AN ACCURATE COUPLED STRUCTURAL-ACOUSTIC ANALYTICAL FRAMEWORK FOR THE PREDICTION OF RANDOM AND FLOW-INDUCED NOISE IN TRANSPORT VEHICLES: ITS VALIDATION

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1. INTRODUCTION

Aircraft cabin interior noise and vibration is mostly generated by the external flow excitation and engine noise. In opposition of what happens during takeoff, where the engine noise is the dominant cabin noise source, during cruise flight the airflow sources are the major contribution for the interior noise [1]. Additionally, turbulent boundary layer pressure (TBL) levels on the exterior of the fuselage increases with the flight speed. Similarly, automotive industry is progressively more concerned with passengers comfort. Since major advances have been made to the reduction of sound transmitted to the interior from the engine, transmission, and tires, the reduction of flow-induced noise is becoming more important.

Reduced cabin noise levels are desirable for both comfort and health-related reasons, and they are balanced with cost, complexity, and physical constraints of noise control systems. However, the successful implementation of noise control techniques is a challenging problem, and is far from being a straightforward task. To efficiently design a noise control system, a clear understanding of the mechanisms of sound radiation and transmission of the structural-acoustic system is crucial. Furthermore, when considering the TBL excitation, the noise reduction problem turns into even more complicated, since the TBL-induced pressure has a random and broadband nature. Mathematically, this problem can be simulated by the interaction of three models: (1) an aerodynamic model, which represents the TBL pressure on the cabin structure; (2) a structural model, defining the vibration of the cabin structure; and (3) an acoustic model representing the cabin interior sound pressure level.

The mains goal of the current research is to develop an analytical framework for the prediction TBL-induced noise into transport vehicles cabins, and its validation. The knowledge of the characteristics of the TBL excitation, its induced vibration on the structure, and the noise radiated into the cabin is essential for the accurate prediction of the interior noise. The effect of the receiving room space is an important factor for the accurate interior noise prediction.

1.1 Turbulent Boundary Layer Wall Pressure Model

The TBL wall pressure field over a flat plate is usually statistically described in terms of the pressure power spectral density (PSD). Corcos [2], proposed a model which considers the cross PSD of the stationary and homogeneous TBL wall pressure field, over a flat panel, for zero pressure gradient, in a separate form in the streamwise, x-, and spanwise, y-direction, as

$$S(\xi_x,\xi_y,\omega) = S_{ref}(\omega)\, e^{-\frac{\alpha_x \omega |\xi_x|}{U_c}}\, e^{-\frac{\alpha_y \omega |\xi_y|}{U_c}}\, e^{-\frac{i \omega \xi_x}{U_c}}, \qquad (1)$$

where $S_{ref}(\omega)$ is the reference PSD; ξ_x=x-x' and ξ_y=y-y' are the spatial separations in x and y directions, respectively; α_x and α_y are empirical parameters, chosen to yield the best agreement with the reality. Recommended values for aircraft boundary layers are α_x= 0.1 and α_y= 0.77.

1.2 Structural Model

Generally, an aircraft fuselage is a conventional skin-stringer-frame structure, with several panels connected between adjacent stringers and frames. The panel was considered to be flat and simply supported in all four boundaries. The plate governing equation, for a given applied external pressure, $p_{ext}(x,y,t)$, is defined by

$$D_p \nabla^4 w + \rho_p\, h_p\, \ddot{w} + \zeta_p\, \dot{w} = p_{ext}(x,y,t)\,, \qquad (2)$$

in which w is the plate displacement, ρ_p is the density of the panel, h_p is its thickness, D_p is the panel stiffness constant, and the term ζ_p was added to account for the panel damping.

1.3 Acoustic Model

The acoustical physical system consists of a three-dimensional rectangular enclosure, with five fixed walls, and one totally or partially flexible wall. The governing equation of this subsystem is the wave equation, defined by

$$\nabla^2 p - \frac{1}{c_0^2}\, \ddot{p} - \zeta_{ac}\, \dot{p} = 0\,, \qquad (3)$$

where p is the interior pressure, c_0 is the speed of sound inside the enclosure, and the term ζ_{ac} was added to account for the acoustic damping in the enclosure.

2. METHOD FOR SOLUTION

The governing equations for the coupled structural acoustic system are obtained from the combination of the previously described equations for the individual uncoupled systems. To perform this combination, some mathematical manipulations are needed (please refer to [3] for details). The final coupled system governing equation becomes

$$\begin{bmatrix} \mathbf{M}_{pp} & \mathbf{0} \\ \mathbf{M}_{cp} & \mathbf{M}_{cc} \end{bmatrix} \begin{Bmatrix} \ddot{\mathbf{q}}(t) \\ \ddot{\mathbf{r}}(t) \end{Bmatrix} + \begin{bmatrix} \mathbf{D}_{pp} & \mathbf{0} \\ \mathbf{0} & \mathbf{D}_{cc} \end{bmatrix} \begin{Bmatrix} \dot{\mathbf{q}}(t) \\ \dot{\mathbf{r}}(t) \end{Bmatrix} + \begin{bmatrix} \mathbf{K}_{pp} & \mathbf{K}_{pc} \\ \mathbf{0} & \mathbf{K}_{cc} \end{bmatrix} \begin{Bmatrix} \mathbf{q}(t) \\ \mathbf{r}(t) \end{Bmatrix} = \begin{Bmatrix} \mathbf{p}_{tbl}(t) \\ \mathbf{0} \end{Bmatrix}, \quad (4)$$

in which $\mathbf{M}_{pp}$, $\mathbf{D}_{pp}$, and $\mathbf{K}_{pp} \in \Re^{M\times M}$; $\mathbf{M}_{cc}$, $\mathbf{D}_{cc}$ and $\mathbf{K}_{cc} \in \Re^{N\times N}$; $\mathbf{M}_{cp} \in \Re^{N\times M}$; $\mathbf{K}_{pc} \in \Re^{M\times N}$; $\mathbf{q}(t)$ and $\mathbf{p}_{tbl}(t) \in \Re^{M\times 1}$; $\mathbf{r}(t) \in \Re^{N\times 1}$; $M=M_x\times M_y$ is the number of plate modes; $N=N_x\times N_y\times N_z$ is the number of acoustic modes; and w(x,y,t) and p(x,y,z,t) were defined, respectively, as:

$$w(x,y,t) = \sum_{m_x=1}^{M_x} \sum_{m_y=1}^{M_y} \alpha_{m_x}(x)\, \beta_{n_y}(y)\, q_{m_x m_y}(t)\,, \quad (5a)$$

$$p(x,y,z,t) = \sum_{n_x=1}^{N_x} \sum_{n_y=1}^{N_y} \sum_{n_z=1}^{N_z} \psi_{n_x}(x)\, \phi_{n_y}(y)\, \Gamma_{n_z}(z)\, r_{n_x n_y n_z}(t)\,. \quad (5b)$$

Since the TBL wall pressure field model is expressed in the frequency domain, Eq.(4) is rewritten in the frequency domain, as: $\mathbf{Y}(\omega)=\mathbf{H}(\omega)\mathbf{X}(\omega)$, where $\mathbf{Y}(\omega)$ is the response of the system to the excitation $\mathbf{X}(\omega)$, and $\mathbf{H}(\omega)$ is the frequency response matrix of the system, defined as following:

$$\mathbf{Y}(\omega)=\begin{Bmatrix} \mathbf{W}(\omega) \\ \mathbf{P}(\omega) \end{Bmatrix} \quad \text{and} \quad \mathbf{X}(\omega)=\begin{Bmatrix} \mathbf{P}_{tbl}(\omega) \\ \mathbf{0} \end{Bmatrix}, \quad (6a)$$

$$\mathbf{H}(\omega) = \begin{bmatrix} -\omega^2\mathbf{M}_{pp}+i\omega\mathbf{D}_{pp}+\mathbf{K}_{pp} & \mathbf{K}_{pc} \\ -\omega^2\mathbf{M}_{cp} & -\omega^2\mathbf{M}_{cc}+i\omega\mathbf{D}_{cc}+\mathbf{K}_{cc} \end{bmatrix}^{-1}. \quad (6b)$$

where $\mathbf{W}(\omega)$, $\mathbf{P}(\omega)$ and $\mathbf{P}_{tbl}(\omega)$ correspond to the frequency domain vectors of $\mathbf{q}(t)$, $\mathbf{r}(t)$ and $\mathbf{p}_{tbl}(t)$. One last step, needed to obtain a solution, is to transform the system equations to the PSD domain, as: $\mathbf{S}_{YY}(\omega)=\mathbf{H}^*(\omega)\mathbf{S}_{XX}(\omega)\mathbf{H}^T(\omega)$, where $\mathbf{S}_{XX}(\omega)$ is the PSD matrix of the random excitation, $\mathbf{S}_{YY}(\omega)$ is the PSD matrix of the random response, and superscripts * and T denote Hermitian conjugate and matrix transpose. Using this methodology, it is possible to analytically determine the PSD of the plate displacement, $S_{ww}(\omega)$, and the PSD of the acoustic enclosure pressure, $S_{pp}(\omega)$,. Please refer to [3] to access the derived analytical expressions for the PSD functions.

3. RESULTS AND DISCUSSION

Several practical cases available in the literature were used for the validation of our analytical framework. In the present summary, results are shown for the study in [4]. It consists of a rectangular simply supported aluminum panel, excited by a TBL, and coupled with an acoustical enclosure. The value of S_{ref} was provided, and the sound

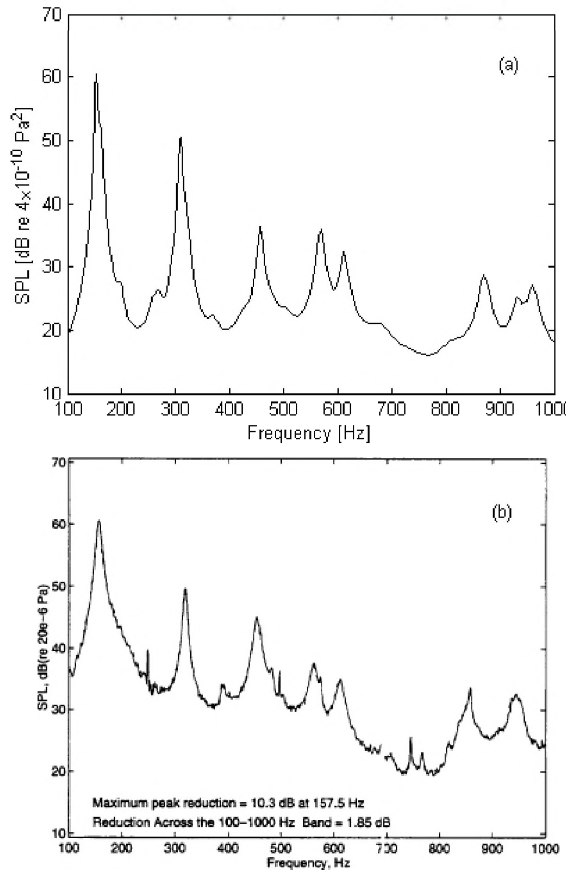

Fig. 1. SPL: (a) from our analytical model; (b) measured, from [4].

pressure level (SPL) was measured inside the acoustic enclosure. To accomplish convergence of the spectral quantities, a total number of M_x=5 and M_y=4 plate modes, and N_x=8, N_y=6 and N_z=5 acoustic modes were used in our analysis. For the frequency range of interest, [0; 1000] Hz, was necessary to include some non-resonant modes to accurately predict the SPL, showing that the number of plate and acoustic modes plays an important role to obtain a truthful prediction. By comparison of parts (a) and (b) of Fig. 1, it can be concluded that our analytical model provides a good approximation to the experimental data. It is shown that, for an accurate interior cabin noise prediction, is important to consider not only the structural natural modes, but also the cabin acoustic modes, as well.

REFERENCES

[1] D. E. Bishop, Cruise flight noise levels in a turbojet transport airplane. *Noise Control* (1961), 7, pp. 37-42.

[2] G. M. Corcos, Resolution of pressure in turbulence. *Journal of the Acoustical Society of America* (1963), 35(2), pp. 192-199.

[3] C. M. Heatwole, R. J. Bernhard and M. A. Francheck, Robust feedback control of flow induced structural radiation of sound. *NASA Report 0278-1* (1997).

[4] J. da Rocha, A. Suleman and F. Lau. An accurate coupled structural-acoustic analytical framework for the prediction of random and flow-induced noise in transport vehicles: Its Validation, *Canadian Acoustics (submitted, July 2009)*.