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Performance of Polyurethane Cored Carpet Composites for Vehicle Interior Noise Control

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1. Introduction

Moulded foam backed carpet composites are now widely established in the UK motor industry as noise control components. There has been little information published on the measurement and evaluation of their performance. This paper reports on the initial findings of a study of the behaviour of polyurethane foam cored systems. Data is presented for the system insertion loss and the system frequency response for a number of PUR foams.

The transmission loss of multiple element constructions is well documented by many authors (Beranek and Work [2], Mangiarotty [8], Au and Byrne [1]. Their theoretical analysis is based upon the impedance of the component layers. The core isolating layer being a fibreglass blanket. Other authors have used viscoelastic materials for vibration damping purposes (Oberst [3]) and in multiple element systems (Reddy et al [6]). Their experimental and theoretical treatment of the subject used bonded systems. Despite the trend towards unbonded carpet systems within vehicles, the only published work known to the authors is that of Satter and Ahmadi [5]. They presented data for the radiation and response of a circular plate with unbonded PUR foam layers to structure borne vibration.

2. Theoretical Treatment

A low density material when applied to a vibrating surface moves with a motion very similar to that of the surface, at low frequencies. This renders the contribution of the applied layer to the insertion loss approaching zero. The crossover frequency, above which the wavelength of the radiated sound and layer thickness are more comparable is given by Schultz [9];

$$f_s = c_0/(10*h) \qquad \langle 1 \rangle$$

Schultz also gives the expressions for IL as

$$IL = 0 \qquad f<f_s \qquad \langle 2a \rangle$$
$$= 8.69ah \qquad f>f_c \qquad dB \qquad \langle 2b \rangle$$

The attenuation constant for porous foams is a modified version of that given by Qunli [11] as;

$$a = 0.163(0.25*1.21*f/R_1)^{-0.592} \qquad \langle 3 \rangle$$

The three layer system used for vehicle carpet treatments consists of a steel floorpan, covered with a PUR foam, on top of which is a heavy or septum layer. The Transmission loss of the this three layer foam cored composite can be derived, [2], from equations <4> to <8> .

$$1/T = \{ x_1\cosh(bd) + x_2 \sinh(bd) \}^2 \qquad \langle 4\rangle$$

where $x_1 = 1 + jw(m_{s1}+m_{s2})/z_0$ <5>

$x_2 = Z_1/z_0 - w^2 m_{s1} m_{s2}/(Z_1 z_0)$ <6>

and Transmission Loss, $R = 10\mathrm{LOG}(1/T)$ <7>

Substituting <4> and <5> into <3>;

$$(1/T)^{1/2} = [\, 1+jw(m_{s1}+m_{s2})/z_0]\cosh(bd)$$

$$+[Z_1/z_0+jwm_{s1}(1+jwm_{s2}/z_0)/Z_1]\sinh(bd) \qquad \langle 8\rangle$$

Using the expressions for propagation constant and material available [10] the resonant frequency of the system can then be derived from <7> as;

$$f_{res} = \frac{1}{2\pi}\left[\frac{E}{h}\left(\frac{1}{m_{s1}} + \frac{1}{m_{s2}}\right)\right]^{1/2} \qquad \langle 9\rangle$$

The insertion loss of the added foam and septum can be readily calculated from equation <9>. For modelling purposes we assume that R_{panel} is given from the limp mass law.

$$IL = R_{system} - R_{panel} \qquad \langle 10\rangle$$

3. Experimental Test Rig and Measurement

The test-rig used for experimental work is illustrated in Figure 1. It has dimensions of approx. $1m^3$, with a aperture of 0.87m x 0.87m (the size of commercially available moulded PUR foam). A 1 mm mild steel sheet is clamped rigidly onto the top surface. Structural vibration is produced by an electrodynamic shaker, and airborne excitation via a loudspeaker within the test-rig cavity. The PUR foam is placed upon the panel and a septum layer upon it, if required. Care must be taken to ensure that the system is well sealed from leakage.

Intensity measurements were made by sweeping the intensity probe above the surface of the top layer. This was carried out for the bare panel and also with the composite in place. The insertion loss of the foam composite system could then be obtained by subtracting the two measurements. Vibration response measurements were carried out with miniature accelerometers on the plate surface.

4. Results

The insertion loss for the foams in Table 1 was obtained using the procedure outlined in Section 3 above. A septum layer of 3.2 kgm^-3 was used. Material properties are detailed in table 1, Appendix 1.

The experimentally determined insertion losses for the plate + PUR foam are shown in figures 4 and 5. The insertion losses for the three layer system are shown in figures 6 and 7. The cross-over frequency, f_c, can be obtained from figures 4 and 5. These are compared with the values predicted by equation <1> in table 2. Likewise, for the three layer system, the resonant frequency of the minima can be obtained from figures 6 and 7. These are compared with those predicted by equation <9> in measured values of resonant frequencies with those predicted by equation <9> in table 3.

The vibration response for foam A is shown in figure 2.

5. Discussion

Figure 2 illustrates the effect of an unbonded foam layer (no septum) on the vibration response of the steel panel. It is seen that the layer reduces the intensity of the modal response. This effect is enhanced by the addition of the septum, and with foams of relatively high damping (approx. 0.3) the modal response is eliminated The effective loss factor, as measured by the decay method, is shown in figure 3. It is seen that the general characteristic is one of decreasing damping with increasing frequency. This behaviour agrees with that previously reported by Reddy et al [6] for bonded systems. Placing the foam layer on the panel increased the system loss factor by an order of magnitude.

The predictions by Schultz for the airborne noise insertion loss of a panel + foam layer system is that the IL is zero where the wavelength of the incident wave is larger than ten times the foam thickness. Figures 4 and 5 demonstrate that this behaviour does occur in practice. Table 2 shows that the predictions of this crossover frequency correlate reasonably well with those measured in practice. The predicted IL above the crossover frequency does not agree so well with the measured values, although the trends are predicted correctly. This could be attributed in part to the differences in flow resistance of the foams layers.

The introduction of a septum layer effectively produces a damped double leaf partition with viscoelastic foam core. Around the minimum in the response the measured IL values are in reasonable agreement with those predicted from equations <8> and <10>. The resonance frequencies predicted by equation <9> using measured values of material properties agree with those found in practice, as seen from table 3. Although the overall response found experimentally is in general agreement with with that predicted by equations <8> and <10>, at frequencies away from resonance the quantitative agreement is poor.

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The data of figure 4 and equation (8) shows that for a given panel the frequency of the resonance trough can be controlled through the thickness and modulus of the foam core and the area mass of the septum layer. The depth of the trough is influenced by the loss factor of the foam, as expected from simple analysis of damped systems.

6. Conclusions

The work reported in this paper has demonstrated some of the fundamental material properties that affect the IL performance of unbonded PUR foam cored systems. A theoretical approach has been introduced, preliminary results presented, and these compared with those obtained experimentally.

7. References

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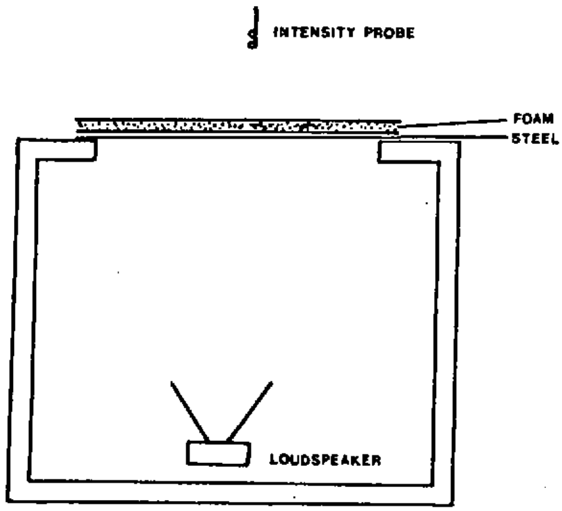


Figure 1 Small Scale Test-Rig

FIGURE 2 FREQUENCY RESPONSE

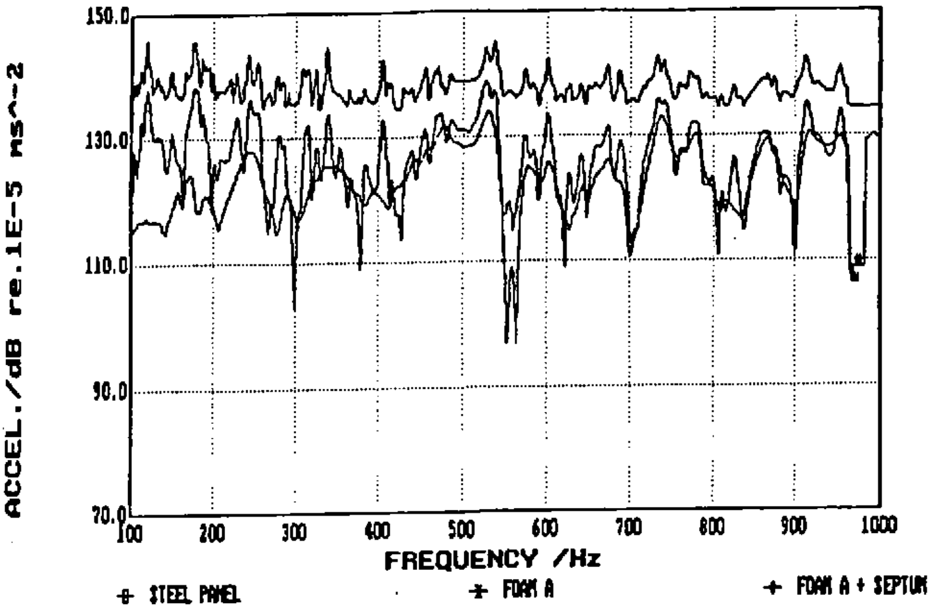


FIGURE 3 LOSS FACTOR

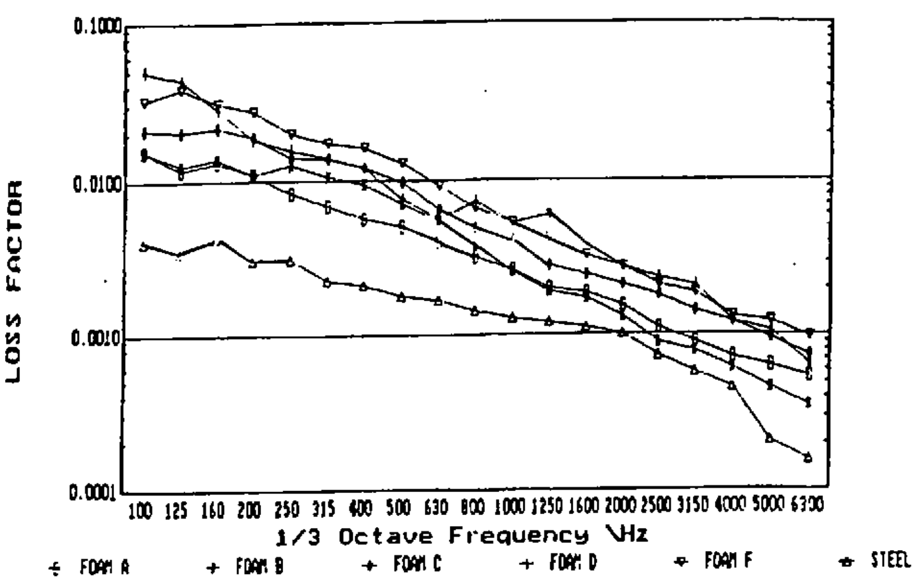


FIG. 4 Insertion Loss

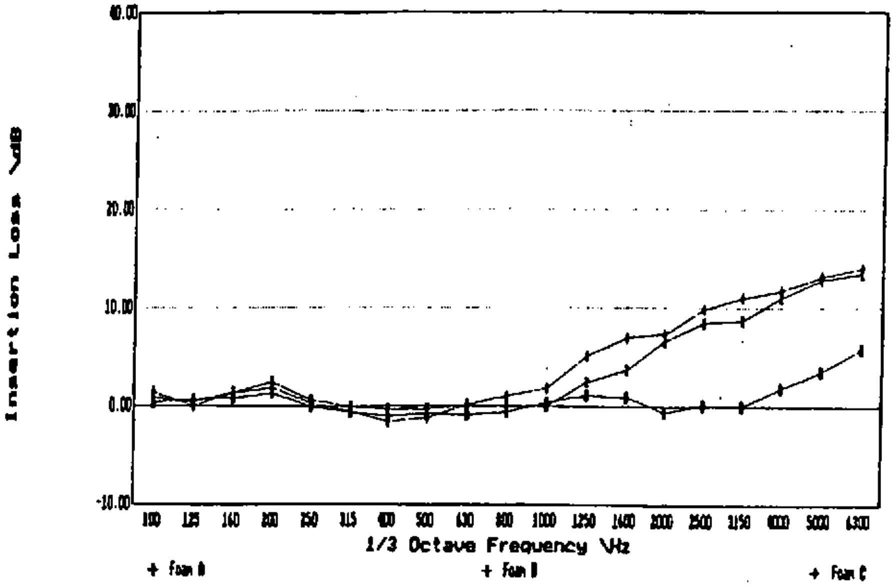


FIG. 5 Insertion Loss

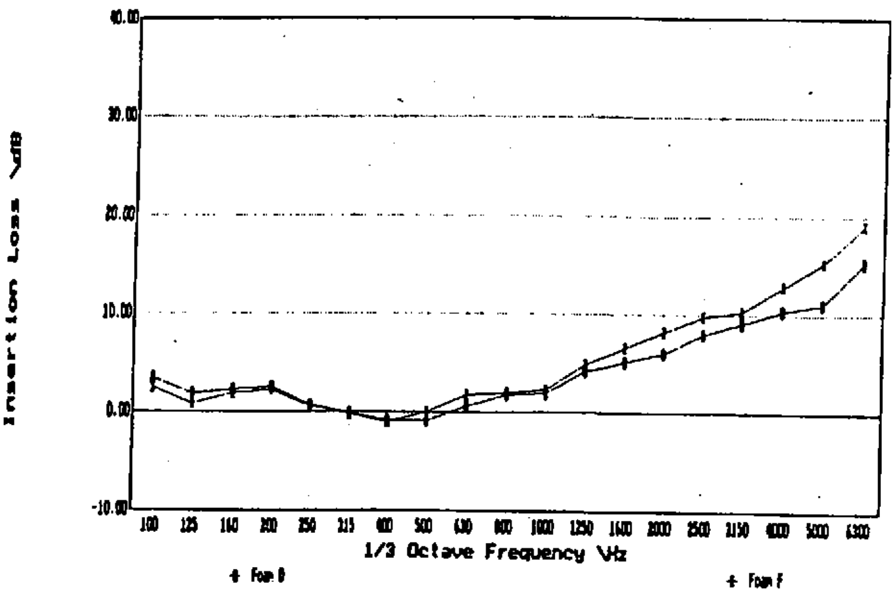


FIG.6 Insertion Loss

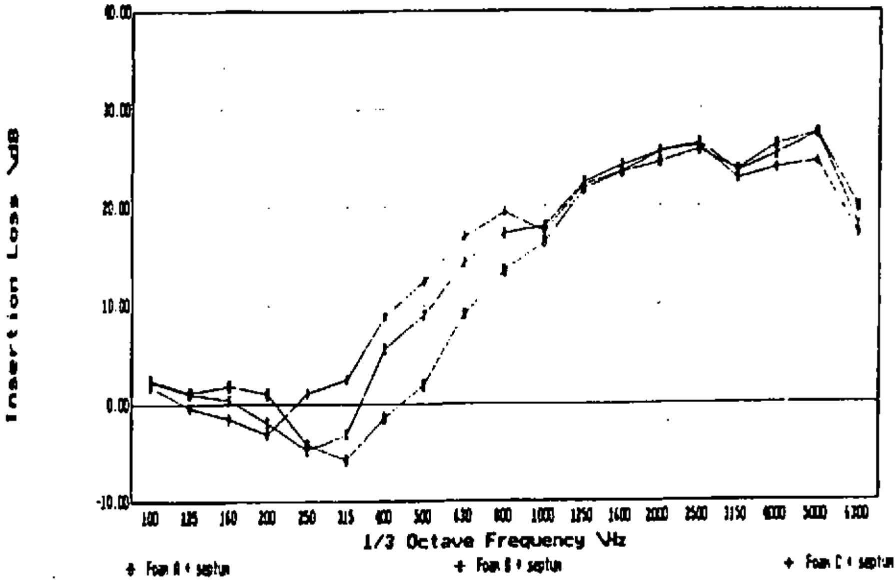


FIG.7 Insertion Loss

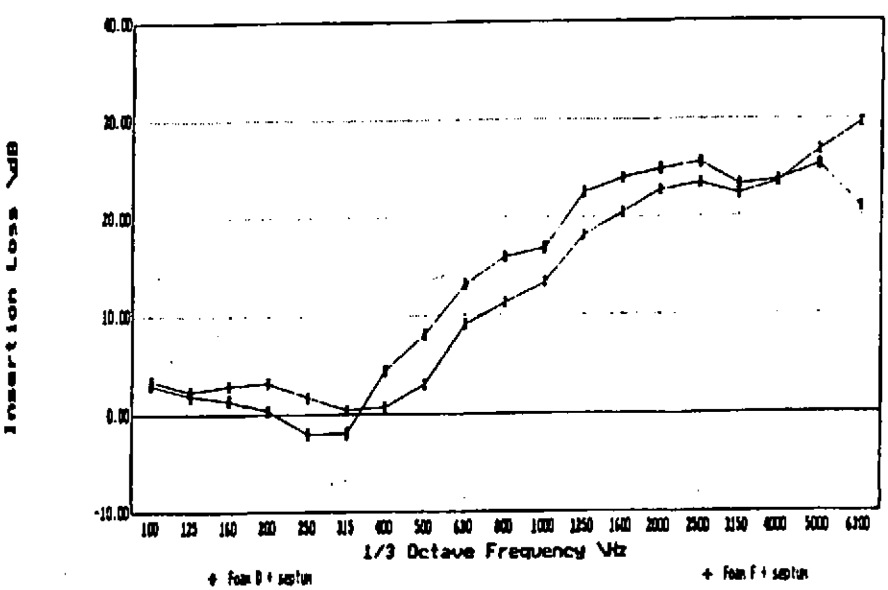
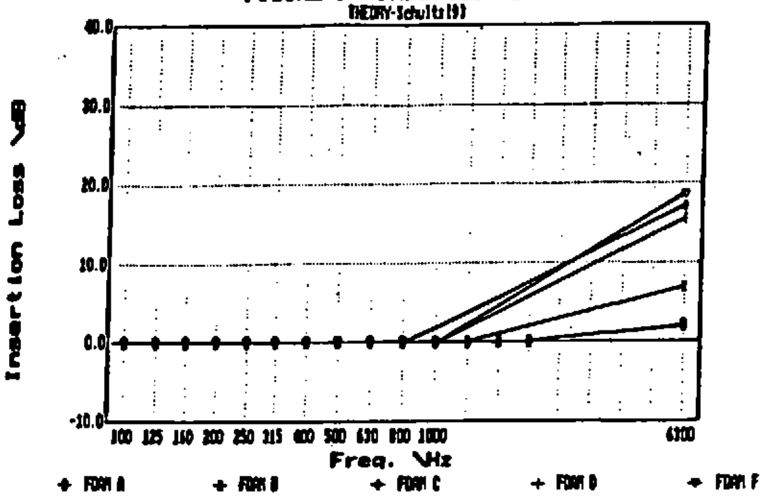


FIGURE 8 Insertion Loss

THEORY-Schultz(9)



Sample	Cut-off Frequency \Hz	
	Measured 1/3 Oct.	Theory Eqn()
A	3150	2450
B	1250	1372
C	630	903
D	1000	1183
F	630	1183

Table 2 Lower Cut-off Frequency

Sample + Septum	Resonant Frequency \Hz	
	Measured 1/3 Oct.	Theory
A	315	420
B	250	292
C	200	237
D	250	241
F	315	311

Table 3 Resonant Frequencies of Composite System

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Appendix 1

Foam	Thickness \mm	Density \kgm^-3	Modulus \10^5Pa	Loss Factor @200 HZ	Flow Resistance \10^3 Rayls
A	14	29	2.2	0.18	1.5
B	25	31	1.9	0.15	4.5
C	38	31	1.9	0.14	10.5
D	29	86	1.5	0.21	26.9 (14)*
F	29	84	2.5	0.65	109.6 (19)*

Table 1 Material Properties
(* skin removed)

Appendix 2

Symbols

a = attenuation constant (dBm^{-1})
A = area of sample (m^2)
b = propagation constant of core (m^{-1})
h = core thickness (m)
m_{s1} = surface mass of base panel (kgm^{-2})
m_{s2} = surface mass of septum layer (kgm^{-2})
R_1 = flow resistivity (Nsm^{-4})
R = sound Reduction index (dB)
T = transmission coefficient
w = angular frequency ($rads^{-1}$))
z_0 = specific impedance of air (415 Nsm^{-3})
Z_1 = acoustic impedance of the core layer (Nsm^{-3})