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EXPERIMENTAL INVESTIGATION ON SOUND TRANSMISSION

THROUGH CAVITY-BACKED PANELS

By

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16. Abstract

A considerable amount of information has been developed in architectural acoustics on sound transmission through panels. Emphasis has been on high-frequency transmission where "mass law" is applicable and coincidence effects are important. However, there has been only limited attention given to low-frequency sound transmission. Low-frequency sound transmission is stiffness controlled in the frequency region below structural resonances, and damping and stiffness controlled at resonances. Furthermore, the closed receiving room (cavity) behind the panel may have a significant effect on the panel dynamics and, therefore, the sound transmission. A panel backed by a closed cavity provides a meaningful model for studying low-frequency sound transmission of the type encountered in aircraft for example. Although some theoretical work has been done on the subject, only limited experimental data are available in the literature. The purpose of this paper is to present some experimental findings on the effects of panel stiffness and receiving space absorption on low-frequency sound transmission through panels into closed spaces. In addition, a simplified method for calculating the low-frequency noise reduction of a cavity-backed panel will be presented.

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INTRODUCTION

A considerable amount of information has been developed in architectural acoustics on sound transmission through panels, with empahsis on middle and high frequencies where "mass law" and coincidence effects are important. However, there has been only limited attention given to low-frequency sound transmission, where stiffness, damping, and resonant behavior are important. Furthermore, the receiving room behind the panel may have a significant effect on the panel dynamics and, therefore, the sound transmission. A panel backed by a closed cavity provides a meaningful model for studying low-frequency sound transmission of the type encountered in light aircraft, for example.

The sound transmission characteristics of a simple panel backed by a closed, absorbent cavity are shown in figure 1. The measure of sound transmission for this case is noise reduction (NR) and is expressed in decibels as a function of frequency. The NR is the difference in noise level (sound pressure level) across the panel. The frequency range for convenience is broken up into four regions corresponding to dominating physical mechanisms, as shown in the figure. The stiffness and resonance regions represent the low-frequency portion of the frequency range shown. As shown in figure 1, the noise reduction below the first resonance is independent of frequency. This phenomenon has been studied theoretically by several investigators (for example, references 1, 2, 3). However, very little data are available demonstrating this noise reduction and also data are not available showing the transistion of noise reduction from this stiffness region into the mass law region (some data are available in references 3, 4, 5). In addition, the theoretical work in references 1, 2, 3, 4 was for panels backed by hard-walled cavities.

The purpose of this paper is to present experimental data showing the effects of adding stiffness to a panel backed by a closed, absorbent cavity, and to present a simple theory for predicting the noise reduction for the stiffness, resonance, and mass regions of a panel backed by a closed, absorbent cavity.

TEST APPARATUS

The test apparatus used in this study is shown in figure 2 and consisted of a cavity having high transmission loss on five sides, with the sixth side left open for mounting test panels. The cavity was 30 cm by 38 cm by 45 cm deep and was lined with 2.5 cm fiberglass on the sides and 7.5 cm fiberglass on the cavity bottom.

The acoustic excitation was normally incident white noise provided by two loudspeakers. The exterior noise level was determined by averaging the outputs of two microphones near the panel surface in one-third octave bands. Similarly, the inside noise level was determined by averaging the outputs of two microphones in one-third octave bands. The noise reduction was obtained by subtracting the average inside level from the average outside level in each one-third octave band.

Three of the four test panels used in this study are shown in figure 3. Each panel is 30 cm by 38 cm and each was clamped to the cavity. The fourth panel (not shown in the figure) was lead-loaded vinyl having the same surface density as the simple panel shown in figure 3. The three panels are each 0.79 mm thick aluminum. The only physical differences in these panels is the stiffeners which were riveted onto the panels as indicated in the figure. These three panels plus the fourth (lead vinyl) represent a large variation in panel stiffness with very little variation in panel mass (0.22 to 0.33 kg/m^2) and, thus, provide a means for studying the effects of stiffness on noise reduction.

RESULTS

Experimental Data

The measured noise reduction for the four panels is shown in figure 4. A calculated mass law curve for the simple aluminum panel is included with each set of data for reference. The data show that adding stiffness increases the noise reduction at frequencies below the fundamental. The data also show that adding stiffness does not significantly change the noise reduction above the first resonance. It is evident from the data that only the fundamental mode has a significant effect on the noise reduction. Finally, the data indicate that adding stiffeners to the aluminum panels not only increased the frequency of the first mode, but also reduced the severity of the resonance dip. The lead vinyl does not follow this trend probably because of the very high internal damping in the panel. The effects of damping will be discussed later in the paper.

Analytical Model

Because only one panel resonance appears in these experimental results, a theoretical model using one panel mode seems in order. The model is illustrated in figure 5. The model consists of a rigid, infinite panel supported on springs and backed by a wall having a frequency-dependent impedance. There are no corrections for finite size effects such as defraction. This model is analogous to that described in reference 6, page 82, with the exception that the backing wall has impedance Z_ℓ instead of being a hard wall ($Z_\ell \rightarrow \infty$).

The equation of motion and the corresponding equation for pressure inside the cavity are shown in figure 5 as equation I. The variables for a particular panel cavity system are frequency and the backing-wall impedance. A simple, frequency-dependent expression for Z_ℓ is needed to complete the equation.

The impedance of fiberglass with a hard wall behind it is approximately ρc (real) at high frequencies. At low frequencies, the fiberglass has virtually no effect, so the wall impedance will approach infinity (assumed real) as frequency goes to zero. Therefore a real, assumed relationship for Z_ℓ that varies from infinity at zero frequency to ρc at high frequencies is appropriate. Such a relationship is given in equation II.

It may be shown from equations I and II that for low frequencies, the noise reduction becomes a function simply of panel and cavity stiffness (independent of frequency as expected) as shown in equation III. It may be observed from this equation that doubling the panel natural frequency may increase NR by up to 12 dB.

It may also be shown that, provided the first panel mode is lower than the first depthwise acoustic mode, the system resonance can be expressed as in equation IV, where it may be observed that the effect of the cavity is simply to add a stiffness term to the stiffness of the panel.

At high frequencies, the noise reduction reduces to mass law (equation V) plus 6 dB for pressure doubling at the surface. This was also expected because the absorbent cavity acts like an acoustic termination at high frequencies since sound waves radiated by the panel are not reflected back to the panel.

If the cavity impedance (Z_{cavity}) equals ρc, equation I becomes applicable to the transmission loss problem of reference 6, page 80, except panel damping is included. It may be shown for this case that the transmission loss (TL) at resonance can be expressed as in equation VI. It may be observed from this equation that the TL at resonance increases as much as 6 dB for a doubling of either damping ratio (ζ), natural frequency (ω_n), or panel mass (m), provided the remaining two variables (i.e., ζ, m, or ω_n) are held constant. Thus, by increasing the resonance frequency, ω_n, (with constant m, ζ) the transmission loss at resonance increases as well as the TL below resonance.

Using equations I and II of figure 5, the noise reduction was calculated and plotted for each of the four panels used in the study. These calculations are shown in figure 6 with the corresponding data for each panel. It may be observed from the figure that the model provides a reasonable method for calculating the noise reduction at low and mid-frequencies. It should be noted that the prediction is poor for the aluminum panels at high frequencies because coincidence effects are not included in the model and because the added mass of the stiffeners, although included in the prediction model, is not uniformly distributed over the panel.

Conclusions drawn from the data presented in this report include: (1) adding stiffness to a panel backed by a closed cavity increases the noise reduction at frequencies below the first resonance, (2) additional stiffness does not improve noise reduction at mid and high frequencies, (3) for a random input and one-third octave band analysis, only the first panel-cavity mode has a significant effect on the noise reduction, (4) as the resonance frequency of the first panel-cavity mode was increased with stiffeners for the aluminum panels, the noise reduction at the resonance dip was increased, (5) the results of the simple analytical model presented compared well with the data for the panels tested.

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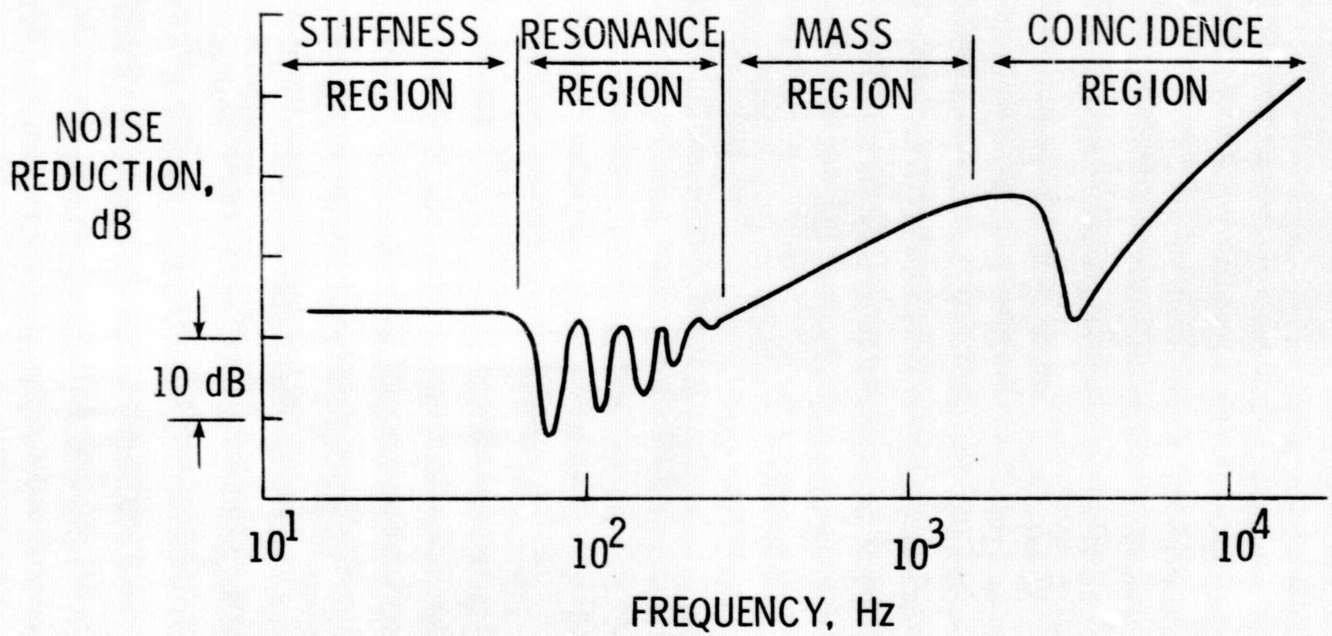


Figure 1.- Predicted noise reduction for cavity-backed panel (absorbent cavity).

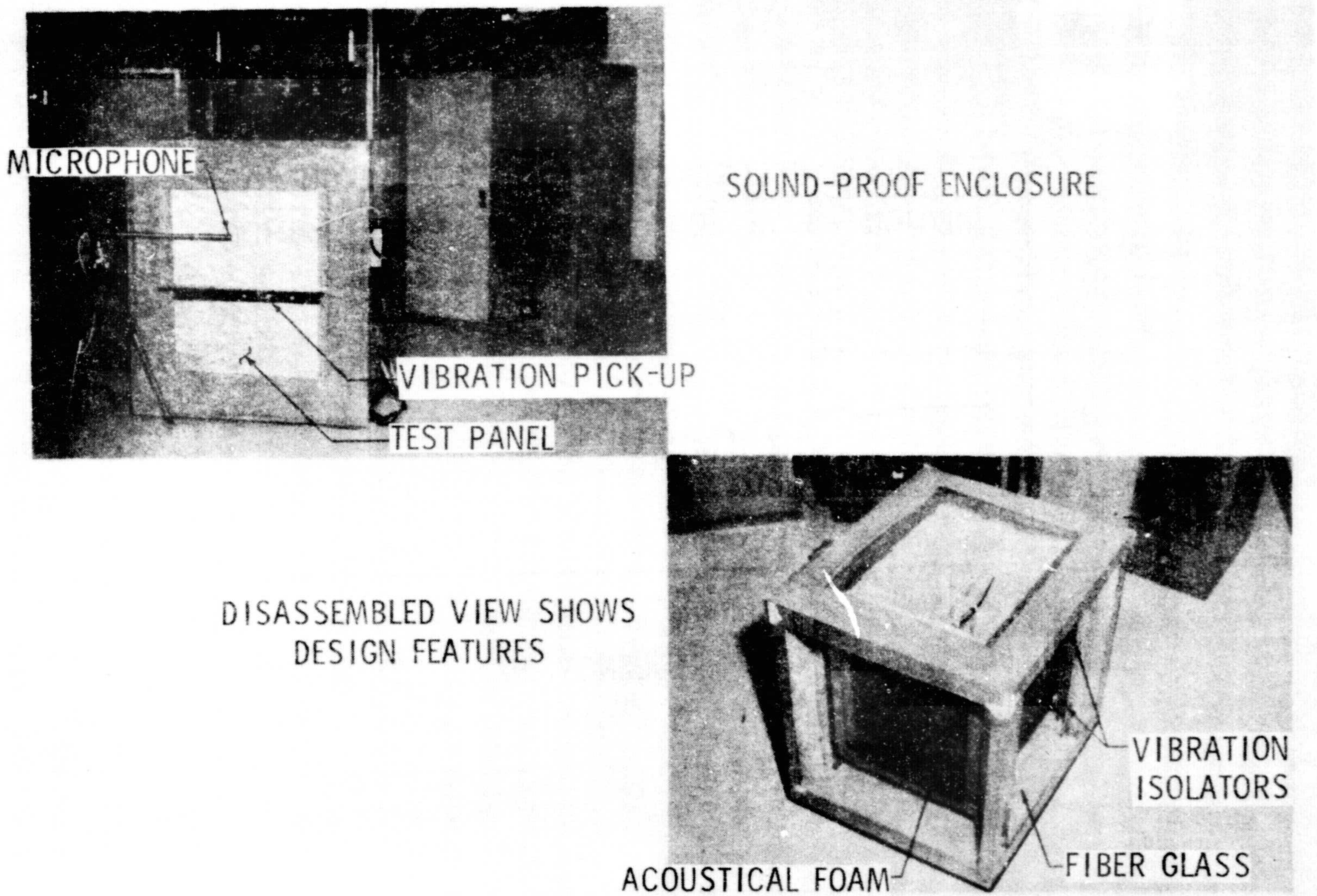


Figure 2.- Noise transmission test apparatus.

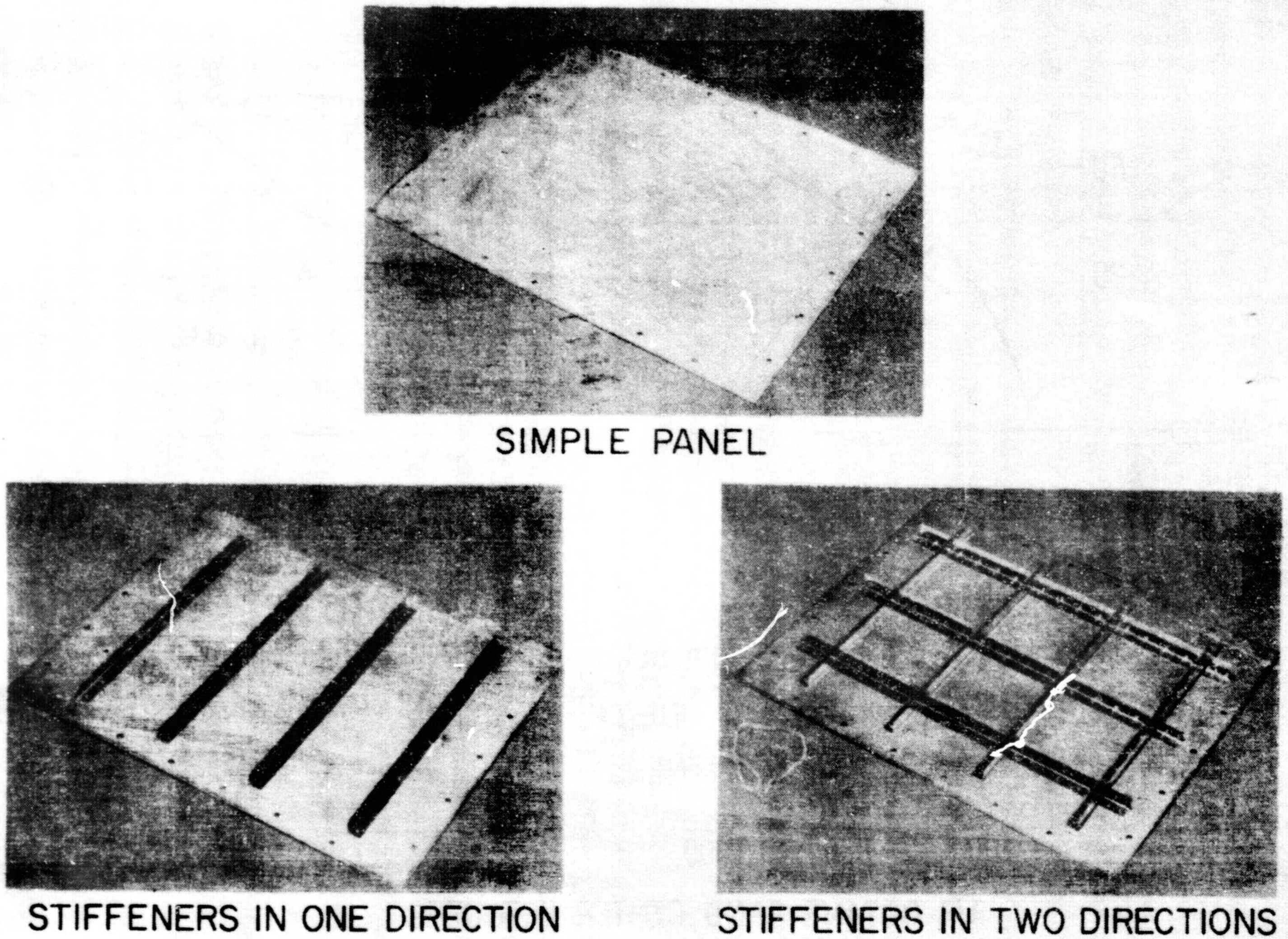


Figure 3.- Noise transmission test panels.

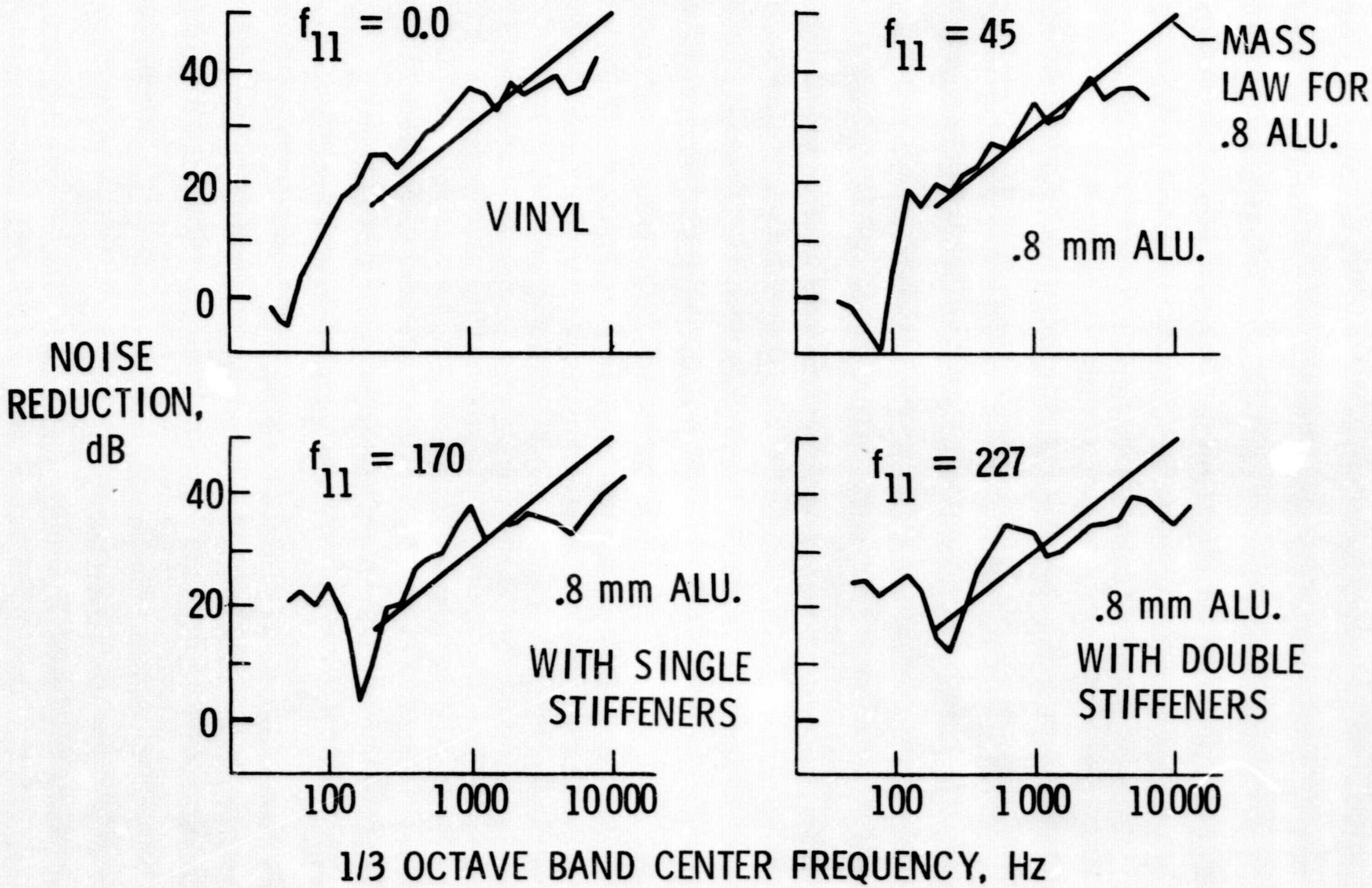


Figure 4.- Experimental effects of stiffness on noise reduction.

$$\left.\begin{aligned} P_o &= \dot{\eta}\, Z_{panel} + \dot{\eta}\, Z_{cavity} + \dot{\eta}\rho c \\ P_i &= \dot{\eta}\, Z_{cavity} \\ Z_{panel} &= 2\zeta m\, \omega_n + j(\omega m - \bar{k}/\omega) \\ Z_{cavity} &= \rho c\, \frac{Z_\ell + j\rho c \tan(k\ell)}{\rho c + jZ_\ell \tan(k\ell)} \end{aligned}\right] \quad \mathrm{I}$$

$$Z_\ell \simeq \rho c \left\{ 1 + \left(\frac{\omega_{100}}{2\omega}\right)^2 \right\} \quad \mathrm{II}$$

$\omega \to 0$:

$$NR \simeq 20 \log \left\{ \frac{k_{cavity} + \omega_n^2 m}{k_{cavity}} \right\} \quad \mathrm{III}$$

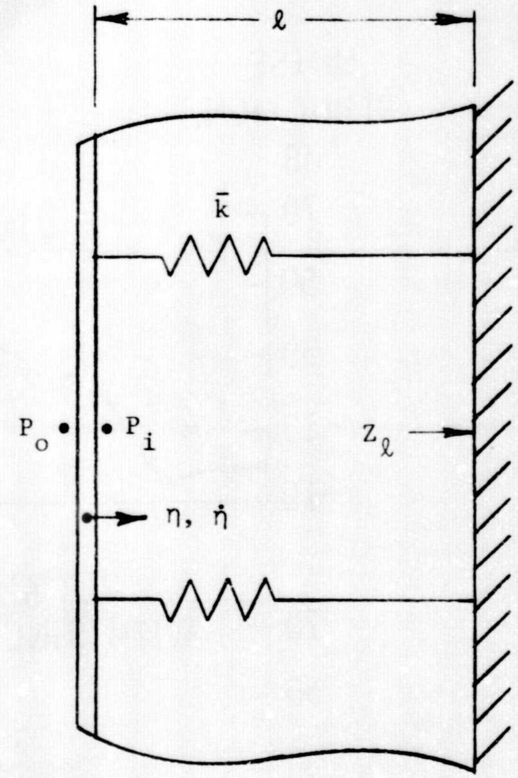
$\omega_n < \omega_{100}$:

$$\omega_n' = \sqrt{\omega_n^2 + \rho c^2/\ell m} = \sqrt{\frac{\bar{k} + k_{cavity}}{m}} \quad \mathrm{IV}$$

$\omega \to \infty$

$$NR \simeq 10 \log \left\{ 1 + \left(\frac{\omega m}{2\rho c}\right)^2 \right\} + 6 = TL_{\substack{mass\\law}} + 6 \quad \mathrm{V}$$

$$TL_{\omega=\omega_n} = 20 \log \left\{ 1 + \left(\frac{\zeta m \omega_n}{\rho c}\right) \right\} \quad \mathrm{VI}$$



$$\bar{k} = \frac{\text{panel stiffness}}{\text{unit area}}$$

$$\bar{k} = \omega_n^2\, m$$

$$m = \frac{\text{panel mass}}{\text{unit area}}$$

$$k_{cavity} = \rho c^2/\ell$$

Figure 5.- Mathematical model for cavity-backed panel.

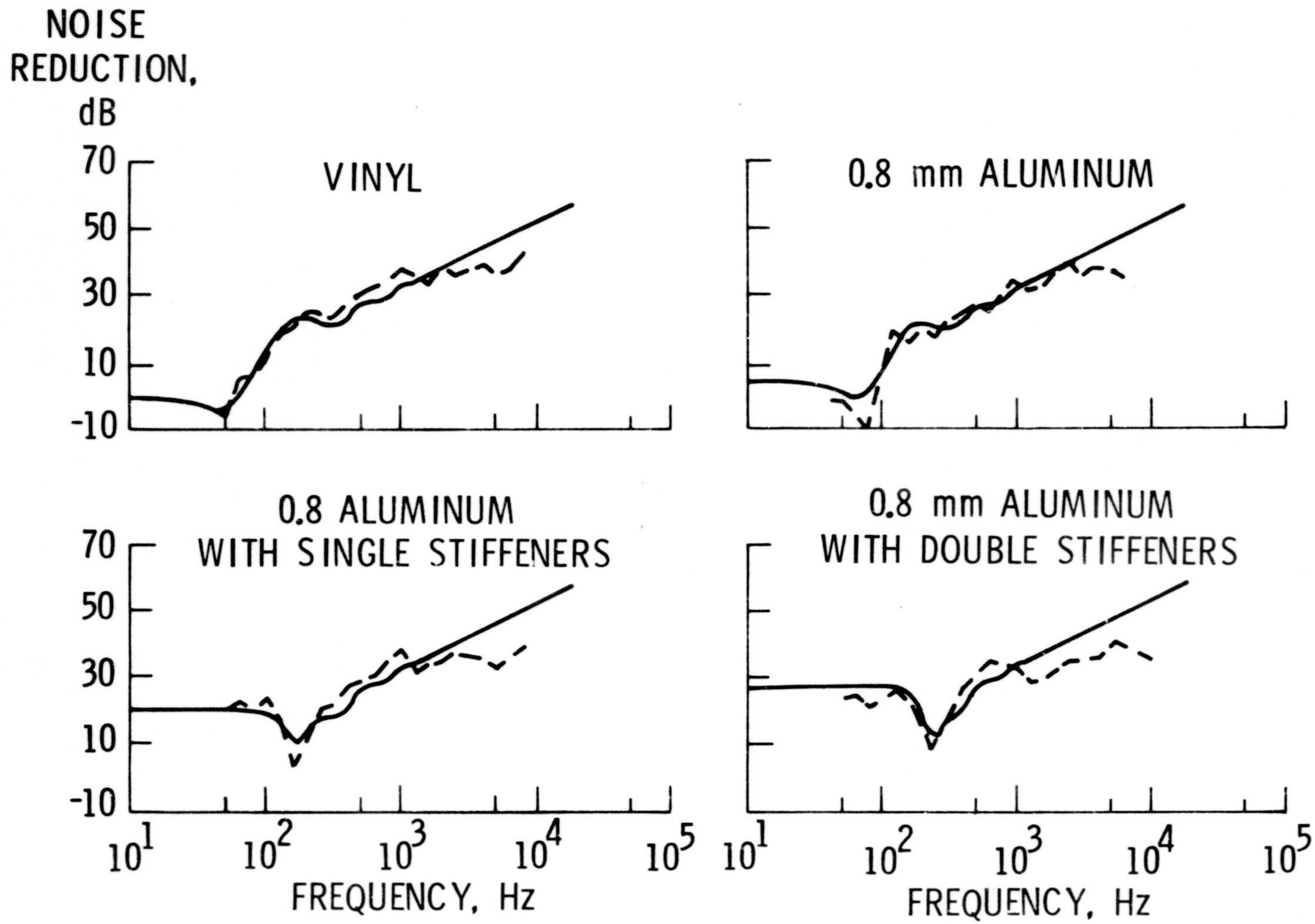


Figure 6.- Comparison of analytical and experimental results.