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AN EFFICIENT METHOD FOR THE PREDICTION OF AIRCRAFT INTERIOR NOISE LEVELS

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1. INTRODUCTION

In recent years, much research effort has been directed towards the prediction of aircraft interior noise levels. Perhaps the most direct approach to the problem involves the use of the finite element method to model both the structure and the interior acoustic space [1]. This method can however face severe computational difficulties: for example, Roozen [2] has shown that a structural finite element model having 550,000 degrees of freedom was required to compute the modes of vibration below 225Hz for a 2m length of the fuselage of the Fokker F50. This computation took 58hrs of CPU time on an IBM 3090, or 2.8 hrs on a CRAY Y-MP/8. Clearly, even with present day computer capabilities, a finite element analysis of the complete aircraft fuselage would be of marginal feasibility.

In view of the difficulties faced by standard analysis procedures, a number of alternative techniques have been developed for the prediction of interior noise levels. These methods sacrifice detailed modelling in attempt to obtain an efficient solution whose accuracy is sufficient for design purposes. The PAIN analysis which is described in references [3,4] is one such approach: the aircraft fuselage is modelled as a smeared cylindrical shell with an enclosed floor. The structural modes of vibration are calculated by using the Rayleigh-Ritz analysis of Peterson and Boyd [5], while the finite difference technique is used to calculate the interior acoustic modes. A modal response analysis is then used to predict the interior noise level which arises from external acoustic excitation. The results of this method have been compared with both model tests and full scale flight data, and a good level of agreement has generally been found.

The present approach to interior noise prediction follows the same general philosophy as the PAIN technique, although advantage is taken here of the dynamic stiffness structural analysis technique which is presented in reference [6]. With this approach, fuselage structures of complex cross-sectional shape may be modelled with relative ease, and the dynamic response of the fuselage may be computed directly without a prior analysis of the modes of free vibration. The interior noise levels are then computed by using the acoustic boundary element method: this approach requires relatively few degrees of freedom, as the boundary element mesh is one-dimensional. Again, a direct solution procedure is adopted, so that the need to compute the interior acoustic modes is obviated. As in the PAIN technique [3,4] the presence of interior trim is allowed for by using a point transfer matrix approach. The following sections describe in turn the structural and acoustic modelling techniques which are adopted, and a number of example results are then presented.

2. DYNAMIC STIFFNESS ANALYSIS OF FUSELAGE VIBRATIONS

The dynamic stiffness approach of reference [6] is applicable to structures which can be represented as an assembly of cylindrical shell elements of the type shown in Figure 1. The curved sides of the element are taken to be simply supported (by, for example, a heavy frame or a bulkhead), which allows the complex amplitude of the displacement components $u=(u\ v\ w)^T$ to be written in the form

$$u(x,y)=\sum_n (u_n(y)\cos k_n x \quad v_n(y)\sin k_n x \quad w_n(y)\sin k_n x)^T, \tag{1}$$

where $k_n=n\pi/b$ and b is the length of the element. The formulation of reference [6] allows for the presence of distributed pressure loading of the form

$$F(x,y)=\sum_m e^{i\mu_m y} X_m(x), \tag{2}$$

which could represent the Fourier decomposition of any arbitrary pressure distribution. With the dynamic stiffness technique, the equations of motion which govern the shell element are solved exactly to yield a relationship between the displacements and tractions which are associated with the straight edges. This relationship may be written in the form

$$K_n d_n = f_n. \tag{3}$$

Here d_n is an 8x1 vector which contains the shell displacements (u_n v_n w_n θ_n) on the two straight edges, and K_n is the associated dynamic stiffness matrix. The loading vector f_n may be expressed in terms of the particular integral which is associated with the distributed loading. If the aircraft fuselage is modelled as an assembly of shell elements, then standard matrix assembly procedures may be used to derive the dynamic stiffness matrix and loading vector for the whole fuselage. The response of the fuselage to pressure loading may then be computed. It may be noted that a separate analysis is performed for each response component (n), and the complete response is then recovered from equation (1). The presence of stringers and frames may be allowed for by using the smearing technique [6], while line masses and heavy longerons may be included as discrete elements at the assembly stage. An allowance for trim mass and damping may also be incorporated into equation (3), as explained in reference [7].

With the present approach, an aircraft fuselage with an enclosed floor would be modelled by three elements and two nodes, located at the edges of the floor. For a specified value of n this model would consist of eight degrees of freedom, and thus the solution of the governing equations requires very little computer time. The number of terms to be included in equation (1) will depend to a large extent on the frequency range and the spatial distribution of the external pressure loading.

3. BOUNDARY ELEMENT ANALYSIS OF INTERIOR ACOUSTICS

A schematic of the interior airspace is shown in Figure 2: the x coordinate coincides with the longitudinal coordinate which is used in the dynamic stiffness analysis of the previous section, while the coordinates z_1 and z_2 range over the cross-section of the airspace. If the boundary conditions on the end-walls of the airspace involve only the pressure p and its derivative $\partial p/\partial x$, then the pressure may be expanded in the form

$$p(z_1,z_2,x)=\sum_n p_n(z_1,z_2)\psi_n(x), \qquad (4)$$

where the function $\psi_n(x)$ will depend on the end-wall boundary conditions. If "hard" end walls are considered then $\psi_n(x)=\cos\gamma_n x$, where $\gamma_n=n\pi/L$ and L is the length of the acoustic space. If this assumption is made then some allowance for end-wall losses can be made by increasing the side-wall absorption coefficient by an amount $\alpha_e A_e/A_s$, where α_e is the end-wall absorption coefficient and A_e and A_s are respectively the end- and side-wall areas. The two-dimensional pressure component p_n which appears in equation (4) must satisfy a Helmholtz equation of the form

$$\nabla^2 p_n+[(\omega/a_i)^2-\gamma_n^2]p_n=0, \qquad (5)$$

where a_i is the speed of sound. Interior losses may be modelled by using a complex value of a_i, although in the present approach such losses are accounted for by an increase in the side-wall absorption coefficient. The boundary conditions which must be satisfied by p_n on the perimeter of the airspace cross-section may be written in the form

$$-u_n=T_1 p_n+T_2 q_n, \qquad (6)$$

$$q_n=\delta_n\sum_m w_m\int_{L_1}^{L_2}\psi_n(x)\sin k_m x\,dx, \quad \delta_n^{-1}=\int_{L_1}^{L_2}\psi_n^2(x)dx. \qquad (7,8)$$

Here u_n is the normal component of the acoustic displacement ($\rho_i\omega^2 u_n=\partial p_n/\partial n$), L_1 and L_2 represent the span of the interior airspace, and T_1 and T_2 are coefficients which are related to the properties of the interior trim [7]. The coefficient T_1 accounts mainly for trim absorption: this may be written approximately as $T_1=i\alpha/4\omega\rho_i a_i$, where α is the trim absorption coefficient. The coefficient T_2 is a form of transmission coefficient, which relates the displacement of the trim interior surface to the fuselage structural displacement w.

In what follows the solution to equation (5), subject to the boundary conditions equation (6), is obtained by using the boundary element method. With this approach the pressure at any point within the cross-section is written in the form

$$p_n(z)=\int_c G_n(z,z')\sigma_n(z')dl(z'), \quad (9)$$

where the integral is around the perimeter of the cross-section and

$$G_n(z,z')=(i/4)H_o^{(2)}(\alpha_n r), \quad r=|z-z'|, \quad \alpha_n^2=(\omega/a_i)^2-\gamma_n^2. \quad (10\text{-}12)$$

The term σ_n which appears in equation (9) represents an initially unknown source distribution which is found by the application of the boundary conditions, equation (6). To this end the perimeter is divided into a number of segments (j) of length l_j over which the source strength is approximated to the constant value a_{nj}. Equation (9), together with equation (6), then leads to the following set of linear equations for the source strengths a_{nj}

$$(A_n+\rho_i\omega^2 T_1 B_n)a_n=-b_n, \quad (13)$$

$$(A_n)_{mj}=-\frac{1}{2}\delta_{mj}l_j+\int_{R_m}\int_{R_j}[\partial G_n(z,z')/\partial n_z]dl_j(z')dl_m(z), \quad (14)$$

$$(B_n)_{mj}=\int_{R_m}\int_{R_j}G_n(z,z')dl_j(z')dl_m(z), \quad (15)$$

$$(b_n)_m=\rho_i\omega^2 T_2\int_{R_m}q_n(z)dl_m(z). \quad (16)$$

Here the integral over R_m, for example, represents integration over the mth boundary segment. It can be noted that the boundary element method normally adopts a collocation procedure in which the boundary conditions are imposed at a number of specified points. With this approach the double integrals which appear in equations (14) and (15) are replaced by a single integral over R_j. The present formulation, which is based on a Galerkin approach to the application of the boundary conditions, has been found to require significantly fewer elements. The integrals which appear in equations (14-15) are evaluated analytically for m=j and by Gaussian integration otherwise [7].

One aspect of the present direct (i.e. non-modal) solution of the acoustic equations is that the result yielded by equation (13) can be sensitive to artificial damping effects. Such effects can arise from small errors in the computation of A_n and B_n which can alter the phase of the matrices and thus introduce numerical damping. These effects are discussed in reference [7] in the context of a simple box test case, and a relatively simple method is presented for their suppression.

Once the coefficients a_n have been calculated from equation (13), the pressure component p_n may be recovered from equation (9). Repeating this procedure for a range of values of n allows the complete interior pressure field to be determined from equation (4).

4. EXAMPLE APPLICATION

As an example the foregoing analysis has been applied to the test article which is described in reference [8]. The structure consists of an aluminium cylinder of radius 0.254m, skin thickness 1.6mm, and length 1.19m. The cylinder is fitted with end caps which approximate to simple supports and which are taken to be acoustically hard; the arrangement is such that the length of the internal airspace is 1.22m, which is slightly greater than the effective structural length. The computed noise reduction for an incident acoustic plane wave at 15° to the cylinder axis is shown in Figure 3(a). This value is defined by $-10\mathrm{Log}(\langle p^2\rangle/\langle p_o^2\rangle)$, where $\langle p^2\rangle$ is the space averaged mean squared internal pressure and $\langle p_o^2\rangle$ is the mean squared external pressure. The bars which are shown in Figure 3(a) represent the peaks and troughs of the results yielded by the PAIN analysis procedure [8]. Clearly the agreement between the two methods is good; most of the differences can be explained by the fact that constant loss factors (0.003 for both cylinder and airspace) have been used in the present analysis, while frequency dependent values were used in reference [8]. The troughs shown in Figure 3(a) correspond in the main to acoustic resonances.

The effect of the presence of an internal floor on the interior noise is shown in Figure 3(b). The floor considered has the same thickness as the cylinder and subtends an angle of 100° at the cylinder centre. The noise reduction is similar to that shown in Figure 3(a) although the effect of structural resonances is more apparent. The reason for this is that the mode shapes of the cylinder with fitted floor are more irregular than the simple cos(my/r) shapes which occur for the cylinder alone. This irregularity tends to lead to greater generalized forces, and thus a greater resonant response, for the case of plane wave excitation. At around 279Hz the proximity of a structural and an acoustic resonant frequency leads to a very low noise reduction.

The effect of interior trim on the noise reduction is shown in Figure 3(c): the solid line represents the untreated cylinder, the dashed line corresponds to a fibreglass layer with vinyl facing, and the dotted line corresponds to a fibreglass layer with a lead-vinyl sound barrier. It can be seen that the trim both shifts the acoustic resonances and leads to a higher level of damping. For the lead-vinyl coating the transmission coefficient T_2 becomes significantly less than unity at the higher frequencies, leading to a further increase in the noise reduction levels.

Finally, Figure 3(d) shows the noise reduction for the bare cylinder in a diffuse sound field. The diffuse sound field was simulated by 288 plane wave components with differing directions and random phase. These results agree favourably with published values, as detailed in reference [7].

The present analysis was implemented on a 386 33MHz personal computer, and for the example considered around 45 CPU seconds were required to calculate the noise reduction at a single frequency. Clearly the required computer time is relatively low when compared to standard approaches such as the finite element method.

5. CONCLUSIONS

An approximate method has been presented for the prediction of interior noise levels. The basic assumption of the method is that the structure may be adequately modelled as an assembly of smeared dynamic stiffness shell elements. This is likely to be a reasonable approach for the lower frequency range where high frequency analysis techniques such as Statistical Energy Analysis (SEA) tend to be unreliable. The present approach may be used to investigate trends and the likely effect of design changes at the early stages of design. It may also be used to assess theoretically the likely effectiveness of various active noise control strategies.

It can be noted that the dynamic stiffness analysis of reference [6] is also applicable to higher frequency *local* vibrations. For example, the method may be used to assess the acoustic fatigue of a panel row which is assumed to be simply supported between two neighbouring fuselage frames. In this case the stringers may modelled discretely rather then through smearing, and the method is therefore exact within the validity of the initial assumptions. The fact that the applied loading is dealt with through the use of particular integrals means that the approach is ideally suited to pressure wave excitation and thus, through the use of the wavenumber-frequency spectrum $S(k,\omega)$, distributed random loading.

6. REFERENCES

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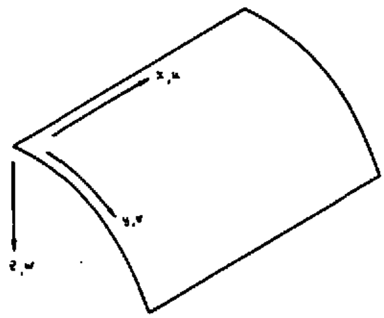

Figure 1: Dynamic Stiffness Shell Element

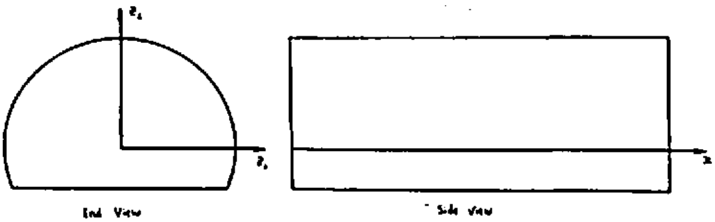

Figure 2: Interior Acoustic Volume

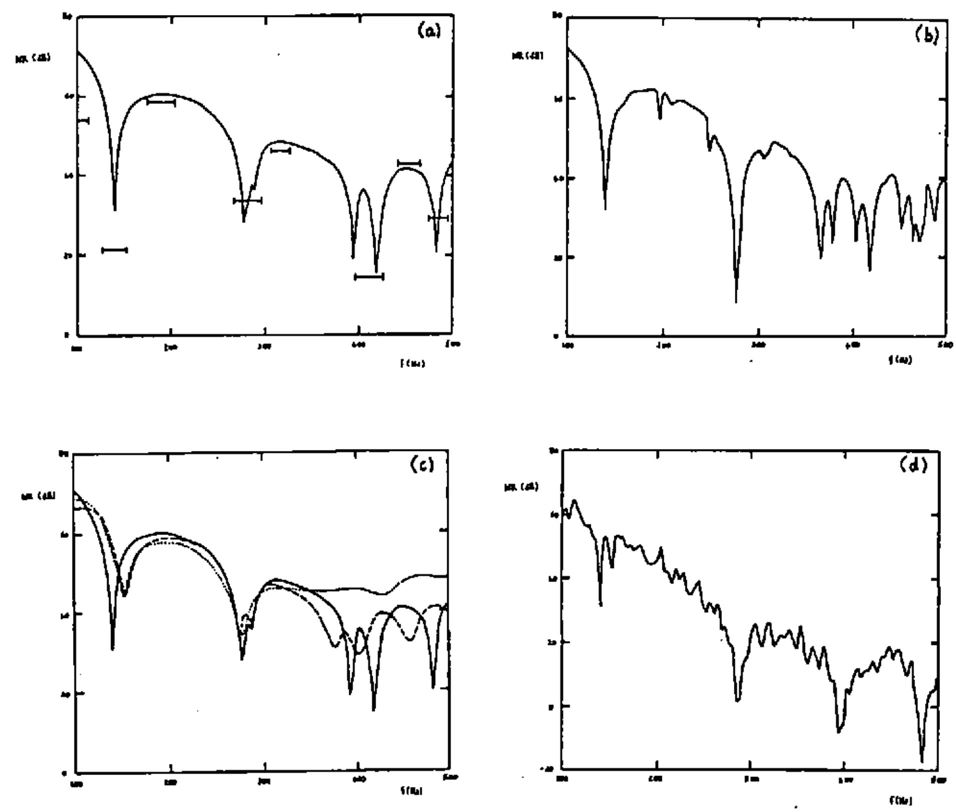

Figure 3: Noise Reduction for the Cylindrical Test Article