (AF10 adhesive). The honeycomb panels (panels 4 - 11) covered a wide range of $\frac{M_T c_o^2}{Gb}$ values, varying from optimised nomex cored panels (panels 4 and 6) to acoustically poor panels (panels 7, 8 and 9). Panel 9 represents the typical honeycomb structure normally used for non acoustic reasons (as mentioned in section 4). Panels 5 and 6 show changes in skin thickness and panels 7 and 8 have varying core thickness. Panel 6 has an increased top skin layer made up of the standard 0.3 mm skin plus a 0.4 mm rigidised surface layer. This provides extra strength for walking and standing on, but for the purposes of the theory, can be considered as a panel with two

0.5 mm skins. Finally two aluminium honeycomb cored panels (panels 10 and 11) were tested, one with fairly low core parameters (thickness and density) and one with high core parameters.

Each panel was approximately 1 metre square in size and was mounted in an aperture between two reverberation rooms.

Transmission loss values were obtained from acoustic excitation tests and radiation efficiency values from vibrational excitation tests.

8. RESULTS AND DISCUSSION

Since this paper is concentrating on transmission loss properties, most of the following discussion is concerned with the transmission loss data. For completeness, however, radiation efficiency values are shown for 3 of the panels.

As a reference point, the measured transmission loss data for the 3 conventional metal panels are shown in figure 8. All data below about 400 Hz should be ignored since the characteristic peaks and troughs shown are due to room effects and are common to all panels. The 3 conventional panels have masses of between 0.4 and 0.6 lb/ft^2 and thus the mass law line (or non resonant TL) of 0.5 lb/ft^2 is shown for comparison. In general all 3 panels give the same results and thus providing that the mass/unit area of the panel remains approximately the same, the addition of frames and stringers or the change to double layer skins makes very little difference to the acoustic transmission properties. This is to be expected since the panels are mass controlled but it is not clear why the measured data shows a slope of about 4 dB/octave instead of the 6 dB/octave for the mass law.

The results for the nomex cored panels are presented in figure 9 and the data in general follows the predicted trends of increasing transmission loss as the core thickness and core shear modulus decrease and the skin thickness increases. The first point to notice is that panel 9, which represents a typical structure normally used in helicopters, has a poor transmission loss particularly in the 1 kHz - 3 KHz frequency range. This compares with a predicted coincidence frequency of980Hz. Panel 4, which the optimisation theory shows to be an acoustically good panel, has a much improved transmission loss but is still lower than panel 6 with the extra rigidised surface layer. Thus, providing that the additional panel weight is permitted, panel 6 is the best panel to go for, or, alternatively a panel with 0.5 mm skins as recommended in section 4. Panel5 which has a slightly lower skin thickness and a slightly higher core thickness does not give a very good transmission loss curve (although still better than panel 9). This confirms the rapid change in results for small changes in parameters shown by figure 5 at low core shear modulus. A comparison of the results of panels 5 and 7 shows that increasing the core shear modulus at constant core thickness decreases the transmission loss as expected. In fact, panel 7 gives the worst results of all over the 1 kHz - 3 kHz range, but then increasing the core thickness at constant core shear modulus (panel 8) gives an unexpected result in that the transmission loss values of panel 8 are better than those of panel 7. At first glance this appears to contradict the theory of figure 5 but figure 6 shows that the modal density decreases (and hence TL increases) as the core thickness is increased. It is suggested, therefore, that in the comparison of panels 7 and 8 the 'modal density effect' is greater than the 'coincidence frequency effect'.

Panels 10 and 11 with aluminium honeycomb cores give vastly different results (see figure 10). Both have low coincidence frequencies and should, therefore, have poor transmission loss properties. Panel 10, however, gives a very good transmission loss particularly at high frequencies and once again it is suggested that this is due to modal density effects (see figure 6). In the light of this explanation, panel 11 results are difficult to explain. According to figure 6 the modal density is low giving a high transmission loss and the value of $\frac{M_T c_o^2}{Gb}$ although very low is not as bad as the value for panel 10. Thus one would expect panel 11 to give the same or even better transmission loss data than panel 10. One explanation could be that the extra mass of panel 10 is having a controlling effect.

Another interesting point concerning figure 10 is that the transmission loss curve for panel 10 has apprxoimately the same slope as the mass law line, whereas the nomex cored panels of figure 9 all have slopes well below mass law. The conventional metal panels of figure 8 have transmission loss slopes midway between these two extremes. No explanation can be offered for these effects at this stage.

The effect of adding damping to the nomex cored panel (panel 10) is shown in figure 11. Since a large amount of damping has been applied (the panel weight has been doubled) it is difficult to judge the true picture, but it is clear that the increase in transmission loss is greater than that given by the extra mass alone.

With regard to radiation efficiency data, the results for 4 of the panels are shown in figure 12. Optimising the nomex cored panels has improved the radiation efficiency, but only at mid to high frequencies (see panels 6 and 9 of fig 12). In fact the complete range of nomex cored panels tested only showed a 10 dB variation between them. The aluminium honeycomb cored panels, however, gave poor radiation efficiency data, being some 5 - 10 dB worse than the nomex panels. The results for panel 10 are shown in figure 12 and these are well above the conventional panel data (panel 2). This is in sharp contrast to figure 10 where the aluminium honeycomb panel 10 gave high transmission loss values.

Finally the effect of damping on the nomex cored panel (panel 9) is shown in figure 13. Adding damping decreases the radiation efficiency which is generally to be expected since the panel vibrational response will have been reduced. A full study of the effect of damping has not, however, been carried out yet.

9. CONCLUDING REMARKS

The process of optimising the core parameters has improved the transmission loss and radiation efficiency properties of nomex cored honeycomb panels. Thus the best acoustic properties are obtained with a low core thickness, a low core shear modulus and a high skin thickness. The radiation efficiency values are, however, less sensitive to changes in the nomex core parameters and even with an optimised core the values are still worse than conventional panels at low and mid frequencies. The transmission loss values show more variation with core parameters but in some cases the changes in modal density offset the benefits gained by increasing the coincidence frequency. This combination of effects is an area requiring further study since it suggests that in some cases a thicker core panel may be more desirable than a thin one.

With aluminium honeycomb cores there is a conflict of interest since good transmission loss properties can be obtained at the same time as bad radiation efficiency properties. The modal density changes play an important part but the results presented in this paper are inconclusive on this aspect.

The addition of damping appears to give promising results on both transmission loss and radiation efficiency but further work is required to distinguish between extra mass and damping effects.

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TABLE 1: RANGE OF PARAMETERS OF TYPICAL HONEYCOMB PANEL MATERIALS

Skin thickness (a)	0.3 - 0.7 mm
Skin density	2700 kg/m^3 aluminium, 1938 kg/m^3 fibreglass, 1438 kg/m^3 carbon, 1438 kg/m^3 kevlar.
Core thickness (b)	10 - 25 mm
Core density	24 - 64 kg/m^3 nomex, 20 - 80 kg/m^3 aluminium honeycomb.
Core shear modulus (G) (longitudinal)	26 - 72 x 10^6 N/m^2 nomex, 50 - 450 x 10^6 N/m^2 aluminium honeycomb

TABLE 2: PANELS TESTED

A. METAL PANELS

PANEL NO.	1	2	3
PANEL DESCRIPTION	0.7mm THICK PLAIN ALUMINIUM	0.7mm THICK Al. + FRAMES AND STRINGERS	TWO 0.3 mm THICK Al. SKINS BONDED WITH AF10 + FRAMES AND STRINGERS
MASS/UNIT AREA (MEASURED) LB/FT^2	0.4	0.5	0.6

B. HONEYCOMB PANELS

PANEL NO.	4	5	6	7	8	9	10	11
SKIN MATERIAL	Al	Al	Al	Al	Al	Al	Al	Al
CORE MATERIAL	NOMEX	NOMEX	NOMEX	NOMEX	NOMEX	NOMEX	Al HONEY COMB	Al HONEY COMB
SKIN THICKNESS (mm)	0.4	0.3	0.7top 0.3 bottom	0.3	0.3	0.3	0.3	0.3
CORE THICKNESS (mm)	10	12	12	12	24	12	20	12
CORE DENSITY (kg/m^3)	24	24	24	50	50	32	50	29
CORE SHEAR MOD ($x10^6 N/m^2$)	26	26	26	55	55	35	250	130
MASS/UNIT AREA (MEASURED) (lb/ft^2)	0.6	0.5	0.8	0.6	0.7	0.6	0.7	0.5
PREDICTED $\frac{M_T C_o^2}{Gb}$ VALUE	1.10	0.73	1.14	0.40	0.25	0.57	0.06	0.15
PREDICTED COINCIDENCE FREQ. (Hz)	above 9000	1200	above 10000	870	450	980	450	700

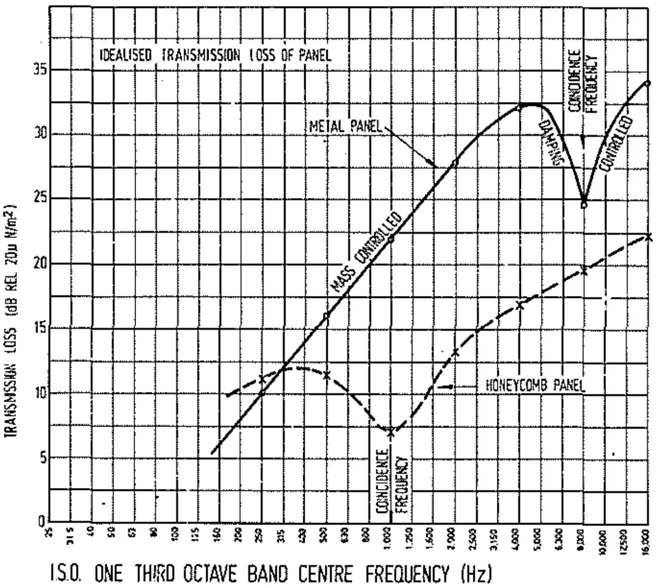

FIG. 1. IDEALISED TRANSMISSION LOSS OF A PANEL.

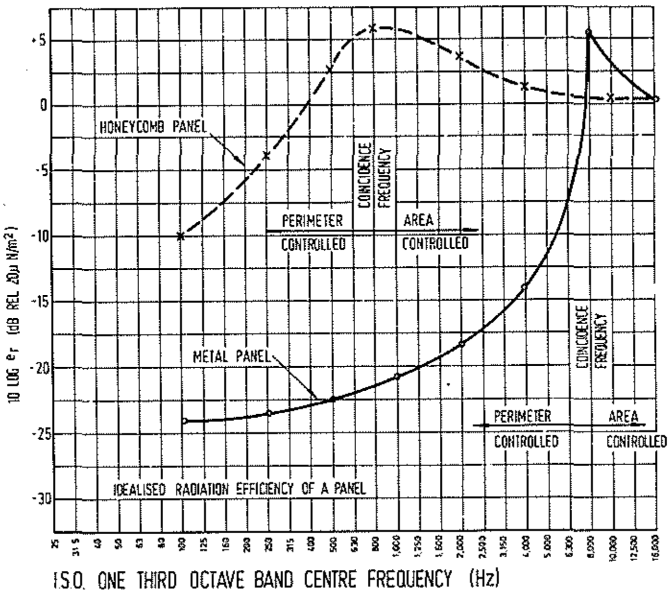

FIG. 2. IDEALISED RADIATION EFFICIENCY OF A PANEL.

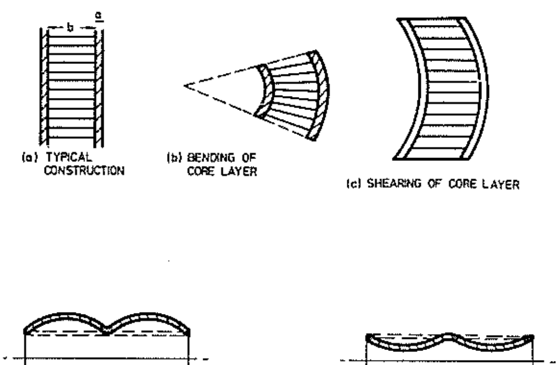

FIG 3. BENDING OF SANDWICH PANEL.

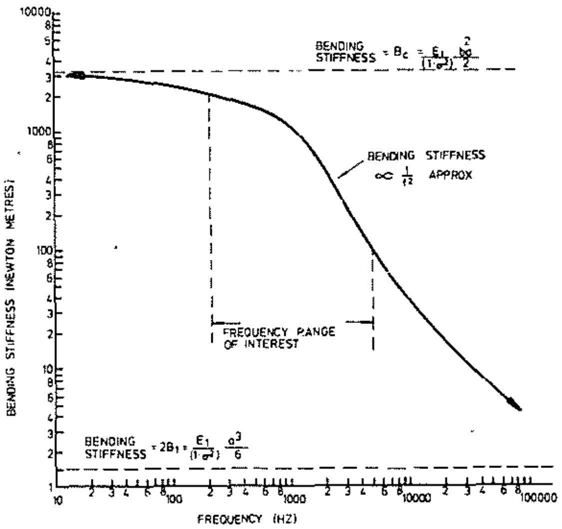

FIG 4 VARIATION OF BENDING STIFFNESS WITH FREQUENCY FOR TYPICAL HONEYCOMB PANEL

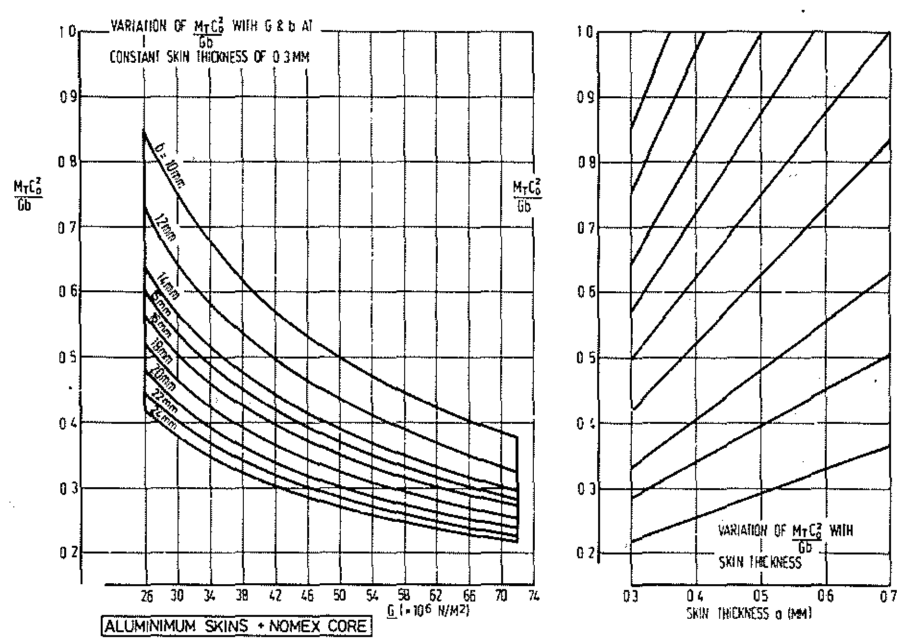

FIG 5 VARIATION OF $\frac{M_T C_0^2}{Gb}$ WITH PANEL PARAMETERS

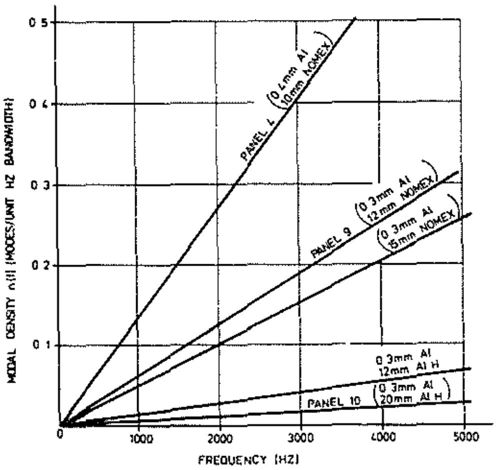

FIG 6 MODAL DENSITY VARIATION WITH FREQUENCY FOR HONEYCOMB PANELS

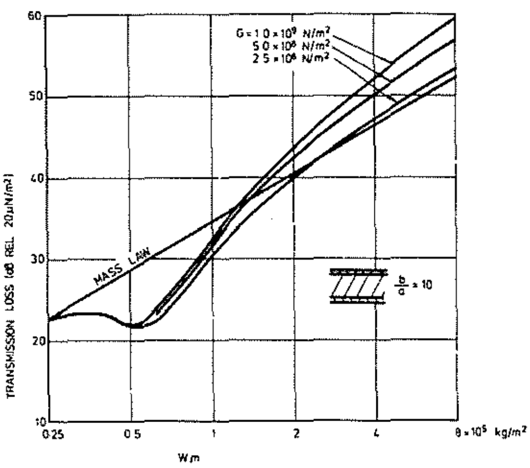

FIG 7 PREDICTED TRANSMISSION LOSS OF SANDWICH PLATE BASE PLATE ALUMINIUM (REPRODUCED FROM REF 15)

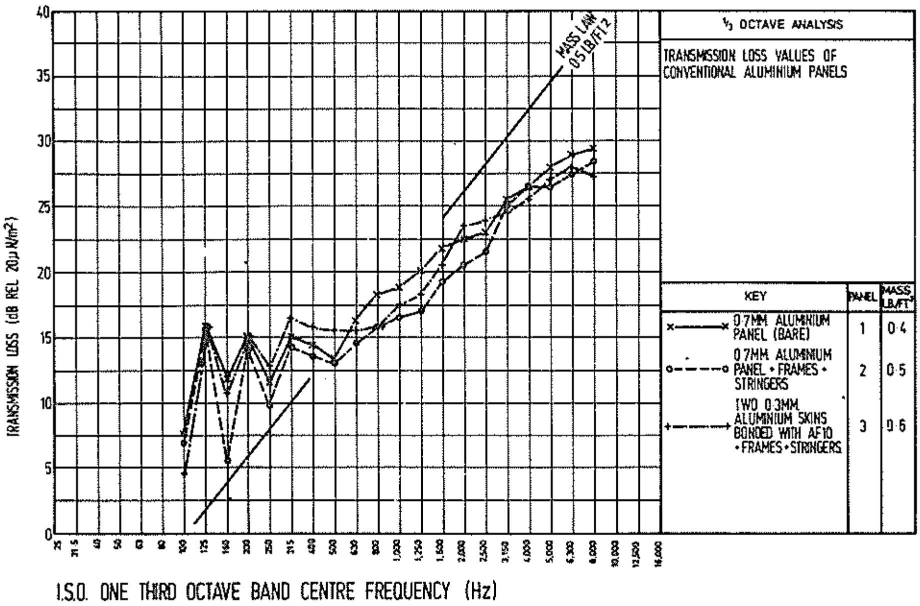

FIG. 8 TRANSMISSION LOSS VALUES OF CONVENTIONAL ALUMINIUM PANELS.

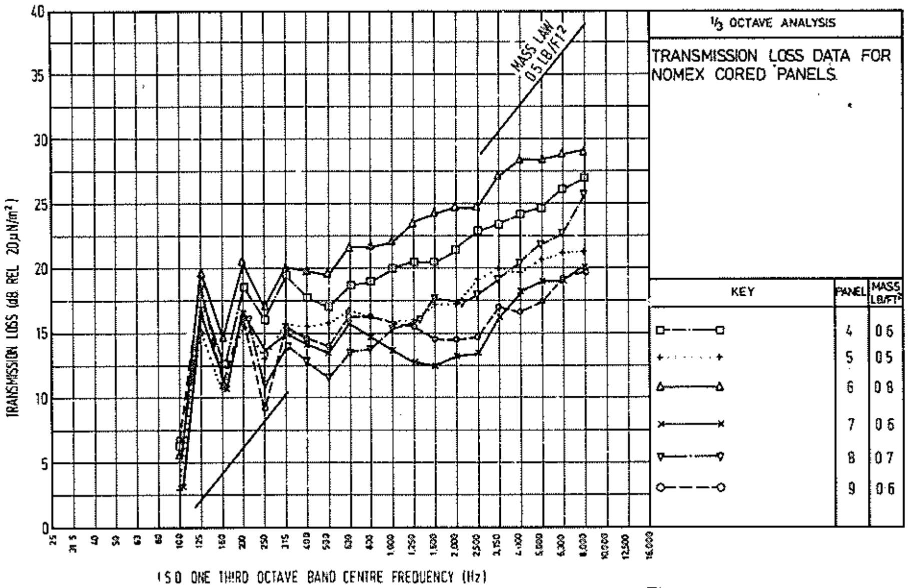

FIG. 9 TRANSMISSION LOSS DATA FOR NOMEX CORED PANELS.

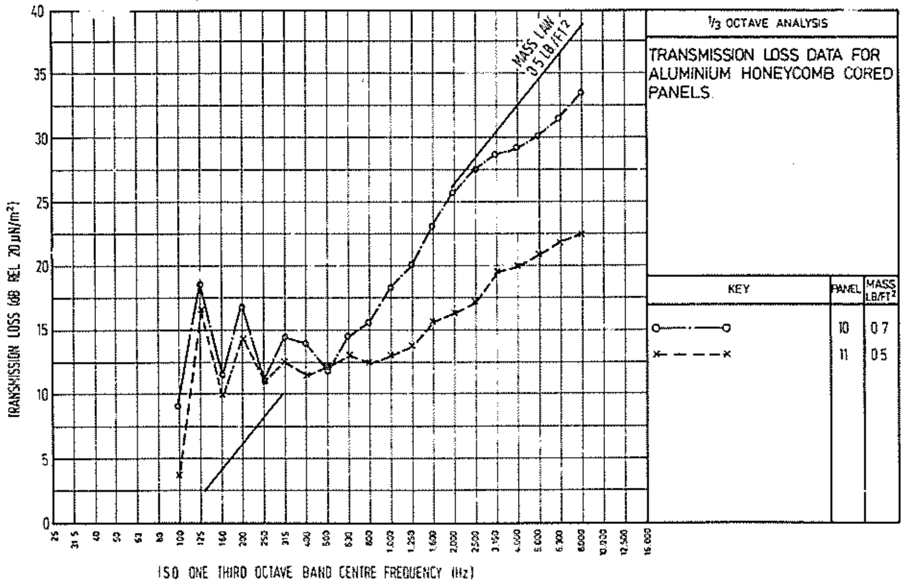

FIG. 10 TRANSMISSION LOSS DATA FOR ALUMINIUM HONEYCOMB CORED PANELS.

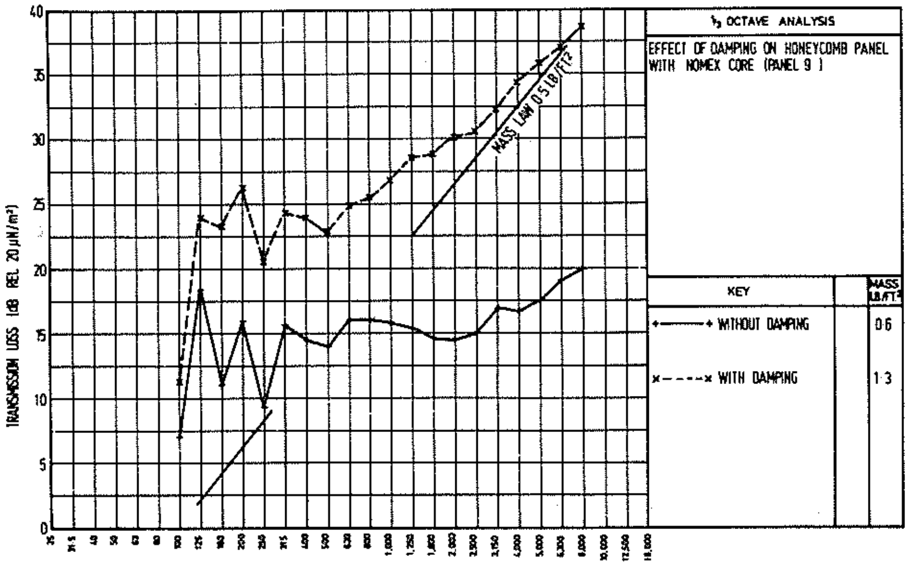

FIG 11 EFFECT OF DAMPING ON TRANSMISSION LOSS OF HONEYCOMB PANEL WITH NOMEX CORE (PANEL 9)

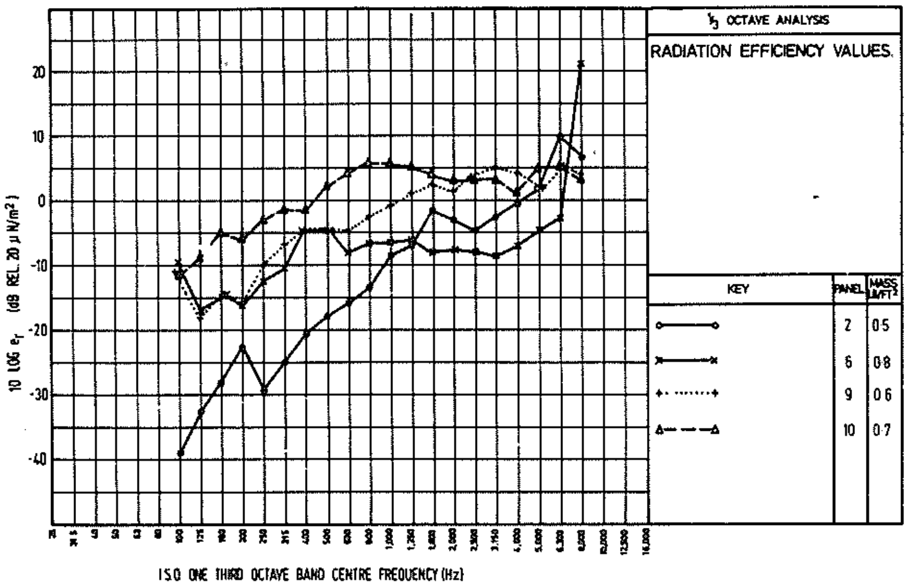

FIG 12 RADIATION EFFICIENCY DATA OF PANELS

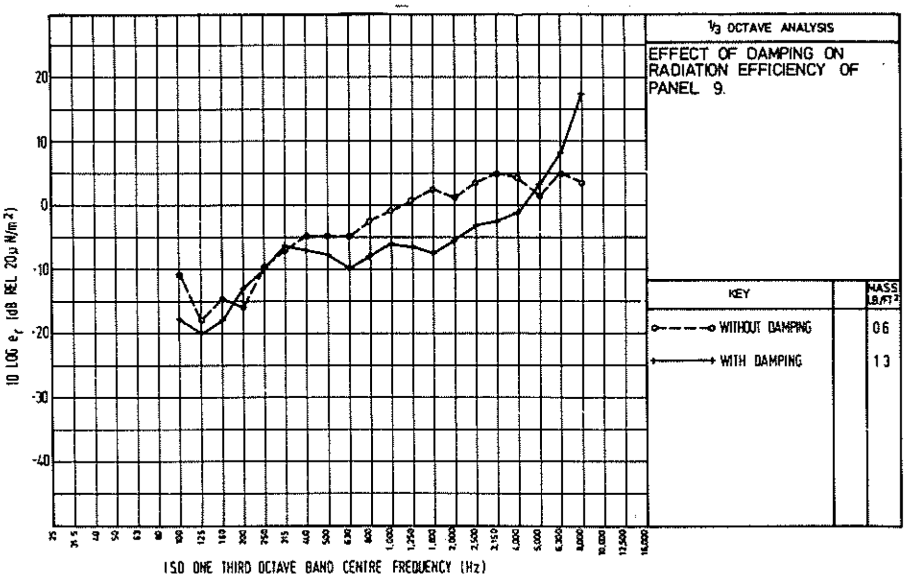

FIG. 13 EFFECT OF DAMPING ON RADIATION EFFICIENCY OF HONEYCOMB PANEL WITH NOMEX CORE (PANEL 9)