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MODELLING THE STRUCTURAL/ACOUSTIC COUPLING OF AN AIRCRAFT FUSELAGE FOR OPTIMISING AN ACTIVE NOISE CONTROL SYSTEM

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1. INTRODUCTION

Finite Element Analysis has been used extensively in the past for modelling the system response of coupled vibro-acoustic systems, and recently has been used for predicting the performance of active noise control systems in cavities enclosed by contiguous structures [1]. The analysis of such fully-coupled models requires considerable computational resources; however, CPU time may be significantly reduced if the systems are modelled separately and then combined using modal coupling theory [2]. The success of this approach requires accurate numerical models which must be validated experimentally and which often involve significant investment of time and effort.

Measured transfer functions may also be used to estimate control system performance [3]. Although this technique provides accurate system data, it requires that each point of interest be excited by a source (even if reciprocity is assumed) in order to identify each transfer function which again can involve significant effort. In this paper, it is shown that use of modal analysis of measured data in conjunction with modal superposition and modal coupling provides an alternative analysis approach which is more efficient than using measured FRF and is potentially as accurate as FEA.

In the remainder of the paper, the latter approach is described and results obtained using this technique are compared with experimental results and those obtained using the FEA approach.

2. THEORETICAL BACKGROUND

The use of modal analysis provides a more efficient technique for obtaining the FRF matrix than measuring the data directly. This is because direct measurement requires excitation of each point of interest to obtain one triangular component of the FRF matrix, whereas modal analysis permits calculation of the complete FRF matrix from only a single excitation point, thereby greatly reducing the experimental effort. Several excitation points may be used for greater accuracy.

The modal analysis technique involves decomposing the measured column(s) of FRF data (for both the cavity and structure) into equivalent modal parameters as described in Equn 1.

$$\mathbf{F}(j\omega) = \mathbf{X}_m + \sum_{n=n_1}^{n_2} \left[\frac{-\omega^2 \mathbf{X}_n}{\omega_n^2 - \omega^2 + j\eta_n \omega_n^2} \right] + -\omega^2 \mathbf{X}_s \qquad (1)$$

where $\mathbf{F}(j\omega)$ is the row vector of measured FRF data at frequency ω, ω_n and η_n are the natural frequency and modal loss factor respectively for the nth mode, $\mathbf{X}_m$ and $\mathbf{X}_s$ are the mass and stiffness residues respectively and account for the modes with natural frequencies outside the measurement interval. $\mathbf{X}_n(\mathbf{x}|\mathbf{y}) = \Phi_n(\mathbf{x})\Phi_n(\mathbf{y})$, where $\Phi_n(\mathbf{x})$ is the mode shape function of the nth mode. These derived modal parameters can then be used in the same manner that the FEA modal parameters are used in estimating the system response.

The modal approach lies somewhere between FEA and direct FRF measurement and incorporates the advantages of both techniques. It should be noted that the system must have a low modal density within the measurement interval to permit an accurate modal decomposition.

3. FINITE ELEMENT ANALYSIS

The coupled vibro-acoustic system investigated comprised a longitudinally stiffened cylinder with an integral floor having shear-diaphragm boundary conditions with rigid end caps, and is illustrated in fig 1.

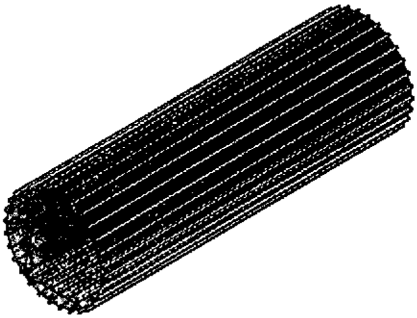

Fig.1 Longitudinally Stiffened Cylinder

In this study, the system was modelled numerically using the commercially available FEA package ANSYS. Although ANSYS is capable of performing a fully coupled modal analysis, the cavity and the structure were modelled separately due to the computational difficulties associated with a fully-coupled structural-acoustic analysis. The two FE models were coupled using the modal-coupling theory outlined in reference 2. The FE models consisted of 11242 acoustic tetrahedral elements (2340 nodes), 2280 triangular shell elements (800 nodes). 50 acoustic modes and 200 structural modes were extracted.

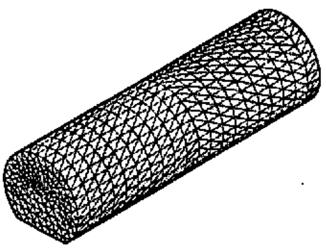

Fig. 2 FEM of the Cavity

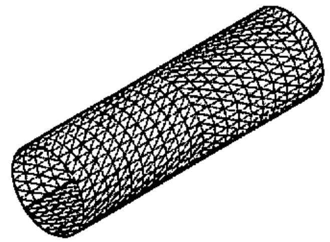

Fig. 3 FEM of the Shell

4. MODAL ANALYSIS

Modal analysis of both the cavity and the structure was conducted to: (1) obtain a measured FRF matrix of all points of interest within the system and (2) measure modal parameters to verify the FEA parameters used in the numerical simulation. Significant reductions in the experimental effort are achieved using this approach as opposed to directly measuring the entire FRF matrix. In addition, modal analysis enables very rapid system performance prediction by assuming that each control source is able to remove the modal contribution of a single (dominant) mode, whether it be an acoustic, structural or transformed mode [4]. A multiple-input multiple-output least squares curve fitting routine (modified from [5]) was used to extract the modal parameters from the measured data. The low modal density of both the acoustic space and the structure permitted accurate estimates of the modal parameters. A synthesised transfer function is compared against a typical measured FRF in fig 4 and the calculated mode shape (black represents maximum pressure, white represents minimum) for the first acoustic mode is presented in fig 5.

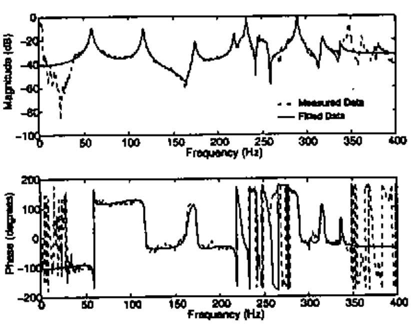

Fig. 4 Synthesised Vs Measured FRF

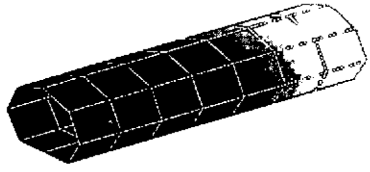

Fig. 5 [0 0 1] Cavity Mode Shape

Residues were used in the curve fitting routine to account for modes outside the frequency range of interest. Inclusion of the residue terms is essential when determining the reduction in cost function for it is these remainder terms that often limit the amount of control that can be achieved. It should be noted that the acoustic mass residues provided the only means of estimating modal parameters for the bulk compression mode of the acoustic space as the instrumentation was incapable of measuring the system response below approximately 40 Hz.

5. RESULTS

A simple experimental configuration was used to compare the numerically and experimentally predicted controlled levels against those measured experimentally. This comprised a single internal primary and control monopole source with four acoustic error sensors.

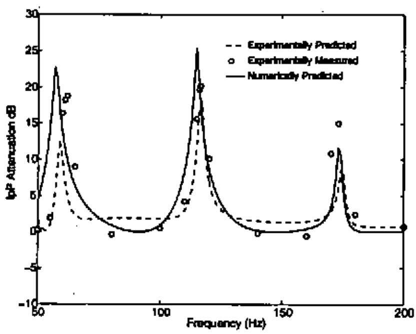

Fig. 6 Comparison of Performance Prediction Techniques

Fig 6 compares the numerically predicted and modally derived reduction in space-averaged squared pressure levels inside the cylinder against experimentally measured levels. As can be seen, both the numerical and experimental prediction techniques provide acceptably accurate performance estimates.

6. CONCLUSIONS

Predicting the performance of an active noise control system via a modal model derived from measured transfer function data offers distinct advantages over conventional analysis techniques such as FEA and direct response measurements. The modal analysis technique simply makes use of the measured data that were necessary for undertaking and validating the FEA, thus avoiding the need for time consuming numerical modelling. The advantage of the modal analysis approach over the transfer function approach is the much smaller effort required to obtain the FRF matrix, at the expense of only a small loss in accuracy.

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