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5-1992

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SOUND TRANSMISSION THROUGH STIFFENED DOUBLE-PANEL STRUCTURES LINED WITH ELASTIC POROUS MATERIALS

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ABSTRACT

This paper presents transmission loss prediction models for a periodically stiffened panel and stiffened double-panel structures using the periodic structure theory. The inter-panel cavity in the double-panels structures can be modelled as being separated by an airspace or filled with an elastic porous layer in various configurations. The acoustic behavior of the elastic porous layer is described by a theory capable of accounting fully for multi-dimensional wave propagation in such materials. The predicted transmission loss of a single stiffened panel is compared with the measured data.

INTRODUCTION

Aircraft fuselage structures are generally double panel structures consisting of an outer surface or skin and an inner surface or trim panel. The gap between the skin and the trim panel is usually filled with fiberglass blankets. The fuselage is stiffened by periodically arranged frames and stringers. For predicting sound transmission loss, the fuselage is often modelled as a plane, infinite double panel. The skin and trim panels are modelled as isotropic plane panels having the average "smeared" stiffness and mass of the real fuselage. The infinite panel modeling of the double panel structure is known to give poor results in the low-frequency region.

Further, the panels are separated by an unsegmented space of uniform depth that may be filled with an assumed rigid porous material. The current prediction models allow only a single longitudinal wave type in the porous material. These models may be appropriate for medium density fiberglass but not for newer materials like polyamide foams. It is now known that three wave types, two longitudinal and one transverse, can propagate in porous material. All these wave types must be included in the porous material theory if accurate sound transmission loss predictions are to be made. Thus there is a need to develop a prediction model that can account for these effects.

The transmission loss prediction model developed in this paper considers the modal character of sub-panels of a stiffened panel and also includes the effects of elastic porous material lining. The prediction model for stiffened panels is based on the periodic structure theory. The inner and outer panels are still assumed to be infinite but are supported by finite stiffness elements at regular intervals. These infinite panels consist of an infinite array of finite panels. Due to the finite nature of the sub-panels, their response is of modal character. Thus the low frequency response of the panel is determined by the mass and flexural stiffness of the panel along with the panel boundary conditions. This contrasts with the infinite panel model in which the low frequency panel response is determined solely by the panel mass.

The elastic porous material theory which is capable of describing all significant known effects of sound propagation in porous materials is used to model the effects of porous material lining. Specifically, it allows for the existence of two longitudinal and a single transverse wave type and can be used to represent both fiberglass and foam. The elastic porous material theory allows for the shear stiffness, as well as for the in vacuo bulk modulus of elasticity, loss factor, Poisson's ratio, porosity, structural factor and flow resistance. Inertial and viscous effects result in coupled motions of the solid and fluid phases of elastic porous materials. Viscous effects, internal friction and heat transfer are the mechanisms causing energy dissipation which makes these materials efficient in noise control applications.

THEORETICAL BACKGROUND

Plates with parallel, equally spaced line stiffeners are common in aircraft and marine structures. At very low frequencies, the flexural wavelength in the plate is much longer than the stiffener separation, and the periodic structure behaves like an orthotropic plate. However as the frequency increases, the flexural wavelength becomes comparable with stiffener separation. In this frequency range, the response of periodic structures to harmonic excitation can be expressed in terms of an infinite series of spatial harmonics based on the stiffener spacing. This approach was used by Mead, Pujara and Mace [1,2,3] to investigate the response of periodically stiffened beams and plates.

All of the theory to be discussed in this paper is adapted from work available in the open literature, most notably work by Mead and his colleagues [1,2,3] and by Bolton and Shiau [4]. The acoustic behavior of the porous layer is modelled using the theory developed by Bolton and Shaiu [4] that takes into account all three types of waves known to propagate in an elastic porous material, including two longitudinal waves and one transverse wave.

If a panel is stiffened in one direction, the two-dimensional, periodically stiffened panel can be reduced to a one-dimensional, periodically supported beam. That transformation is illustrated in Figure 1. Since the beam is periodically supported, satisfactory solutions for its flexural vibration as well as the reflected and transmitted acoustic waves can be expressed in the form of a particular series of spatial harmonics. In the present work, the beam is treated as Euler-Benoulli beam and the supports possess mass, rotational and translational stiffnesses. Mead and Pujara [2] have given the relevant theory of the space harmonic solution to the periodically supported beam based on the principle of virtual work.

Due to the periodic characteristics of the structure, the flexural motion of panel can be expressed in terms of a series of space harmoincs: i.e.,

$$W(x) = \sum_{n=-\infty}^{+\infty} A_n e^{-j\frac{\mu+2n\pi}{L}x} e^{j\omega t} \quad (1)$$

where W(x) is the panel transverse displacement, L is spacing of stiffeners, and μ is the characteristic propagation constant.

This series must be made to satisfy the boundary conditions at elastically restrained supports by appropriate restrictions on the coefficients A_n. The wave velocity potentials on the incident and transmitted side of the panel can be similarly expanded in terms of series of space harmonics:

$$\Phi_1(x,y) = e^{-jk_x x} e^{-jk_{y0} y} \sum_{n=-\infty}^{+\infty} B_n e^{-j\frac{\mu+2n\pi}{L}x} e^{jk_{yn}y} e^{j\omega t} \quad (2)$$

$$\Phi_2(x,y) = \sum_{n=-\infty}^{+\infty} C_n e^{-j\frac{\mu+2n\pi}{L}x} e^{-jk_{yn}y} e^{j\omega t} \quad (3)$$

where $\Phi_1(x,y)$ and $\Phi_2(x,y)$ are the velocity potentials in the incident and transmitted regions, respectively, and k_x, k_z are components of the acoustic wave number along x and z axes respectively. The wave number component along y axis can be obtained from the following wave number relationship:

$$k_{yn} = \sqrt{(\frac{\omega}{c})^2 - (k_x + \frac{\mu+2n\pi}{L})^2 - k_z^2} \quad (4)$$

where ω is the radian frequency and c is the speed of sound.

The modal amplitudes of the reflected and transmitted waves on the beam can be expressed in terms of modal amplitudes of flexural wave in the panel by applying boundary conditions requiring continuity of normal velocity at the panel surface.

The equation of motion for the unstiffened panel is:

$$D\frac{d^4W}{dx^4} - m_b\omega^2 W - j\omega\rho_o(\Phi_1 - \Phi_2) = 0 \quad (5)$$

where D is the panel flexural stiffness per unit width, m_b is the panel mass per unit area and ρ_o is the density of air.

Based on the principle of virtual work (which states that if a system is in equilibrium, the total virtual work done by the applied forces in any virtual displacements compatible with the constraints is equal to zero), the governing equations can be derived which in turn can be rearranged to obtain a complex matrix equation in the unknown modal amplitudes of the flexural wave on the beam. The application of this principle in the present case results in the requirement that the sum of the contributions to the virtual work from one periodic panel element (without stiffener) should be equal to zero. The virtual work for one periodic element (without stiffener) can be calculated from virtual displacement and integrating over the length of one periodic element. The virtual work for the stiffener motion can be obtained for the appropriate elastic restraint characteristics of the stiffener. For the case of a locally-reacting stiffener (assumed to possess mass, and translational and rotational stiffnesses that are constant and independent of frequency or incidence angle), the virtual work for the mass, translational and rotational stiffnesses of stiffener located at $x = 0$ are obtained by multiplying the associated inertia force, translational spring force and rotational spring moment by virtual displacement and virtual angular displacement at $x = 0$.

The transmission loss (TL) of the stiffened panel can now be defined as:

$$TL = 10 \log \frac{I_i}{I_t} \quad (6)$$

where I_i and I_t are incident and transmitted normal intensities respectively and are given by:

$$I_i = \frac{\omega\rho_o k_{y0}}{2} \quad (7)$$

and

$$I_t = \frac{\omega\rho_o}{2}\sum_n |C_n|^2 Re[k_{yn}] \quad (8)$$

The transmission loss characteristics of periodically stiffened double-panel structures was also modelled using the space harmonic expansion approach. The two panels can either be separated by an air cavity or filled with a layer of porous material. The acoustical behavior of a porous layer between the two peridically stiffened panels is modelled using the theory developed by Bolton and Shiau [4] that takes into account all three types of waves known to propagate in an elastic porous material. The two panels, lined with elastic porous material, and assumed to be periodically reinforced in one direction are shown in Figure 2. Due to the periodic nature of the structure, all the field variables can be expressed in terms of series of space harmonics, including the six field variables in the porous layer which are the downward and upward traveling wave components of two longitudinal waves and one transverse wave. The flexural motion of panels, for example, can be expressed in terms of series of space harmoincs; i.e.,

$$W_1(x) = \sum_{n=-\infty}^{+\infty} A_{1n} e^{-j\frac{\mu+2n\pi}{L}x} e^{j\omega t} \quad (9)$$

$$W_2(x) = \sum_{n=-\infty}^{+\infty} A_{2n} e^{-j\frac{\mu+2n\pi}{L}x} e^{j\omega t} \quad (10)$$

As in the single panel case, boundary conditions are used to express the modal amplitudes of the panel's flexural motion. The continuity of normal velocity at both faces of both panels is then used to determine relationships between the modal amplitudes of the acoustic fields and those of the flexural motions of the panels. Application of the principle of virtual work to the two panels and to the corresponding stiffeners then yields two infinite sets of simultaneous equations. The transmission loss of the double-panel structure can be defined using the incident and transmitted intensities following the procedure outlined for the single stiffened panel earlier.

DESCRIPTION OF TEST PROGRAM

Experiments were conducted to measure the transmission loss of a baffled, stiffened flat panel (dimensions: 1.75m × 1.14m × 1.27mm). The panel was mounted in the window section of the Douglas Aircraft Company transmission loss test facility consisting of two anechoic rooms. The panel was excited by a loudspeaker array, placed in the source room. The sound intensity measurements over the panel surface were made using the Norwegian Electronics Real Time Analyzer 830 and Type 216 intensity probe. The transmission loss of a bare stiffened panel was measured for comparison with predicted results.

DISCUSSION OF RESULTS

Experimental TL data of stiffened test panels has shown that stiffeneing effects are important in the low to mid- frequency region (50 - 1000 Hz). Past experience has also shown that the mass-law, which is based on infinite panel theory, does not provide good estimates of TL of a stiffened panel at these frequencies. The transmission loss (TL) of the two configurations of a stiffened panel were predicted using the new TL prediction schemes.

Although the baseline (bare) test panel is stiffened in two-directions with frames and longerons, it was modelled with one-dimensional stiffeners accounting for the main frames with 19 inches spacing. The normal and oblique incidence (0 and 45 degrees) TL predictions for the test panel are compared with the experimental data and with the mass-law estimates in Figure 3. It may be observed from these comparisons that the mass-law predictions for the test panel not only underestimate the TL but also fail to follow the main trends of the experimental curve. The normal and oblique incidence stiffened TL predictions of the test panel, on the other hand, provide improved TL estimates. The peaks (or humps) and valleys of the predicted TL curve also appear to follow those observed in the experimental curve. The new stiffened panel TL prediction model, therefore, provides improved modelling of such structures.

In the next phase of the program, the transmission loss of a stiffened double panel structure lined with elastic porous foam will be measured and compared with the predicted TL data.

CONCLUSIONS

This paper presents analytical modelling of sound transmission through stiffened panels lined with elastic porous materials. A new transmission loss prediction model for a fuselage sidewall consisting of stiffened double-panel structures is developed using the periodic structure theory. In the case of double-panel structures, the panels can be separated by airspace. The inter-panel cavity can also be modelled as being filled with an elastic porous layer in various configurations.

The predicted normal and oblique transmission loss values of a bare stiffened panel are compared with the experimental data. The predicted TL curve is in good agreement with the experimental TL values. The predicted TL curve also follows the trends observed in the experimental TL curve. The mass law predictions for the test panel, on the other hand, not only underestimated the TL but also failed to follow the main trends of the experimental curve.

ACKNOWLEDGEMENTS

This research work was conducted under the McDonnell Douglas Independent Research and Development program.

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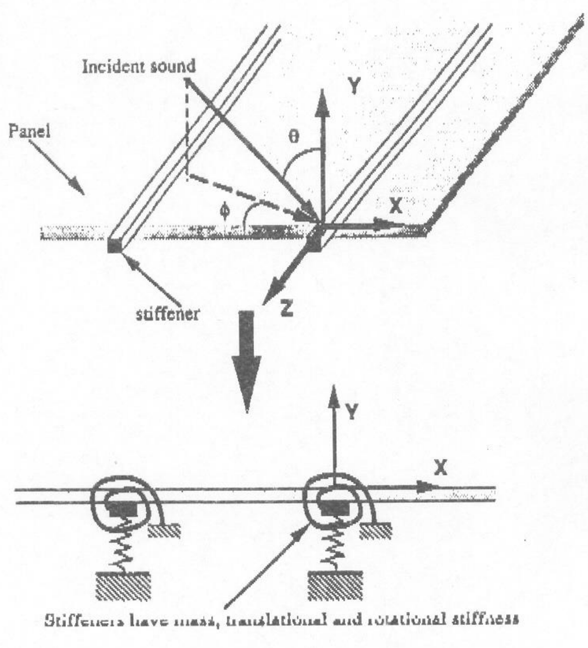

FIGURE 1. Schematic Representation of a Panel Stiffened in One Direction.

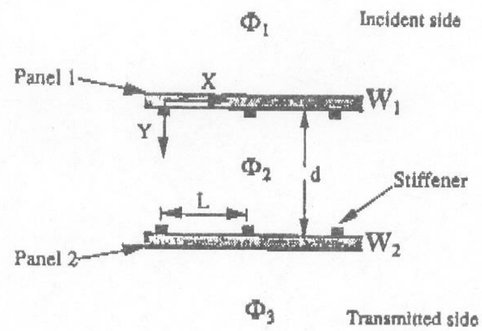

FIGURE 2. Schematic Representation of an Unlined Double-Panel.

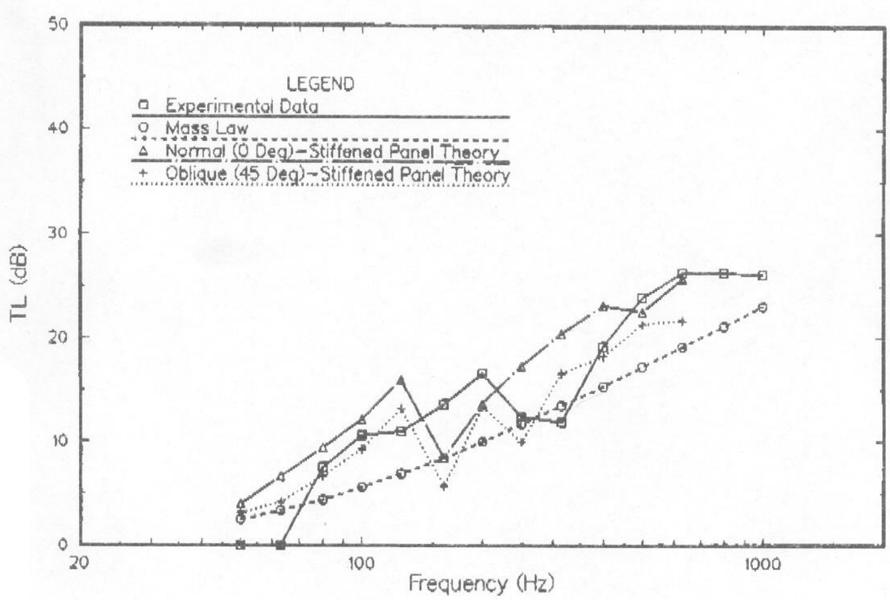

FIGURE 3. Comparison of Predicted and Measured Transmission Loss of a Bare Stiffened Panel.