inter•noise 96
25th Anniversary Congress – Liverpool

INCE: 43

PREDICTION OF STRUCTURE-BORNE SOUND IN BUILT-UP PLATE STRUCTURES

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1. INTRODUCTION

A waveguide model is proposed to predict the propagation of structure-borne sound in built-up plate structures, as encountered in passenger and freight transport. Applications are of interest in noise control at the first-stage design of marine and aircraft structures or engine foundations.
The waveguide model, based on the concept that vibrational power propagates mainly as flexural waves throughout the structures, compares well with experimental results as shown in [1], [2] and [3].
Mathematical expressions obtained from the flexural wave equation, solved for single plate elements, provide a calculation technique using transfer matrices relating exciting moments and angular displacements at the plate junctions. These parameters enable the computation of power dissipated in the plate elements. The calculation procedures are implemented in computationally efficient programs demonstrating how various parameters affect power transmission throughout the structures. By means of computer simulations, local damping influence and exciting force locations are readily analysed for a defined structure.

2. VIBRATIONAL POWER BETWEEN COUPLED PLATES

Analysis of a Plate System
Consider the structure, in Fig. 1, consisting of a set of four thin rectangular plates 1, 2, 3 and 4, rigidly coupled at junctions 2, 3 and 4. The extreme sides of the system are assumed to be simply supported in the width (parallel to the y-axis) and the length (parallel to the x-axis). All the plates having the same width L_y are made of an isotropic material and may be defined with different lengths $L_{x,p}$ and thicknesses t_p (where plate index p = 1, 2, 3 and 4). A damping η_p is taken into account by using a complex Young's modulus $E_p = E_0\,(1+i\eta_p)$. The point force F exerted on plate 1 at (x_0, y_0) is assumed to be harmonic and expressed by

$$\vec{F} = -F_0 \cdot e^{i\omega t} \cdot \delta(x - x_0) \cdot \delta(y - y_0) \cdot \vec{e}_z$$

driving plate 2 and subsequently plates 3 and 4, causing bending moments at the junctions.

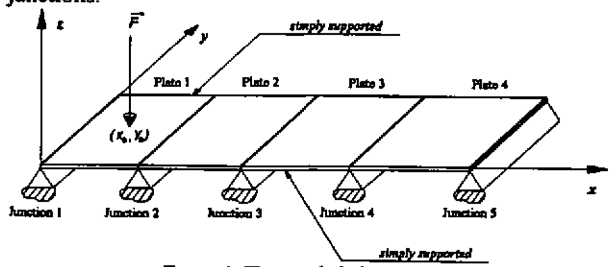

Figure 1. The coupled plate system.

Plate dimensions
$L_{x,1}$ = 0.15 m, t_1 = 0.03 m
$L_{x,2}$ = 2 m, t_2 = 0.012 m
$L_{x,3}$ = 2.5 m, t_3 = 0.012 m
$L_{x,4}$ = 2 m, t_4 = 0.012 m
L_y = 0.80 m

Material
Steel
Young's Modulus:
E_0 = 210 GPa
Poisson's coefficient: ν = 0.3
Volumic mass:
ρ = 7600 kg/m^3

At each junction, the mounting assumed lossfree is such that free rotation is allowed while displacement is zero. The point force action on the structure produces flexural waves. Since the boundaries are assumed to be simply supported along the x-axis, the displacement W_p is separable according to

$$W_p(x, y) = \sum_n w_{p,n}(x) \cdot \varphi_{p,n}(y) \qquad (1)$$

where the x-dependence for the nth mode is given by

$$w_{p,n}(x) = A_{p,n}e^{-i\kappa_{2,n}x} + B_{p,n}e^{i\kappa_{2,n}x} + C_{p,n}e^{-\kappa_{1,n}x} + D_{p,n}e^{\kappa_{1,n}x} \qquad (2)$$

where $p = 1, 2, 3, 4$ and $\kappa_{1,n}, \kappa_{2,n} = (\kappa^2 \pm k_n^2)^{1/2}$ with the wave number κ and the eigenvalues $k_n = n\pi/L_y$ corresponding to the mode functions $\varphi_{p,n}(y) = \sin(k_n \cdot y)$. The time harmonic dependence $e^{i\omega t}$ is implicitly considered throughout this paper. To determine the coefficients $A_{p,n}$, $B_{p,n}$, $C_{p,n}$, $D_{p,n}$, boundary and continuity conditions at junctions $j = 1, 2, 3, 4$ and 5 are used: no displacement at each junction j, no bending moment at $j = 1$ and 5, angle and moment continuity at $j = 2, 3$ and 4. When all the coefficients of Eq. (2) are determined, the transfer matrix $[T_p]$, corresponding to plate p, is constructed from computation of angular displacements Θ_j and bending moments M_j calculated as derivatives of w_p.

$$\text{Plate } p\text{:} \quad \begin{pmatrix} \Theta_{j+1} \\ M_{j+1} \end{pmatrix} = \begin{bmatrix} c_{11} & c_{12} \\ c_{21} & c_{22} \end{bmatrix} \begin{pmatrix} \Theta_j \\ M_j \end{pmatrix} = [T_p] \begin{pmatrix} \Theta_j \\ M_j \end{pmatrix} \quad j = 1,\ldots,4 \qquad (3)$$

The various procedures and calculations mentioned above are detailed in [4].

Power Calculation by the Junction Method

The power dissipated in the plates is obtained from a calculation of the power at the plate junctions. Applying the boundary conditions and taking the temporal and plate-width average, the power at the plate junction is

$$P_j = \omega L_y \cdot \text{Real}\{M_j \cdot (i\Theta_j)^*\}/4 \qquad \text{where } j = 1,\ldots,5 \qquad (4)$$

The input power is determined by

$$P_{\text{input}} = \omega \cdot \text{Real}\{F \cdot (iW_0)^*\}/2 \qquad (5)$$

where W_0 is the displacement at the excitation point (x_0, y_0).
The power dissipated, ΔP_p, in the pth plate is

$$\Delta P_p = P_j - P_{j+1} \qquad \text{where } j{=}p \text{ for } p = 2, 3 \text{ and } 4 \tag{6}$$

Note the particular case for plate 1: $\Delta P_1 = P_{\text{input}} - P_2$ (7)

3. PARAMETER INFLUENCE ON THE VIBRATIONAL POWER

This section investigates the vibrational behaviour of the four plate structure excited, on plate 1, by a spring-mass system subjected to a harmonic force F_H. A mathematical description of the system is developed to compute and plot the time- and space-average of the squared plate-velocities, added onto the third-octave bands. The effects of some structural parameters are then examined by means of this program. The force exerted by the spring-mass system on plate 1 is given by

$$F = -Y_0 \cdot \omega_0^2 \cdot F_H \Big/ \left[(\omega_0^2 - \omega^2) \cdot Y_0 + k \cdot \omega^2 \right] \tag{8}$$

where Y_0 is the dynamic stiffness at the excitation point (x_0, y_0), k and m are the spring constant and the mass, respectively, with $\omega_0^2 = k/m$. Eq. (8) is used as an input force to execute the program calculating and plotting the averaged squared velocities in decibels, the speed of reference being $5 \cdot 10^{-8}$ m/s.

Point Force Location

When observing the curves, in Fig. 2, the higher velocity levels of plate 4 are obtained for center excitations of the source plate, i.e. at A(0.15, 0.40) center of plate 1 where the higher peaks reach an amplitude of 35 dB. Generally, the closer the excitation is set to the source plate edges, the more the velocity level in the receiving plate decreases. This phenomenon is amplified near the plate corners, where a decrease of nearly 20 dB is registered at 1000 Hz when comparing excitations at A(0.15, 0.40) and C(0.05, 0.05), respectively.

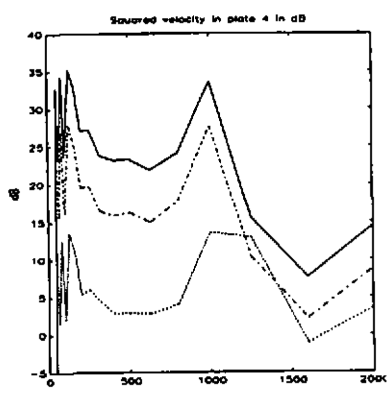

Figure 2. Averaged squared velocity of plate 4. Excitation positions in [m]: ——, A(0.15, 0.40); —·—·, B(0.05, 0.40); ——, C(0.05, 0.05)).

Damping

The influence of three different dampings applied to plates 1 and 2 is investigated by observing the effects on the averaged squared velocity of plate 4, in Figs. 3 and 4. Originally, the damping η_p equals 0.01 for the entire structure. Increased damping on plate 1 does not greatly decrease the averaged squared velocity of plate 4, except around the peak at 1000 Hz. This proves that damping introduced locally, even at the source plate, may have limited effect in reducing the vibration level of plates situated further away in the structure. Adding damping on plate 2 produces a large diminution of the averaged squared velocity of plate 4 especially when η_2

equals 0.1. Experiments on a scale model, in [5], tend to confirm these computer simulations.

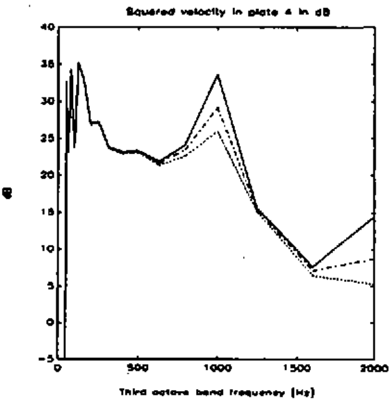

Figure 3. Averaged squared velocity of plate 4 for three different dampings in plate 1: ——, η_1=0.01; —·—·, η_1=0.05; ——, η_1=0.1.

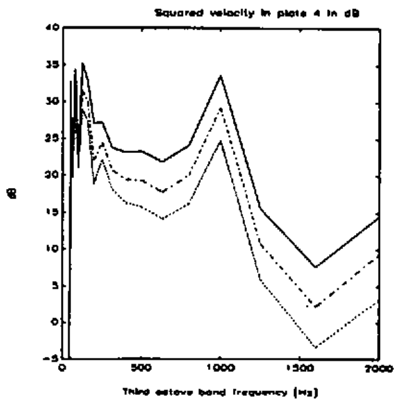

Figure 4. Averaged squared velocity of plate 4 for three different dampings in plate 2: ——, η_1=0.01; —·—·, η_1=0.05; ——, η_1=0.1.

4. CONCLUSION

The calculation procedure, describing the vibrational behaviour of built-up structures and developed according to the waveguide model, is easily adaptable to similar structures with various boundary conditions and excitations, for computation of vibrational power or averaged squared velocities in sub-structures, i.e. single plates. This finally aims at investigating and predicting the vibration propagation throughout entire structures. The method provides a computationally efficient tool for a systematic analysis of vibrational power in built-up plate systems, enabling convenient and rapid parametric studies.

References:

[1] A. C. Nilsson 1977 *J. of Sound and Vibration* 55(1), 71-91. Attenuation of structure-borne sound in superstructure on ships.

[2] B. M. Gibbs and Y. Shen 1987 *J. of Sound and Vibration* 112(3), 469-485. The predicted and measured bending vibration of an L-combination of rectangular thin plates.

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[4] R. Haettel 1994 Prediction of Vibrational Power in built-up plate structures, TRITA-FKT report 9427, MWL, KTH.

[5] A. C. Nilsson 1978 *J. of Sound and Vibration* 61(1), 45-60. Reduction of structure-borne sound in simple ship structures: results of model tests.