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A THEORETICAL AND EXPERIMENTAL INVESTIGATION OF THE BENDING VIBRATION OF CONNECTED PLATES.

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INTRODUCTION

In investigations of structure-borne sound propagation through structures such as highrise buildings and ships there remains a requirement for deterministic solutions which will yield vibration levels at any point on a structural element such as a plate or beam due to vibrational sources at other points on the element or on connected elements. The problem rapidly becomes intractable when excitation frequencies are high and modal densities great. Edge conditions are likely to be non-uniform and the effects of damping need to be included. The sources of vibration, such as plant machinery and ships engines, may impart translational and rotational components in unknown proportion and the spatial variation of forcing function may be complicated and unknown. An engineering solution has been achieved by reference to spatial and spectral averages of the input and response (1) and there are examples of the application of these techniques in the assessment of sound transmission in buildings (2) and ships (3). There are likely to be discrepancies between measurement and prediction however when modal densities are low and when structures display periodicity.

In recent work by the authors, an approximate solution is presented for the bending vibrations of a combination of rectangular plates (4). The method is similar to those used for calculating the bending vibrations of a single plate where the displacement amplitude function is expressed as a linear combination of co-ordinate functions. The displacement amplitude function vector of the global system of a combination of rectangular plates is also expressed as a linear combination of co-ordinate function vectors which satisfy all the boundary conditions of the global system. Thus for a series of L-combinations of rectangular plates the co-ordinate function vectors can be organised from the eigen functions of the series of L-combinations of beams of unit width perpendicular to the coupled edges and those of the single beams of unit width parallel to the coupled edges.

The solution is not general in that the junctions of the plates are constrained with respect to displacement and thus the generation of in-plane vibration on the plates is not allowed. Similarly the non-coupled edges are assumed pinned but with a variable complex rotational stiffness which allows consideration of edge constraints ranging continuously from the simply supported condition to the clamped condition, including the effect of damping. As a test of the theory it was decided to mathematically and physically model the case of an L-combination and a T-combination of rectangular plates subjected to a point excitation force for different non-coupled edge conditions.

THE COMPUTER MODEL

It was foreseen in this investigation that there would be difficulties in obtaining the 'correct' material constants needed for the computer model.

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The values used lie within the range of what is practically attainable or were obtained from static and dynamic tests. The model dimensions are as indicated in Figure 1. The model was to be of 2mm perspex sheet joined to form a right-angled junction. The point force position was as indicated and five receiver positions in total were considered; results are presented as level differences both on the same plate (eg. $1_4 - 1_1$) and across the junction (eg. $1_5 - 1_4$). Results are presented at a frequency interval of 20 Hz for a frequency range 0 - 2 kHz. This represents, in the computer model, the organisation of 20 eigen modes for the L-combinations of beams perpendicular to the coupled edges; and 7 eigen modes for the single beams parallel to the junctions, therefore 140 co-ordinate function vectors are used to calculate the forced vibrations.

In assigning material constants a value of plate bending stiffness of 4.8 Nm was obtained from static deflection tests. The modelling of plate and edge damping is more problematical and various combinations of internal and edge dissipative losses were investigated. It was found that best fits of predicted to measured response were obtained for an assigned internal loss factor of 3×10^{-2} and a boundary loss factor of 5×10^{-2}. Two non-coupled edge conditions were to be investigated one of which approximated the simply supported condition and the other the clamped condition. Ideally the first should be described in terms of zero rotational stiffness and the latter by an arbitrarily large value of 10^{12} Nm, say. These extremes are not possible to model physically and the values of an experimental model is likely to lie within this range. In Figure 2 is shown the predicted upward frequency shift of level difference $(1_3 - 1_1)$ when edge rotational stiffness is increased from 50N to 1000N; a range likely to be attainable in practice.

EXPERIMENTAL MODEL

The experimental validation presented many difficulties. What would appear to be a straight-forward process of simulation of edge constraint and input force can easily become complicated in a model system which is easily perturbed. The perspex plates were clamped into a demountable frame of bolted 24mm square section mild steel. The whole was mounted on resilient pads onto a support frame in a anechoic chamber. The choice of plate and frame material was such to insure a large vibrational level difference between plate and frame; in the main the difference did not fall below 20 dB.

The simulation of the clamped non-coupled edge condition would appear to be a simple matter but this proved not to be the case. The clamping, if too tight, caused plastic deformation of the perspex; if not tight then some small displacement was possible and the edge condition is not correctly simulated. Measurements were for the highest clamping pressure attainable (dictated by the tolerances of the bolts and the deformation of the perspex plate) and results were repeatable.

Various attempts were made to simulate the simply-supported edge condition, including the use of thin sheet metal shims. The condition which proved easiest to model was that of a square notched edge. The notch at the boundary was of depth 1.5mm and width 1.5mm. Static deflection measurements were carried out on a beam with such a notch and the equivalent rotational stiffness was calculated to be 50N. This value was ultimately incorporated in the computer model.

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In driving a small model and sensing the vibrational response care must be taken not to introduce excessive mass, stiffness or damping to the system. This proved not to be a problem with the small accelerometer transducers employed which contributed less than 1% to the mass of the plate (per accelerometer). The problem of driving the system correctly and repeatably proved more difficult. Electro dynamic shakers, speech coils with distanced permanent magnets and magnetic transducer pairs in push-pull mode were employed. All offered advantages and disadvantages but it was found that the magnetic transducers offered repeatable and consistant results and were employed in the main.

Both slow swept sine and impulsive signal sources were tried; the latter offered the advantage of allowing rapid and repeatable readings and was employed throughout this investigation. The input pulse was rectangular in shape and of duration 0.1 ms and the source spectrum could reasonably be assumed flat up to a frequency of 5 kHz. Results were obtained by means of a dual channel Fast Fourier analyser and the level difference from a matched pair of accelerometers was obtained direct as a transfer function with a resolution of 5 Hz. Phase information is not presented in this paper. The input was repeatable and signal noise ratio was increased by averaging; typically of 1024 pulses.

RESULTS AND DISCUSSIONS

Results are presented first for the notched plate case. In Figures 3 and 4 results are presented for $(1_5 - 1_4)$ and $(1_5 - 1_3)$, respectively, where the accelerometer positions are removed from the plate edges. Good agreement was obtained for a bending stiffness $D^* = 4.8\ (1 + i\ 3 \times 10^{-2})$ NM and rotational stiffness $\zeta^* = 50\ (1 + i\ 5 \times 10^{-2})$ N. It can be seen that the curves compare well both in terms of the dynamic range and response signature. Results for positions near to the plate edges, such as those for position 2, were less good and were thought to result from difficulties in correctly simulating the required edge conditions.

In Figures 5 to 6 results are presented for the clamped plates case for $(1_5 - 1_4)$ and $(1_4 - 1_3)$ respectively. The best fit was obtained for an edge rotational stiffness of 200 $(1 + j\ 5 \times 10^{-2})$ N. This stiffness, which is only four times that of the notched case, is surprisingly low and results from the conflict of practical requirements of the clamping frame where edge displacement was to be avoided but the perspex was not to be deformed. Again there is agreement between measurement and prediction but, in general, less so than for the notched case.

REFERENCES

(1) R.H. Lyon, 1973, Statistical Energy Analysis of Dynamical Systems: Theory and Applications. MIT Press.

(2) R.J.M. Craik, 1982, Journal of Sound and Vibration 82, 505-516. The prediction of sound transmission through buildings using Statistical Energy Analysis.

(3) A.C. Nilsson, 1984, Journal of Sound and Vibration 94 (3), 411-429, A method for the prediction of noise and velocity levels in ship constructions.

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(4) Y. Shen and B.M. Gibbs, 1986, Accepted for publication in The Journal of Sound and Vibration 104 No. 3. An approximate solution to the vending vibrations of a combination of rectangular thin plates.

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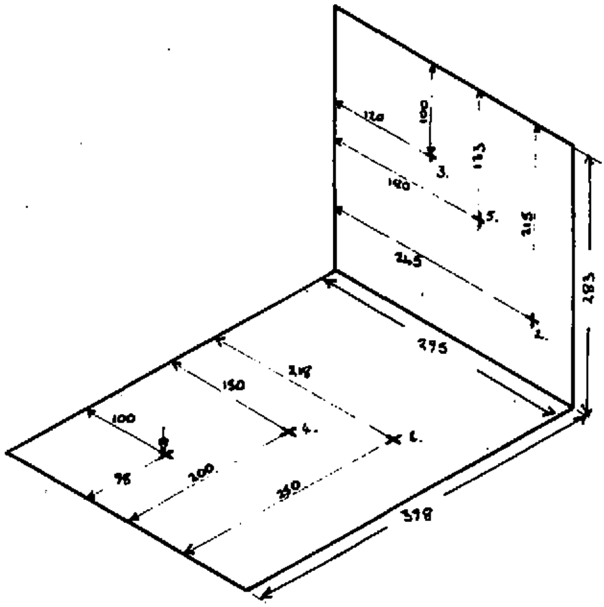

Figure 1. Dimensions and transducer positions for the model L-combination of plates. Dimensions in m.m.

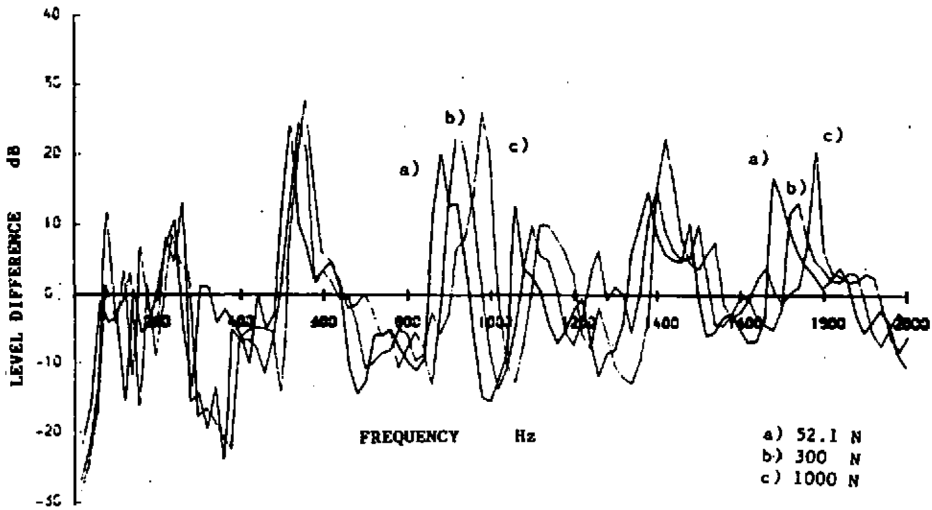

Figure 2. Predicted level difference for edge rotational stiffness

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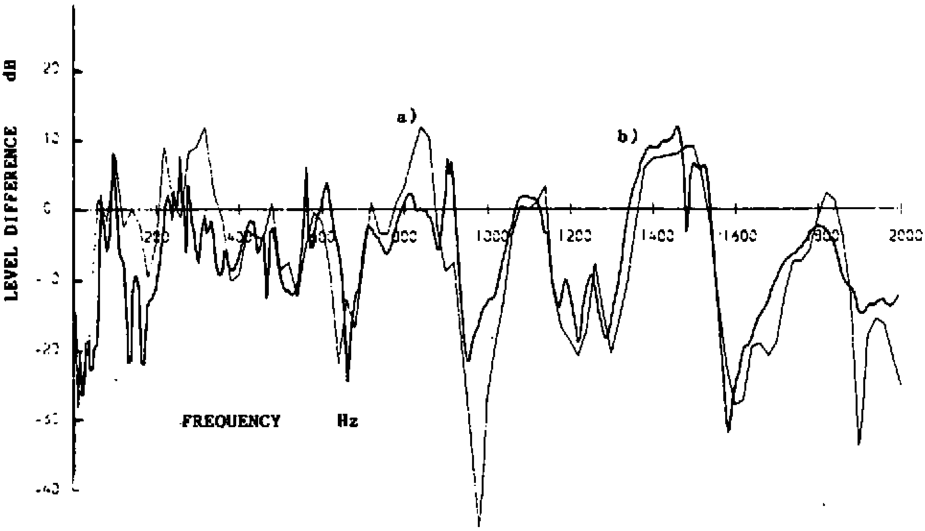

Figure 3. Measured and predicted level difference ($l_5 - l_4$) for the notched plate case.

a) theory
b)measurement.

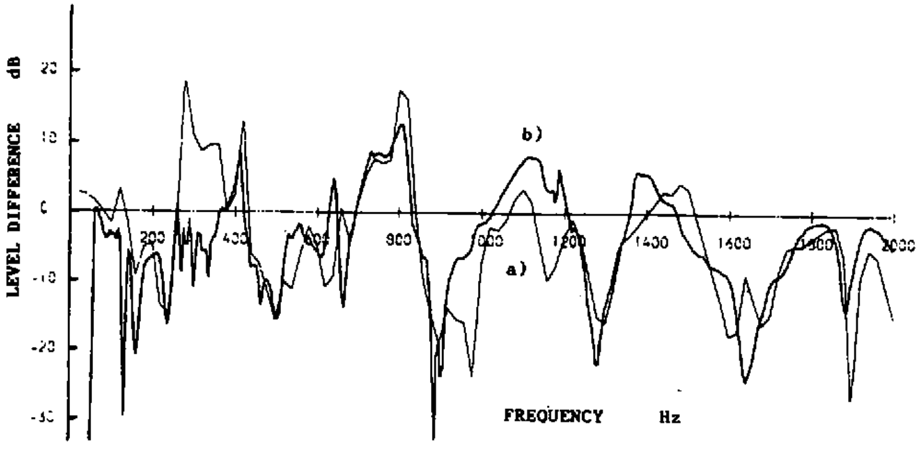

Figure 4. Level difference($l_5 - l_3$)

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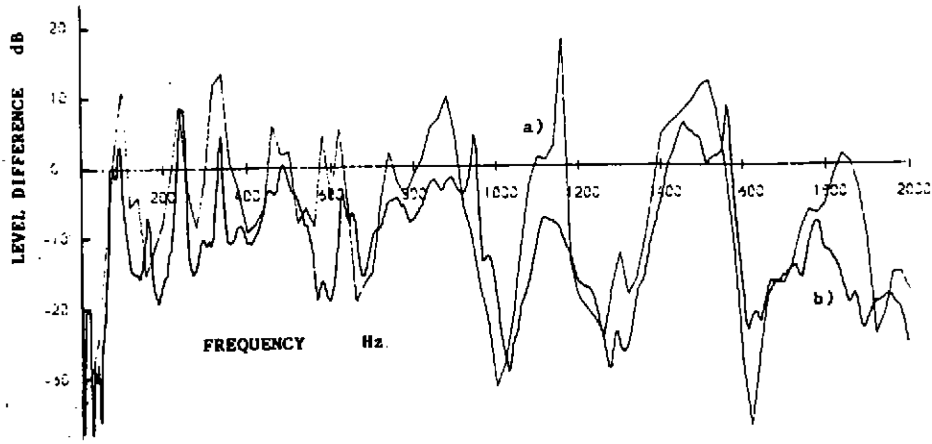

Figure 5. Measured and predicted level difference ($l_5 - l_4$) for the clamped plate case.

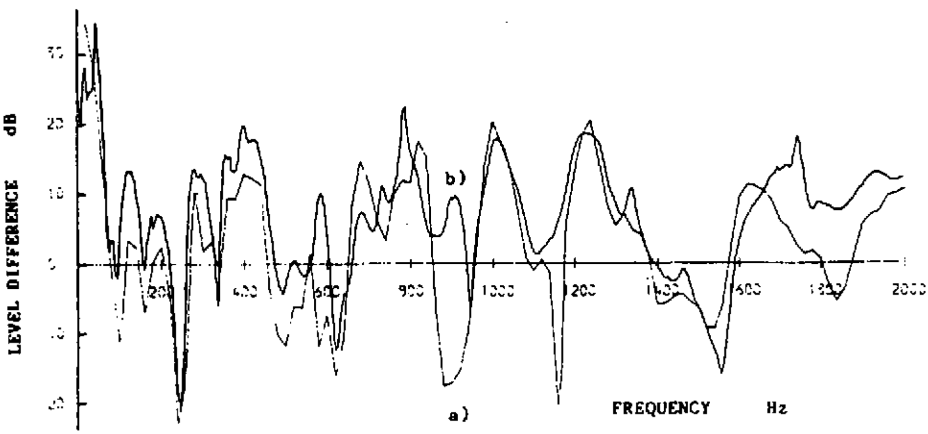

Figure 6. Level difference ($l_4 - l_3$)