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The Influence of Boundary Conditions and Constraints on the Performance of Noise Control Treatments: Foams to Metamaterials

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Introduction

- Effect of front and rear surface boundary conditions on foam sound absorption
- Influence of edge constraints on transmission loss of poroelastic materials including effect of finite mass supports
- “Metamaterial” Barrier
CENTRAL TECHNIQUES IN THE MEASUREMENT OF ACOUSTIC REFLECTION COEFFICIENTS, WITH APPLICATIONS TO THE DETERMINATION OF ACOUSTIC PROPERTIES OF ELASTIC POROUS MATERIALS

by

John Stuart Bolton

Institute of Sound and Vibration Research
Faculty of Engineering and Applied Science
University of Southampton

Thesis submitted for the degree of Doctor of Philosophy

July 1984
Normal Incidence Measurement of Reflection

![Diagram of normal incidence measurement of reflection](image)

*Source

\[ r_1 \]

Receiver

\[ r_2 \]

*Image Source

Experimental geometry.
Film–faced Polyurethane Foam

Scanning electron micrographs of the foam sample

- 25 mm layer of foam – one side covered with flame-bonded film, the other open.
- Many intact membranes
Reflection Impulse Response

(Film-faced surface up)

(Foam-open surface up)
One-Dimensional Poroelastic Material Theory

Equations of motion:

Fluid:
\[- \frac{\partial p_2}{\partial x} = \rho_2 \frac{\partial v_2}{\partial t} + \rho_2 (\epsilon - 1) \frac{\partial (v_2 - v_1)}{\partial t} + \phi h^2 (v_2 - v_1).\]

Solid:
\[- \frac{\partial p_1}{\partial x} = \rho_1 \frac{\partial v}{\partial t} + \rho_2 (\epsilon - 1) \frac{\partial (v_1 - v_2)}{\partial t} + \phi h^2 (v_1 - v_2).\]

- Based on Zwikker and Kosten, plus Rosin with complex density and air stiffness taken from Attenborough.
Boundary Conditions

- Open foam surface
- Foam surface sealed with an imperious membrane
- Foam fixed to a hard backing
Reflection Impulse Response – Predicted

Open Surface Foam

Film–faced Foam

\[ \rho_f = 20 \text{ kg/m}^3, \quad t = 25 \text{ mm}, \quad \varphi = 0.9, \quad B_m = 8.125 \times 10^5 \text{ Pa}, \quad \eta = 0.265, \]
\[ \varepsilon = 6.025, \quad \sigma = 130 \times 10^5 \text{ Nks Rayls/m}, \quad \nu = 0.485, \quad m_s = 0.045 \text{ kg/m}^2 \]
Film–faced Foam / Thin Air Gap

at $x = l + \Delta$, $v_a = 0$;

at $x = l$, $P_1 = P_a(1 - h), P_1 = P_a h, v_a = v_1(1 - h) + v_2 h$;

at $x = 0$, $v_1 = v, v_2 = v, P - p_1 - p_2 = m_s \frac{dv}{dt}$

Impedance: $j\omega z = -\omega^2 m_s - N'/D'$

The solution of this set of seven equations presents no difficulties in principle, but is algebraically tedious. The complete solution is outlined in Appendix 6.2; only the result is given here. The impedance takes the form

$j\omega Z = -\omega^2 m_s - N'/D'$
Film-forced Foam / Thin Air Gap

Inverted reflection from rear surface

Effect of rear surface boundary condition on film normal incidence absorption coefficient: model of section 6.4.3.2; model of section 6.4.3.3, air layer depth 0.001m.
Rear Surface Boundary Conditions

25mm foam layer with bonded membrane

1. No Airspace:

2. Airspace:

\( \Delta = 1 \)
Absorption treatments

- Bonded/Bonded

- Bonded/Unbonded

- Unbonded/Bonded

- Unbonded/Unbonded
Normal Incidence Absorption

Effects of Airspace at front and rear

1. Film/Foam/Backing
2. Film/Space/Foam/Backing
3. Film/Foam/Space/Backing
4. Film/Space/Foam/Space/Backing

- Foam - 25 mm, 30kg/m³
- Membrane - 0.045 kg/m²
- Airspaces - 1 mm
Impedance Tube Testing

- **Melamine Foam (8.6 kg/m³)**
  - 100 mm diameter
  - 25 mm thick

- **Each sample fit exactly by trimming the diameter & checking the fit with a TL measurement**

- **Two Facing & Two Rear Surface Boundary Conditions**
  - Multiple trials
  - Multiple samples
Sample Fit: TL Qualification

- Non-Zero TL = Sample Constrained
- Zero TL = Sample Free to Move

Transmission Loss

As-Cut
1st Trim
2nd Trim
3rd Trim
4th Trim

No Leakage
Surface Configurations

**Front Surface:**

1) Plastic film near, but not adhered to foam

2) Plastic film glued to foam

**Rear Surface:**

1) Small gap between foam & rigid wall

2) Foam adhered to rigid wall
Absorption vs. Configuration – Test

Absorption Coefficient

\[ l = 25mm, \Delta_1 = 4.5mm, \Delta_2 = 1mm, m_s = 50 \text{ g/m}^2, h = 0.99 \]
\[ \sigma = 9.5 \times 10^3 \text{ mks Rayls/m}, \quad \varepsilon = 1.4, \]
\[ P – \text{wave modulus} = 6.5 \times 10^5 \text{ Pa}, \eta = 0.2 \]
Helmholtz Resonator Effect

Mechanical Impedance

- Mass: \( m = \rho_0 S L' \)
- Stiffness: \( s = \rho_0 c_0^2 S^2 / V \)

Total Acoustic Impedance: \( z = 1/(1/z_H + 1/z_f) \)
Helmholtz Resonator Effect

Combined Foam + Helmholtz Resonator System is Similar to Measured System
Helmholtz Resonator Effect

But is it really due to edge gaps?

Measured Glued Facing + Fixed with Edge Sealed
Sound absorption of elastic framed porous materials in combination with impervious films: effect of bonding

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Abstract

The absorption characteristics of elastic framed absorbers in combination with impervious films has been investigated. The effect of bonding the film to the absorber and the absorbers to their rear surface was examined. The results have been modelled using established methods for predicting the absorption of elastic framed porous materials. The absorption of a foam with a film bonded to its top surface was most sensitive to the rear surface bonding condition. Plain foams and foams with loose-laid surface films were less sensitive to the rear surface bonding condition. The results demonstrate that test data used to predict absorption performance need to reflect the absorber mounting conditions. © 2002 Elsevier Science Ltd. All rights reserved.
Table 1
Parameters used for the modelled results in Fig. 1

<table>
<thead>
<tr>
<th>Thickness (mm)</th>
<th>Tortuosity</th>
<th>Bulk density ( \rho_1 ) (kg/m(^3))</th>
<th>Flow resistivity ( r ) (mks rayls/m, or Ns/m(^4))</th>
<th>Porosity ( h )</th>
<th>Complex shear modulus ( N ) (N/cm(^2))</th>
<th>Poison’s ratio ( \nu )</th>
<th>Form factor ( c )</th>
</tr>
</thead>
<tbody>
<tr>
<td>24</td>
<td>2.85</td>
<td>43</td>
<td>22000</td>
<td>0.98</td>
<td>20 + 10i</td>
<td>0.3</td>
<td>4</td>
</tr>
</tbody>
</table>

Fig. 1. Measured (○) and modelled (−) absorption of film faced foam at 24 mm thickness; foam was placed on rear surface (floor of reverberation room).

Fig. 2. Measured (○) and modelled (−) absorption of film faced foam at 24 mm thickness; foam was bonded to rear surface (gypsum board).

\[ m_s = 35 \, g/m^2 \]
Tensioned Membranes Model Verification – Velocity Measurement

[Diagram showing the setup with components such as Power Amplifier, Pre-Amplifier, Signal Analyzer, Microphone, Membrane, Amplifier, Sound Source, Finite Backing, Laser Sensor, etc.]

[Images of the experimental setup, including the membrane and measurement equipment.]
Model Verification – Vibrational Modes

**Theory**

Absolute velocity of membrane - Theory

1\textsuperscript{st}

Absolute velocity of membrane - Theory

2\textsuperscript{nd}

**Experiment**

Absolute velocity of membrane - Experiment
Model Verification – Experiment Set-up

Diagram of the experiment set-up:
- Power Amplifier
- Pre-Amplifier
- Signal Analyzer
- Sound Source
- Microphone
- Anechoic Termination
- Test Sample

Photo of the experiment set-up:
- B&K Pulse System
- Speaker Amplifier
- Microphones
- B&K Standing Wave Tube
- Computer
Model Verification – Model Optimization

- Given experimental results as input, find appropriate material properties ($T_0$, $\rho_s$, $\eta$)

- Why this behavior? – Finite size, held at edge, finite stiffness.
Glass Fiber Material Inside of Sample Holder
Anechoic Transmission Loss (Green)

Increase in TL due to edge constraint (10dB)
# Poroelastic Material Properties Used in Calculations

<table>
<thead>
<tr>
<th>Material</th>
<th>Bulk density (Kg/m³)</th>
<th>Porosity</th>
<th>Tortuosity</th>
<th>Estimated flow resistivity (MKS Rayls/m)</th>
<th>Shear modulus (Pa)</th>
<th>Loss factor</th>
</tr>
</thead>
<tbody>
<tr>
<td>Yellow</td>
<td>6.7</td>
<td>0.99</td>
<td>1.1</td>
<td>21000</td>
<td>1200</td>
<td>0.350</td>
</tr>
<tr>
<td>Green</td>
<td>9.6</td>
<td>0.99</td>
<td>1.1</td>
<td>31000</td>
<td>2800</td>
<td>0.275</td>
</tr>
</tbody>
</table>
Variation of Shear Modulus

- As shear modulus increases, the minimum location of TL moves to higher frequencies
Variation of Flow Resistivity

- Flow resistivity controls TL at low and high frequency limit
Investigation of Vibrational Modes of Glass Fiber Materials
Vibrational Modes of Fiber Glass Materials (1st and 2nd Modes, Green)

1\textsuperscript{st} (133 Hz)

2\textsuperscript{nd} (422 Hz)
Internal Constraint to Enhance the Sound Transmission Loss
Sound Transmission Loss (Experiment, Green) [Density of Plexiglass: 1717 Kg/m3]
Effect of Releasing the Internal Cross–Constraint (Measurement)

- Relatively heavy constraint required to realize low frequency benefit.
Effect of Releasing the Internal Cross–Constraint (FEM Prediction)

![Graphs showing TL (dB) vs. Frequency (Hz) for Cardboard and Plexiglass Constraint cases.]

- **Cardboard Constraint**
- **Plexiglass Constraint**
Metamaterials

- Metamaterials are artificial materials engineered to have properties that may not be found in nature. Metamaterials usually gain their properties from structure rather than composition, using small inhomogeneities to create effective macroscopic behavior.

From: Meta-Material Sound Insulation by E. Wester, X. Bremaud and B. Smith, Building Acoustics, 16 (2009)
Membrane-type metamaterials: Transmission loss of multi-celled arrays

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Acoustic metamaterials with negative dynamic mass density have been shown to demonstrate a five-fold increase in transmission loss (TL) over mass law predictions for a narrowband (100 Hz) at low frequencies (100–1000 Hz). The present work focuses on the scale-up of this effect by examining the behavior of multiple elements arranged in arrays. Single membranes were stretched over rigid frame supports and masses were attached to the center of each divided cell. The TL behavior was measured for multiple configurations with different magnitudes of mass distributed across each of the cell membranes in the array resulting in a multipeak TL profile. To better understand scale-up issues, the effect of the frame structure compliance was evaluated, and more compliant frames resulted in a reduction in the TL peak frequency bandwidth. In addition, displacement measurements of frames and membranes were performed using a laser vibrometer. Finally, the measured TL of the multi-celled structure was compared with the TL behavior predicted by finite element analysis to understand the role of nonuniform mass distribution and frame compliance. © 2011 American Institute of Physics. [doi:10.1063/1.3583656]
Proposed Mass–Neutral Material

Cellular panel

Homogenized mat.

$nL$$\gg L$  

Frame (Mat. A)

Plate (Mat. B)

$M_{eff} := M_{eff}f$

$T = \frac{2\rho_0 c}{2\rho_0 c + j2\pi f M_{eff} f}$

$STL = -20\log|T|$

$M_{eff}$: Mass per unit area

$STL$: Sound Transmission Loss

- Cellular material with a periodic array of unit cells
- Unit cell has components with contrasting mass and moduli
- Characteristics of infinite, periodic panel are same as that of a unit cell for normally incident sound
A clamped plate has high STL at very low frequencies due to the effect of boundary conditions and finite size and stiffness.
Material–Based Mass Apportioning

- Each unit cell
  - Overall mass constant
  - Different materials for frame and plate

- A series of cases for $\mu$ between 0.1 and 10000
  - $\rho_p$ and $\rho_f$ varied
  - $E_f$ varied keeping $E_p$ constant so that $E_f/E_p = \rho_f/\rho_p$
Experimental Validation

- A good qualitative agreement is observed between measurements and FE predictions.
Material–Based Mass Apportioning

- As $\mu \uparrow$
  - High STL region broadens in the low frequency regime
  - Region between the first peak and dip is widening
  - The dip – being shifted to the right – desirable

- $\mu \rightarrow O(100) \rightarrow$ saturates

<table>
<thead>
<tr>
<th>$\mu$</th>
<th>$\rho$ [kg/m$^3$]</th>
<th>Fr.</th>
<th>$E_{fr}$ (GPa)</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.1</td>
<td>3910</td>
<td>107</td>
<td>0.055</td>
</tr>
<tr>
<td>0.5</td>
<td>2868</td>
<td>393</td>
<td>0.274</td>
</tr>
<tr>
<td>1</td>
<td>2151</td>
<td>590</td>
<td>0.549</td>
</tr>
<tr>
<td>10</td>
<td>391</td>
<td>1073</td>
<td>5.490</td>
</tr>
<tr>
<td>100</td>
<td>43</td>
<td>1168</td>
<td>54.900</td>
</tr>
</tbody>
</table>

$E_p = 2$ GPa
Effective Mass as a Function of Frequency

- Magnitude of $M_{\text{eff}}$ higher than space-averaged areal mass in the range of 0-1000 Hz
- An order of magnitude higher in 800 – 1000 Hz range
- Shows strong negative mass effect in the peak STL region

\[ T = \frac{2\rho_0 c}{2\rho_0 c + j2\pi fM_{\text{eff}} f} \]
Mechanism Behind High STL

- Averaged displacement phase switches from negative to positive value at the STL peak
- Parts of the structure move in opposite directions—similar to observations in LRSMs—resulting in zero averaged displacement
- “Negative mass” observed without locally resonant elements
Hybrid Material

- Cellular structure increases STL at low frequencies
- Lightweight, fine fiber fibrous layer can be used to recover performance at higher frequencies
Hybrid Material

Low Sound Speed Front

Directs non-normally incident sound to core

Metamaterial Core

Locally resonant core

Fibrous Cell Filling

Fibrous cell filling

Increases STL at high Hz

- Predicted Sound Transmission Loss in Hybrid System with Fibrous Cell Filling
Conclusions

• Front and rear boundary conditions have a profound effect on the sound absorption offered by poroelastic materials

• Those effects are predictable and measureable

• Internal constraint of poroelastic materials can increase their transmission loss, but finite weight of required supports should be accounted for

• Metamaterials for transmission loss typically depend on the presence of constraints, geometry and flexural stiffness for their performance

• A proposed mass-neutral “metamaterial” barrier featuring spatially-periodic internal constraints gives low frequency advantage with respect to the mass law, but would require supplementary material to mitigate performance loss at high frequencies
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References