WAVE PROPAGATION IN CURVED LAMINATE COMPOSITE STRUCTURES

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1. INTRODUCTION

The principal aim of this work is to present a model to compute the transmission loss of sandwich composite cylindrical shells. The effects of membrane, bending, transverse shearing as well as rotational inertia are considered in all of the layers composing the structure. The elastic constants of any layer are related to the orthotropic angle-ply defined as the angle of the principal directions of the layer's material to the global axis of the shell. Fundamental relations are expressed using the dynamic equilibrium relations of the unit forces in the structure. A general eigenvalue approach to compute the dispersion curves of such structures is presented. Using these curves, the radiation efficiency, the modal density, the group velocity, the resonant and non-resonant transmission coefficients are computed and used within SEA framework to predict the sound transmission loss of these structures. The described model is shown to handle accurately, both laminate and sandwich composite shells. Comparisons with existing models and experimental data are also discussed.

2. THEORY

This paper describes the SEA modeling of the transmission loss through finite laminate and sandwich composite curved panels. Both laminate composite and sandwich composite are modeled using a discrete thick laminate composite theory. The studied transmission problem has three primary resonant systems such as: two reverberant rooms separated by the composite curved panel. The dispersion curves of the structure are used to compute the modal density and the radiation efficiency. Several models to compute the radiation efficiency were tested[1,2,3]. Identical results were obtained but the model of Leppington[3] was preferred for its accuracy and fast convergence. These indicators allow the calculation of the radiation loss factor and also the resonant contribution of the transmission loss. The standard flat panel theory[15] is used to compute the non-resonant transmission but it is adapted here to the particular vibration behaviours of the curved panels. In particular, a sub-coincident modes selection method is used to compute the non-resonant transmission contribution. Moreover, the classical wave approach non-resonant contribution is corrected using the spatial windowing method presented in reference [2]. Finally, a transmission loss experimental result of a curved sandwich composite panel is successfully compared with numerical estimation.

2.1. Geometry

Figure 1 represents the global geometrical configuration of the composite shell, where R is the curvature radius and h is the total thickness. The layered construction is considered, asymmetrical.

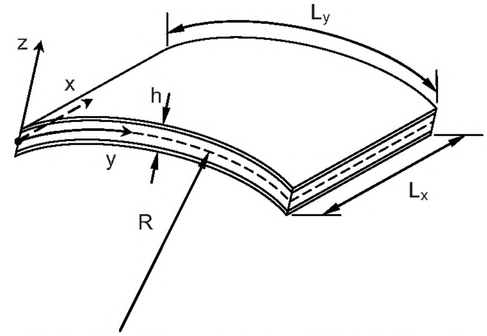

Figure 1. The composite shell co-ordinates

2.2. Dispersion relation

The displacement field of any discrete layer "i" of the panel is of Mindlin's type. For any layer of the shell, Flüge's theory is used to describe the strain-displacement relations. The resulting dynamic equilibrium system has 5N+3(N-1) variables regrouped in two vectors; a displacement–rotation vector {U}, and an interlayer forces vector {F}. The associated 5N+3(N-1) equations are composed of 5 equations of dynamic equilibrium for each of the N layers plus 3 equations of interlayer continuity of displacements for each of the N-1 interlayer surfaces.

To solve for the dispersion relations, the system of dynamic equilibrium equations is expressed in terms of a hybrid displacement-force vector $<e>^T=\{U\ F\}$.

Assuming $<e>=\{e\}\exp(jk_x x+jk_y y-j\omega t)$, a harmonic solution, the system is expressed in the form of a generalized polynomial complex eigenvalue problem:

$$k_c^2[A_2]\{e\}-ik_c[A_1]\{e\}-[A_0]\{e\}=0 \qquad (1)$$

where, $k_c=\sqrt{k_x^2+k_y^2}$, $i=\sqrt{-1}$ and $[A_0]$, $[A_1]$, $[A_2]$ are real square matrices of dimension 5N+3(N-1). Relation (1) has 2(5N+3(N-1)) complex conjugate eigenvalues and represents the dispersion relations of the laminated composite shell.

At any heading direction the curved panel has two propagating solutions below the ring frequency. At the ring frequency a third solution becomes propagating thus, in the dispersion field context the ring frequency is mathematically perceived as a cut-off or transition frequency. Two other cut-off frequencies appear at high frequencies where two additional solutions become propagating.

2.3. Transmission problem

Below the ring frequency, the non-resonant transmission is dependent on the direction of the incident acoustic waves. For a given excitation frequency band (with ω_{cen} the center band frequency and ω_1, ω_2 the frequency limits of the band), and an incidence direction (θ, φ) the structural and the forced wave numbers are calculated from the dispersion relation (1) and the following condition is checked to ensure that the forced modes are non-resonant:

$$k_0(\omega_{cen})\sin\theta < k_s(\omega_1) \quad \text{OR} \quad k_0(\omega_{cen})\sin\theta > k_s(\omega_2) \cdot \qquad (2)$$

This accounts for both mass and stiffened controlled non-resonant modes. Usually, stiffened controlled modes contribution is neglected and the mass-controlled non resonant transmission coefficient is used. The allowable heading directions are obtained using the dispersion equation (1) and the first condition in (2).
In order to improve the low frequency predictions of the non-resonant transmission coefficient, a geometrical windowing correction method is also used. The correction method used here, is detailed in reference [2] and examples of its validation are given in references [2] and [4].
The resonant transmission coefficient is calculated from the radiation efficiency of the panel and its modal density.
A simple SEA acoustic transmission scheme is used here and consists of two reverberation rooms separated by the studied curved panel. One of the rooms is excited by a diffuse field and the acoustic transmission problem is assumed to encompass two transmission contributions: resonant and non resonant transmission.

3. EXPERIMENTAL VALIDATION

Transmission loss tests were performed on the singly curved sandwich composite panel described in the previous section. The tests were performed at the Canadian National Research Center transmission loss facility located in Ottawa. Measured transmission loss and predictions with both the wave approach and the modal approach are given in Figure 2. It should be noted at this stage that the mounted panel damping was not measured, and that a nominal modal damping ratio of 2.5% was assumed in the analysis. The wave approach with the geometrical correction leads to a very good agreement throughout the frequency range of the test apart from the ring frequency region. On the other hand, the modal approach shows a higher transmission loss than the test below the panel ring frequency. Above the panel critical frequency, both the wave and modal approaches yield almost identical results. Note that the number of resonant modes is not sufficient in the first 1/3 octave bands for the modal method to be reliable at low frequencies (less than 1 mode at 100 Hz).

4. DISCUSSIONS AND CONCLUSIONS

An efficient model to compute the transmission loss of sandwich and laminate composite curved panels was developed. The physical behaviour of the panel is represented using a discrete lamina description. Each lamina is represented by membrane, bending, transversal shearing and rotational inertia behaviours. The model is developed in the context of a wave approach. Using the dispersion relation's solutions, the modal density, the radiation efficiency as well as the resonant and non-resonant transmission loss are calculated. The acoustic transmission problem is represented within statistical energy analysis context using two different schemes for the non-resonant path. It is observed that for the presented problem, the modal energy exchange is dominated by the first wave solution. It is concluded that for classical acoustic transmission problems, the SEA scheme used here is accurate. The results were compared successfully to the transmission loss test of a singly curved sandwich panel and to two other models. In particular, the presented model is applicable to both sandwich panels and composite laminate panels with thin and/or thick layers.

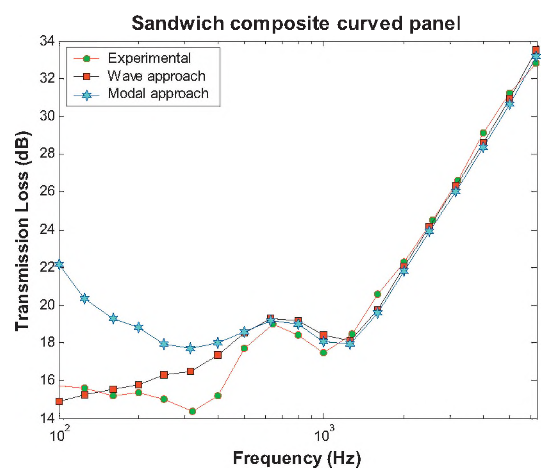

Figure 2. Total transmission loss of a sandwich composite curved panel.

REFERENCES

1. Villot M, Guigou C. and Gagliardini L, "Predicting the acoustical radiation of finite size multi-layered structures …", JSV, 245(3), 433-455, 2001.
2. Ghinet S and N. Atalla « Sound transmission loss of insulating complex structures », Canadian Acoustics, Vol. 29, 3, 26-27, 2001.
3. Leppington F. G, Broadbent E. G. and Heron K. H. "The acoustic radiation from rectangular panels with constrained edges", Proc. R. Soc. Lond. A382, pp 245-271, 1982
4. Osman H., Atalla N., Atalla Y., Panneton R. "Effects of acoustic blankets on the insertion loss …", Tenth international congress on sound and vibration, ICSV 10, Stockholm, Sweden, July 2003