HIGH TEMPERATURE ACOUSTIC LINER TECHNOLOGY

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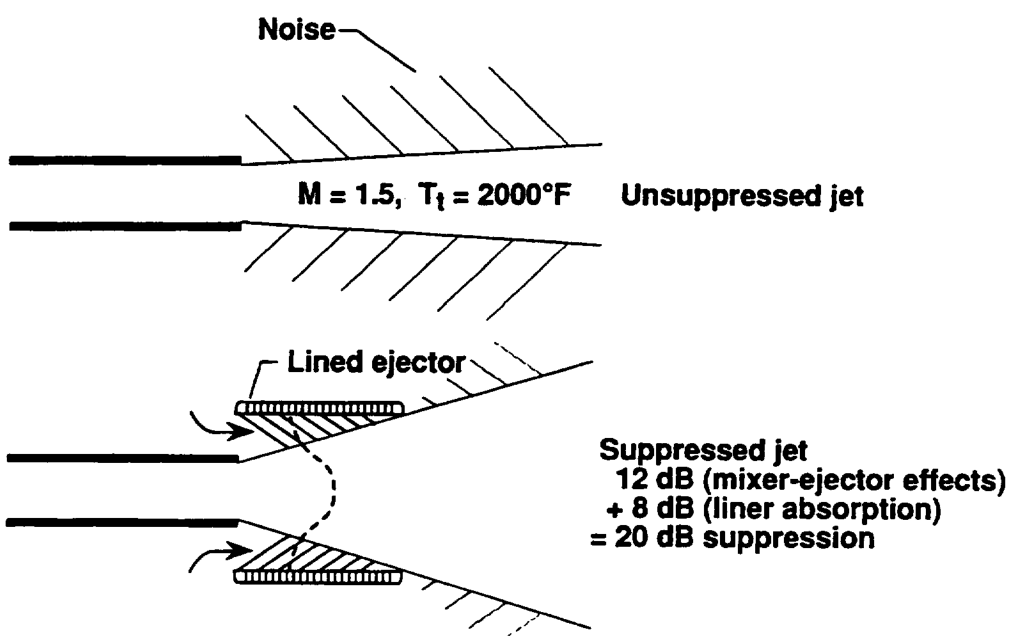
THE EJECTOR NOISE SUPPRESSION PROBLEM: Reduce radiated noise by 20 dB



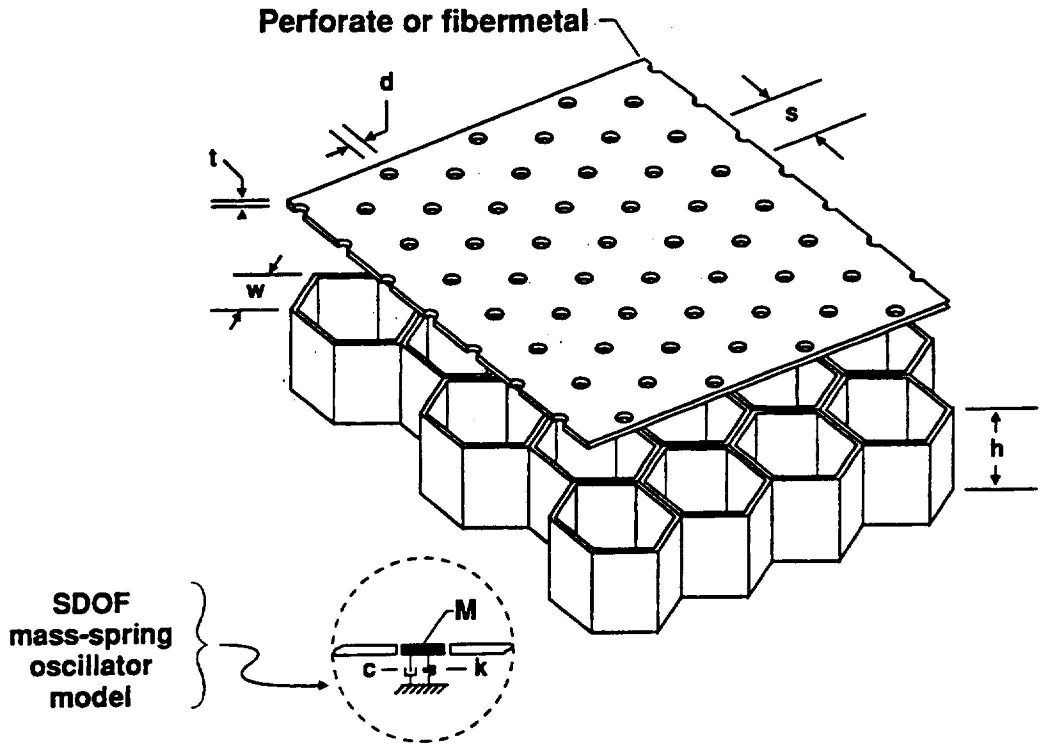
This paper describes work currently in progress at Langley on liner concepts that employ structures that may be suitable for broadband exhaust noise attenuation in high speed flow environments and at elevated temperatures characteristic of HSCT applications. Because such liners will need to provide about 10 dB suppression over a 2 to 3 octave frequency range, conventional single-degree-of-freedom resonant structures will not suffice. Bulk absorbers have the needed broadband absorption characteristic; however, at lower frequencies they tend to be inefficient.

HSCT LINER
CONCEPT DEVELOPMENT

- <u>OBJECTIVE</u>: Achieve 10 dB suppression over 3 octaves via absorptive liner
- <u>APPROACH</u>: Exploit CMC technology to achieve 'bulk-like' absorption characteristic

At Langley, we are investigating two concepts that exploit the characteristics of both resonant and bulk absorbers to provide the needed broadband exhaust noise suppression. For both concepts, the resistive component at the liner surface is supplied mainly by internal viscous dissipation. This possibility should allow more accurate impedance predictions at high temperature. If evolving ceramic matrix composite (CMC) materials technology permits the fabrication of such structures to withstand the harsh environment of HSCT exhaust nozzle systems, then reliable source noise/duct propagation analysis should enable one to accurately predict the noise reduction of lined jet mixer/ejector systems.

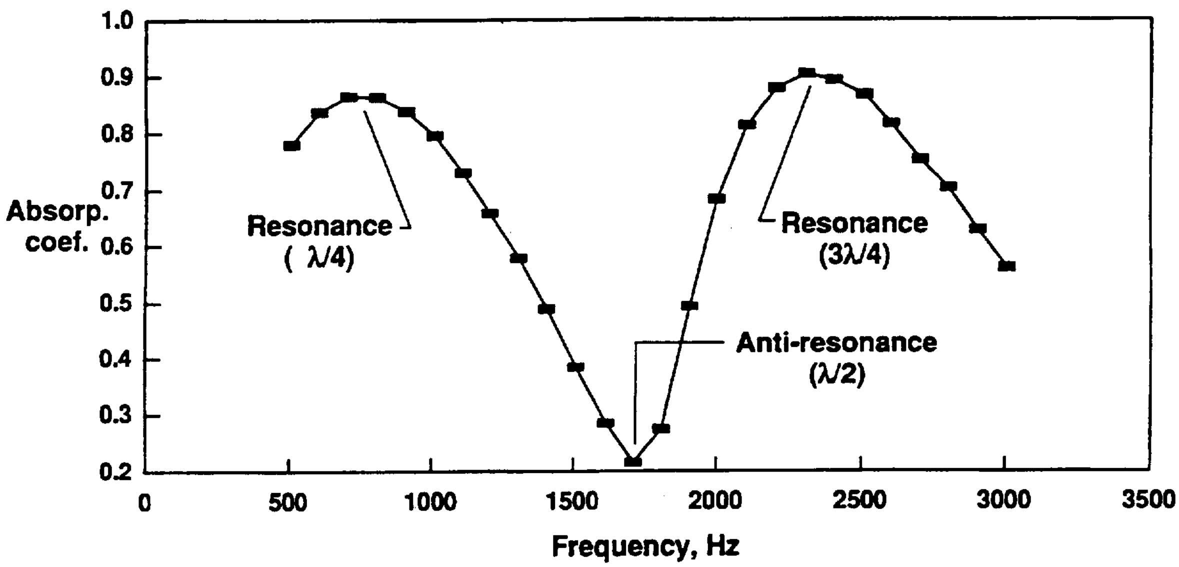
SINGLE LAYER RESONANT ABSORBER



The conventional single layer resonant absorber is most simply implemented by attaching a perforate facesheet to airtight partitioned cavities as shown. The simplest model for the acoustics of such a system is an array of mass-spring oscillators as depicted in the sketch. The oscillator masses are defined by concentrated packets of kinetic energy in and around the perforate holes. The spring is provided by the trapped air in the honeycomb cavities and the damping (acoustic resistance) is dominated by fluid dynamic inertial losses (resulting in turbulent eddies) associated with the high sound pressure amplitudes that typically occur in aircraft applications (as opposed to viscous losses at low amplitudes).

Because of the inertial loss dominated resistance at high sound pressure levels, the resistance tends to be amplitude dependent. Grazing flow increases the resistance and tends to desensitize it to large sound pressure amplitudes. These nonlinear effects can be minimized by making the hole t/d large, or by using fibermetal facesheets that cause viscous losses to play a relatively greater role. Generally, it is necessary to characterize the resistive component with the aid of empirical procedures (ref 1). These procedures are unreliable outside the parameter range underlying the database for a particular structure of interest. Note that the mass-spring model for this liner predicts an absorption maximum at resonance. One effect of the non linear behavior mentioned above is to broaden the absorption spectrum.

ACOUSTIC POWER ABSORPTION
RESONANT ABSORBER

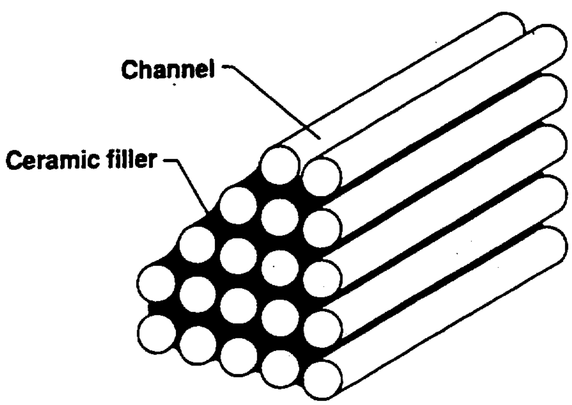


FM125/4 inch cavity depth
resistance = 0.24ρc

When wave motion in the partitioned cavities of the single layer resonant absorber is taken into account, higher harmonic resonances are predicted. An example of such an absorption spectrum measured at normal incidence is shown for a single layer absorber consisting of a fibermetal facesheet with a normalized flow resistance of 0.24 rho-c units (100 Rayls). Absorption maxima occur at the first and second resonances corresponding approximately to cavity lengths of 1/4 and 3/4 acoustic wavelengths. Note the absorption minimum at the anti-resonance. It is these absorption minima at the anti-resonances that restrict the bandwidth of resonant absorbers.

Because resonant behavior is set up by reflected waves in the cavities, the anti-resonance absorption minimum can be modified by coupling a second resonator in series (i.e. a double layer absorber) or by attenuating the propagating wave in the backing cavity. One way to attenuate the propagating wave is by stuffing the honeycomb cells with a fibrous material to a predetermined density that provides an appropriate propagation constant. Another more appealing way to accomplish the same thing is to reduce the diameter of the honeycomb cells to near capillary size, eliminate the facesheet and at the same time maintain a high frontal area porosity. This concept is depicted schematically in the next figure.

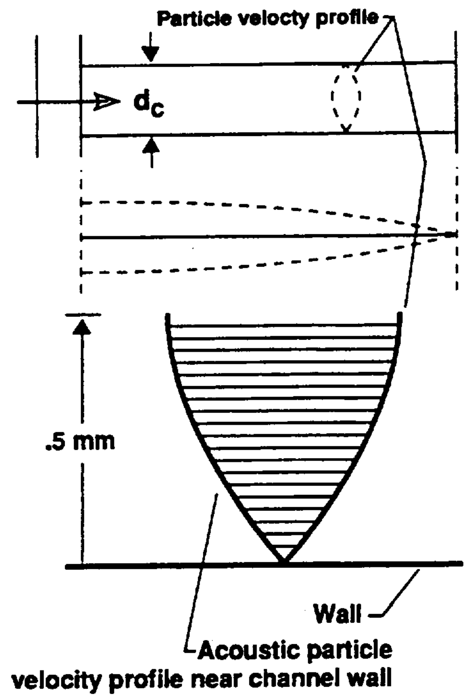
ACOUSTIC VISCO-THERMAL DISSIPATION IN TUBES (Approach to Broadband or 'Bulk-like' Absorbers)



Surface averaged impedance of N capillary-like channels of depth L and diameter d_c:

$$S = \frac{ik}{N\Gamma}\left(\frac{d_s}{d_c}\right)^2 \operatorname{Coth}(\Gamma L)$$

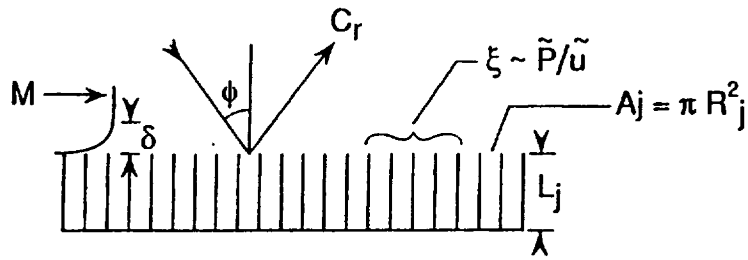
Where $\Gamma = \alpha + i\beta$ } Propagation constant in typical channel



For a parallel array of capillary channels depicted, acoustic resistance is provided by viscous dissipation. The absorption spectra peaks can now be controlled by channel diameter, d_c, length, L, and frontal porosity. Instead of the dissipative process being concentrated near the face, it is now dispersed throughout the channel length. In addition to 'smoothing out' the resonant behavior of the absorption spectrum, the acoustic impedance is accurately predictable from first principles, i.e. the propagation constant in the channels depends on channel geometry and gas properties alone. Because the gas properties (sound speed, density, viscosity and thermal conductivity) are well known functions of temperature, acoustic absorption can be predicted at elevated temperatures. Furthermore, because the acoustic resistance arises from internal dissipation, grazing flow is expected to have minimal effect (ref 2).

Ceramic tubular structures (ceramic honeycomb) that can withstand temperatures up to 1800° F are available in the dimensions needed to provide a useful range of acoustic impedance for mixer/ejector models. While these structures are certainly not viable for direct HSCT exhaust nozzle applications (too heavy and mechanically fragile), they can serve as a development tool. In what follows, absorption spectra for two ceramic honeycomb geometries will be discussed.

IMPEDANCE MODEL FOR CERAMIC HONEYCOMB



Surface admittance: $\beta = \frac{1}{\xi} = \frac{\rho c}{A} \sum_{j=1}^{N} \frac{A_j}{W_j \coth(\Gamma_j L_j)}$

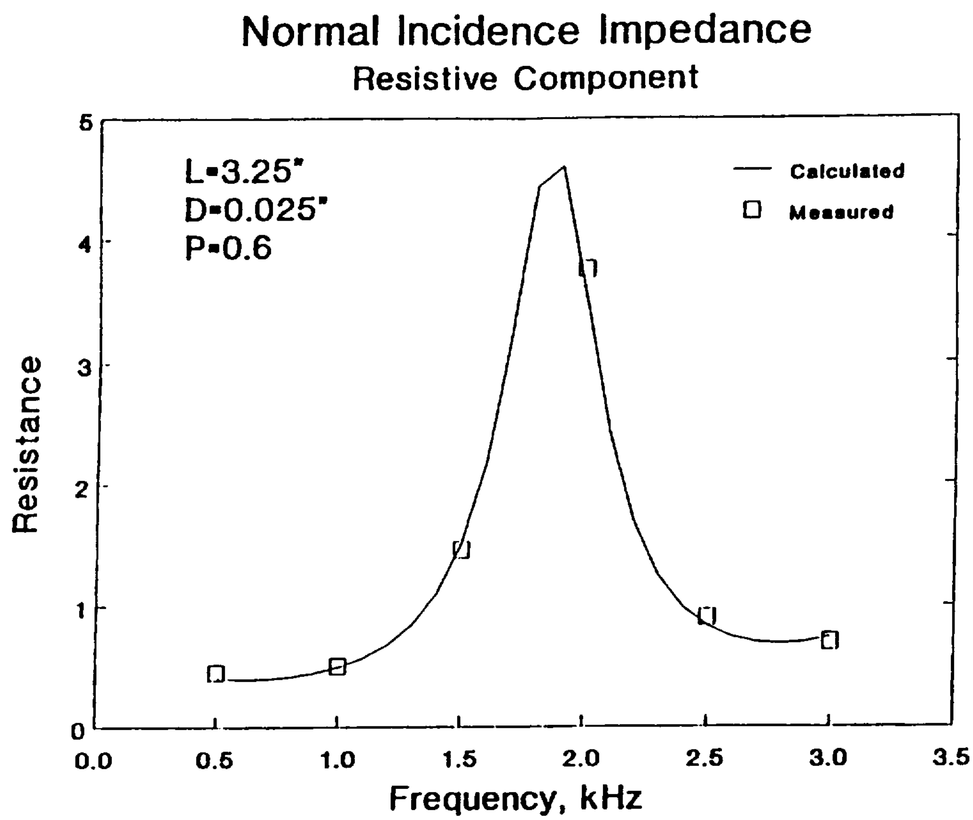
Channel characteristic impedance: $W_j = \frac{ik\rho c}{\Gamma_j}$

Channel propagation constant

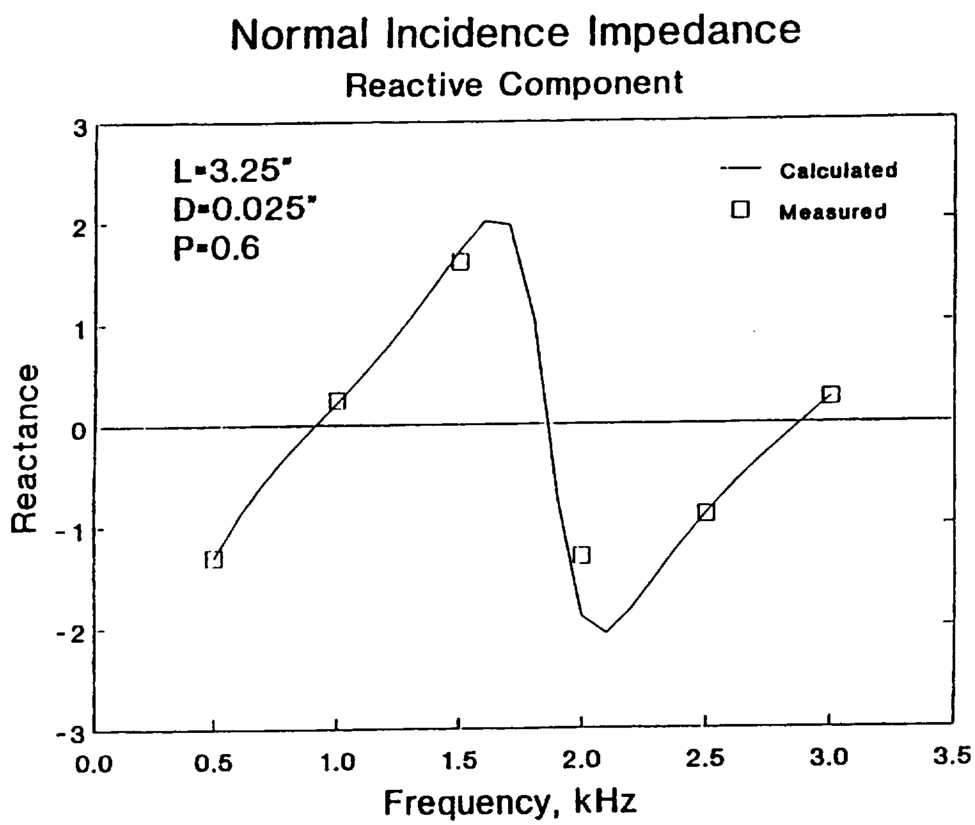
$$\Gamma_j = k \left\{ \frac{1}{\sqrt{2}} \left(\frac{\gamma - 1 + \sigma}{s\sigma} \right) + i \left[1 + \frac{1}{\sqrt{2}} \left(\frac{\gamma - 1 + \sigma}{s\sigma} \right) \right] \right\}$$

$$s = R_j \sqrt{\frac{\rho\omega}{\mu}}\ , \sigma = \sqrt{Pr}$$

The effective surface admittance over an array of channels, occupying a frontal area A, is the sum of the individual channel admittances, taking into account the continuity of mass flow into the material. The accuracy of this model is critically dependent upon a knowledge the propagation constant inside a channel. Propagation in capillary tubes has been studied extensively and is summarized by Tijdeman (ref 3). A fairly accurate model for the wavelength range of interest here is shown at the bottom of the figure. In this formula, Pr is the Prandtl number and k is the acoustic free-space wavenumber. The so-called shear wavenumber, s, involves channel radius, gas density, viscosity and sound frequency. The key feature of note is that only the tube geometry and fundamental gas properties appear, i.e. no empirical constants are present.



Calculated and measured resistance at the surface of a 3.25 inch length of ceramic honeycomb with channel diameters of 0.025 inches and a frontal porosity of 0.6 is shown in the figure. Note the resistance increase in the vicinity of the anti-resonance near 1.8 kHz which is well predicted by the theory.



This figure shows a comparison of the calculated and measured reactive component for the surface impedance of the same material. The zero crossings with positive slope are resonances and the zero crossing with negative slope is the anti-resonance.

CERAMIC HONEYCOMB AS LABORATORY TEST LINER MATERIAL

• Temperature Tolerant	1800°F
• Thermal Shock Resistance	Excellent
• Tangential Modulus of Rupture	1200 psi
• Compressive Strength	4000 psi
• Bulk Density	30 lbs/ft^3 (s.g.≈0.5)
• Pore Diameter	0.025 in.
• Internal Surface Area/Volume	133 in.^{2}/in.^{3}
• Cells/in.^{2}	1400
• Face Porosity	73%
• Cost	\$15-\$30/in.^{2}

Parameters of interest for ceramic honeycomb are listed. The bulk-like acoustic absorber properties arise from its high volume porosity of 73% and internal surface area per unit volume of 133 in^2/in^3. In addition to the possibility that such a structure may provide efficient acoustic absorption, it can also function at temperatures up to 1800° F and thus can serve as a laboratory test material for developing liner concepts for mixer/ejectors.

INSERTION LOSS MATH MODEL

Assumptions:

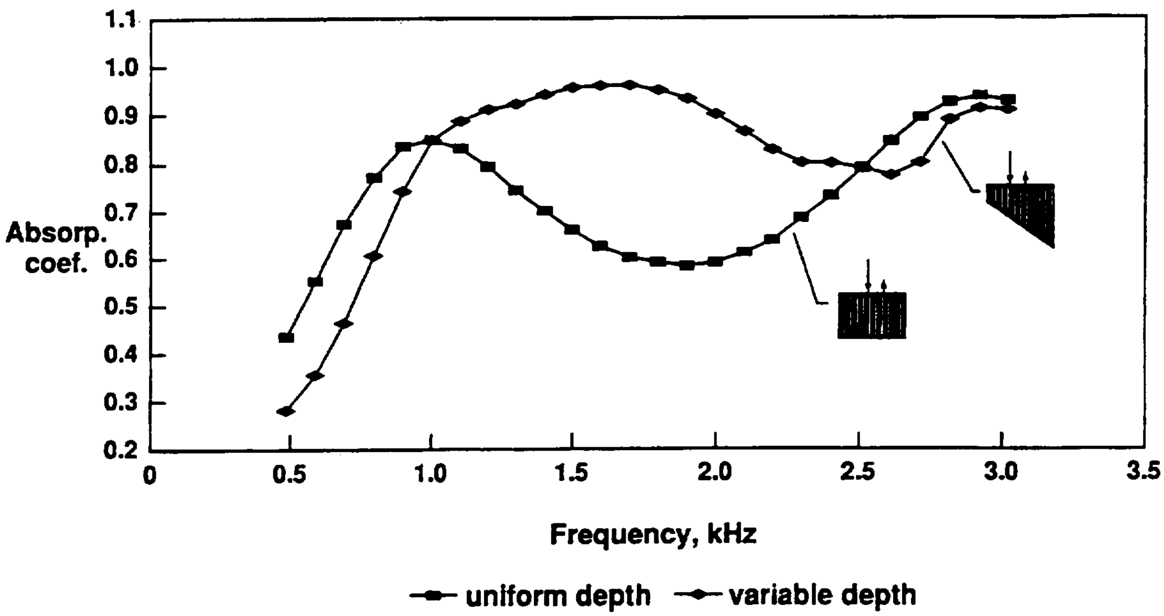
- **Acoustic waves experience one reflection at liner surface**
- **Acoustic attenuation due to liner absorption only**
- **Small boundary layer thickness to wavelength ratio,**
- **i.e. $\delta/\lambda < 1$**
- **No sound generation by liner roughness**

$\therefore$ **Insertion loss = 10 Log** $|C_r|^2$

$$\text{where } C_r = \frac{\cos\phi\,(1 + M\sin\phi) - \beta}{\cos\phi\,(1 + M\sin\phi) + \beta}$$

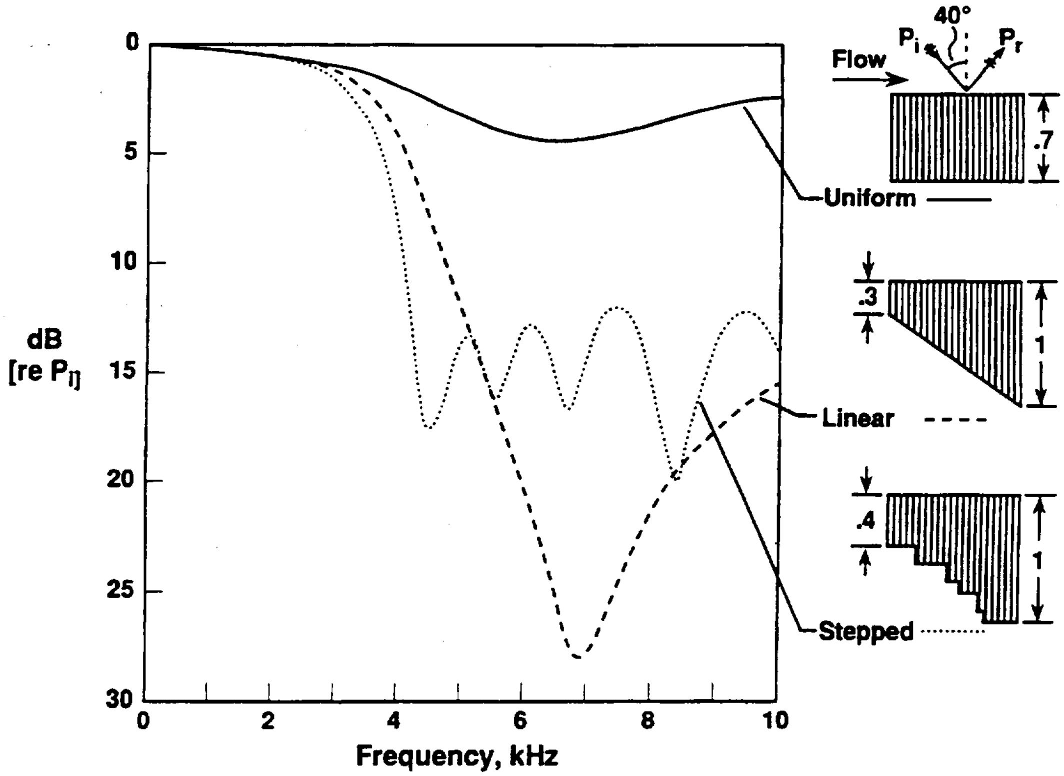
Given the effective surface admittance, ray acoustics can be employed to estimate liner insertion loss for a mixer/ejector configuration. An acoustic ray is assumed to experience one reflection before being convected out the ejector exit. The power lost to the absorber can be interpreted as an insertion loss in the reflected wave relative to that for a perfectly reflecting surface. The reflected wave amplitude, C_r, is given as a function of the incident angle, flow Mach number and surface admittance. We use the results of this calculation to provide a simple figure of merit to estimate relative performances of test liners in mixer/ejectors models.

ACOUSTIC ABSORPTION FOR CERAMIC HONEYCOMB



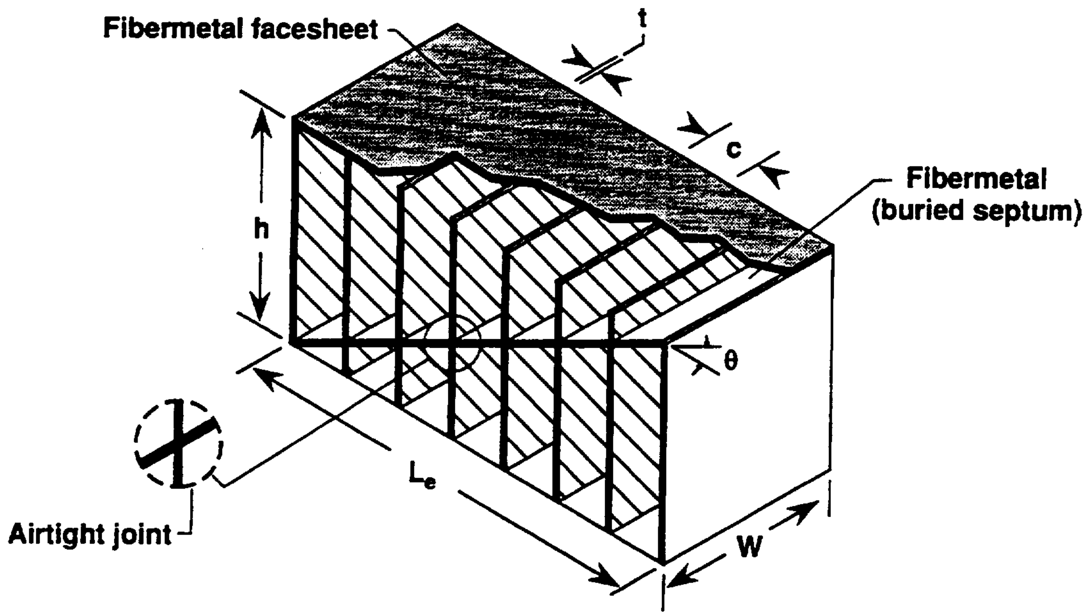
Absorption coefficient spectra for a uniform depth and a constant slope varying depth ceramic honeycomb structure is shown in this figure. Note that the resonant character of the channel length is still in evidence although significant absorption is occurring in the neighborhood of the anti-resonance at 1.8 kHz. For the constant slope variable depth specimen, a broad peak in absorption occurs across the entire span between the first two resonances. Thus the basically resonant system is behaving much like a bulk absorber. We would like to exploit this behavior to provide useful design concepts for HSCT liners.

POWER LEVEL OF REFLECTED WAVE RELATIVE TO INCIDENT WAVE FOR VARIABLE DEPTH LINER



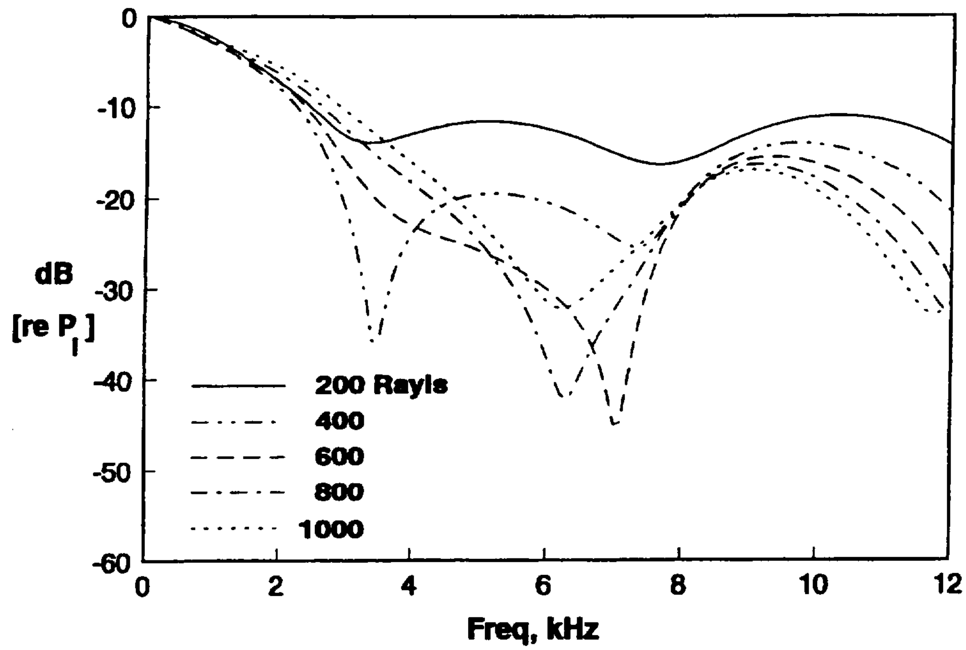
This figure shows effects of nonuniform liner depth on the reflected power relative to the incident power (insertion loss relative to hard wall) for a grazing flow of Mach number 0.5 and incidence angle of 40°. Note that the insertion losses for variable depth liners cannot be achieved by replacing them with their uniform average depth 'equivalents'. Also, changes in the variable depth profile (compare linear slope depth with stepped depth) causes significant changes in the absorption spectra. The solid line depicts the insertion loss of a liner with a uniform depth of 0.7 inches. The long- and short dashed lines show the insertion losses of a linear slope depth and stepped depth variation respectively. The stepped depth variation was an attempt to optimize the absorption bandwidth.

VARIABLE DEPTH SEPTUM, PARALLEL PLATE ACOUSTIC CHAMBER



A second concept for achieving spatially variable impedance with the dominant dissipation component internal to the structure is illustrated in this figure. Here, a typical element consists of a series of channels covered with a porous face sheet and embedded with a variable depth, porous septum that supplies a resistive coupling between the upper and lower sections of each channel. The system is essentially an array of contiguous two-degree-of-freedom systems that provide a spatially varying impedance by changing the location of the coupling element (i.e. the porous buried septum). A key distinction between this concept and the capillary channel concept is that the small channel widths or diameters are no longer necessary, i.e. viscous dissipation which occurs along the channel walls is relatively insignificant. The parallel plate structure is intended to enhance heat transfer out of the structure for high temperature environments.

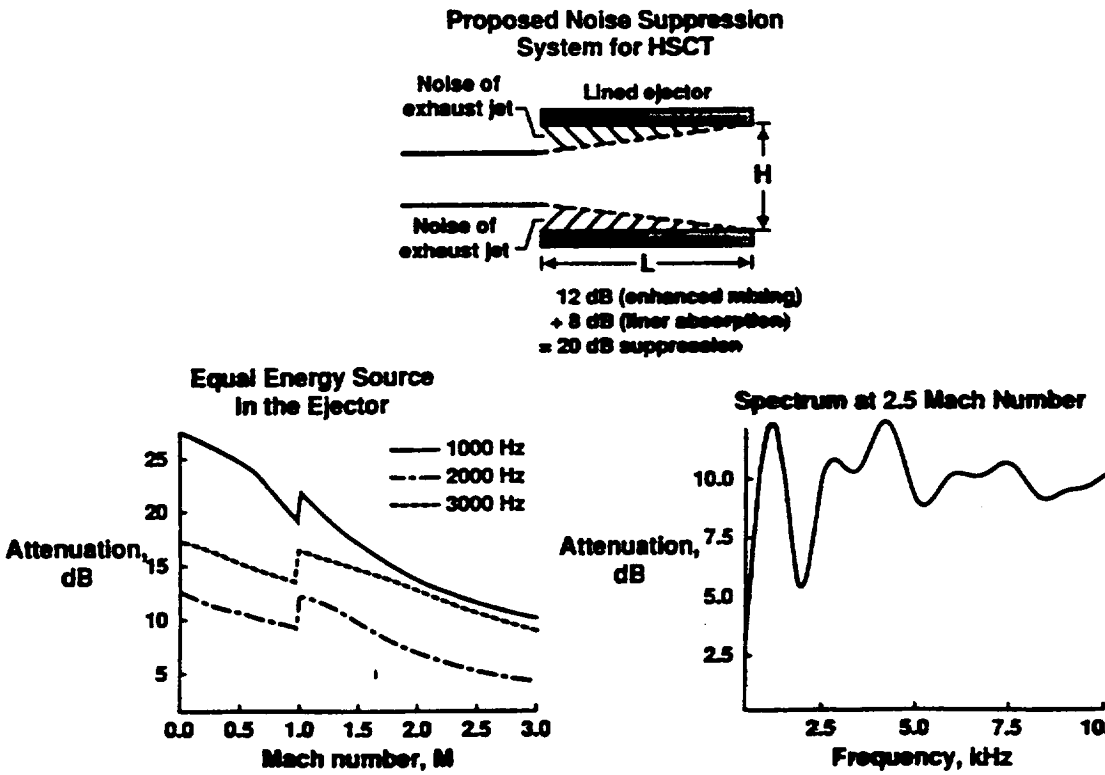
Power Level of Reflected Wave relative to Incident Wave for Variable Depth Septum



M = 0.5, T = 500 deg, Angle of Incidence = 40 deg

These curves show progressive changes of insertion loss spectra (i.e power loss in reflected wave relative to a hardwall reflector) for increasing values of the variable depth, buried septum flow resistance from 200 Rayls to 1000 Rayls. Note the progression from a double peak in the insertion loss for low resistance to a single peak at the higher resistance values. Clearly, insertion loss spectra can be tailored to some extent with an appropriate choice of septum flow resistance.

CALCULATIONS OF THE NOISE SUPPRESSION OF LINED EJECTORS



Ray acoustics provides an estimate of insertion losses for a jet mixer/ejector assuming all sound rays encounter a single reflection at the liner surface, i.e. that duct modes play no significant role. This figure shows the result of a modal analysis for a mixer/ejector like duct configuration with uniform flow and lined with a one inch thick layer ceramic honeycomb discussed above. The calculation was done for the first ten modes assuming no reflections at the duct exit. The duct length to height ratio was 2.6. A uniform flow and temperature profile was assumed. Dr. Willie Watson at Langley is further developing this approach to handle impedance discontinuities as well as continuously variable impedances. He also intends to include nonuniform flow and temperature profile effects.

The encouraging aspect of these results is the significant attenuations calculated for supersonic flows. Up to Mach numbers of 1.5, attenuations of at least 10 dB are calculated for a liner L/H of 2.6. Furthermore, the attenuation spectrum is broadband in character, even for a spatially uniform impedance. These results are encouraging.

SUMMARY

'Bulk-like' absorption characteristic potentially achievable by:

- **Variable depth, capillary channel structures**
- **Variable depth, porous septum structures**

Based on simple ray acoustic modeling, broadband absorption can apparently be achieved with variable depth, capillary tube structures. Such structures are available commercially. This material, although not viable for HSCT applications, can serve as a means to develop and validate acoustic liner concepts for high speed flow, elevated temperature jet-mixer/ejectors.

Broadband absorption spectra may also be achieved by spatially variable impedances implemented with built-up parallel plate structures with variable depth, porous, buried septa. Several such test structures are currently being fabricated for testing at Langley.

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