

TBL Modeling for Aircraft Interior Noise Prediction Using Statistical Energy Analysis

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ABSTRACT

The turbulent boundary layer (TBL) that forms on the outside of a commercial airplane in flight is a significant source of noise. During cruise, the TBL can be the dominant source of noise. Because it is a significant contributor to the interior noise, it is desirable to predict the noise due to the TBL. One modeling approach for the acoustic prediction is statistical energy analysis (SEA). This technique has been adopted by North American commercial airplane manufacturers. The flow over the airplane is so complex that a fully resolved pressure field required for noise predictions is not currently analytically or numerically tractable. The current practice is to idealize the flows as regional and use empirical models for the pressure distribution. Even at this level of idealization, modelers do not agree on appropriate models for the pressure distributions. A description of the wall pressure is insufficient to predict the structural response. A structural model is also required. For interior noise predictions a common assumption is that the motion of the structure is small and does not affect the boundary layer and wall pressures. Thus the boundary layer and structural modes are independent of each other. Therefore they can be solved and applied as a force to the structural model. Current aircraft models reported in the open literature are reviewed. Then a brief review of TBL modeling is presented. The application of these models to flat plate prediction is presented. The performance of the models is compared and contrasted. The effect of the choice of structural model is highlighted.

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INTRODUCTION

The turbulent boundary layer (TBL) that forms on an aircraft in flight can be the dominant noise source in the aircraft interior. The drive for more fuel efficient aircraft leads to the desire for lighter sound package treatments at constant sound pressure level. Implementing model based design helps achieve this constrained optimization process. The model tool of choice for North American producers for aircraft interior noise has been statistical energy analysis (SEA). This paper looks at the TBL contribution to aircraft interior noise and its modeling.

A review of aircraft interior noise can be found in Wilby [1]. Bremner and Wilby [2] laid out a case for simulation based design. Lin [3] presented a SEA modeling effort on the Boeing 737 in 1999. Davis [4] documented the move from modeling to production support and an organized modeling effort. Hawker-Beech applied SEA to an aluminum fuselage [5] and the Hawker Horizon [6]. The application of SEA to

aircraft sound package design was presented in 2007 by Davis [7]. Uncertainty in aircraft SEA modeling was covered at the same conference by Cousineau-Bouffard [8]. The load cases modeled were extended to shock cell noise in 2007 by Davis, Fung, and Ganz [9]. The application of SEA to the Gulfstream G150 is described in Maxon and Malluck [10]. Further modeling technique refinement from this application can be found in Wang and Maxon [11]. Another business jet example can be found in de Lima and Ravindran [12]. SEA was applied to a commercial freighter in Moeller, Allen, Barker, and Poling [13]. It can be seen from the brief review above that SEA is broadly adopted across North American aircraft manufacturers. As in the Automotive Industry, SEA modeling has moved from a research tool to a production environment.

The ability to predict the response to TBL forcing is critical to the design of new lighter more efficient aircraft.

The role of the turbulence models and the corresponding structural models are examined in this paper.

REVIEW OF TURBULENT BOUNDARY LAYER WALL PRESSURE FLUCTUATIONS

The TBL wall pressure fluctuations have been an area of active research. Bull [14] reviewed forty years of TBL wall pressure research. Much of the focus has been on zero pressure gradient naturally evolving flat plate boundary layers. This will also be the focus of the discussion here. Even something as notionally simple as a flat-plate boundary layer remains analytically and computationally intractable to this day. The focus of this paper is the wall pressure fluctuation measurements and the empirical wavenumber-frequency models. The Finnveden, Birgersson, Ross, and Kremer [15] experiment is used to evaluate the performance of the empirical wall pressure models in predicting flow induced noise and vibration.

There are several parameters that are important to the development of models of the TBL that forms on a flat plate at zero pressure gradient. Important parameters include fluid density, fluid viscosity, freestream velocity, distance along the panel in the flow direction and temperature. One derived parameter is the speed of sound in the fluid. The Mach number is the ratio of the freestream velocity to the sound speed. Another dimensionless parameter is the Reynolds number which can be characterized as the ratio of the inertial forces to the viscous forces. The Reynolds number is given by $\rho U_\infty x_0 / \mu$ where ρ is the fluid density, U_∞ is the freestream velocity, x_0 is the distance along the panel in the flow direction, and μ is the fluid viscosity. The boundary layer grows thicker as it progresses down the plate. There are several length scales that have been used to characterize this growth. The boundary layer thickness δ is defined as the location in the boundary layer where the flow has reached 99% of the freestream velocity. The displacement thickness, δ^* , is the equivalent amount of fluid displaced by the boundary layer. Similarly the momentum thickness, θ , is the equivalent momentum displaced by the boundary layer. Close to the wall, the flow is controlled by viscosity and this region is called the viscous sublayer. These distinct length scales lead to scaling laws that are called inner scale which is friction controlled or outer scale which is momentum controlled.

One measure of the pressure fluctuation intensity is the mean square pressure. There have been attempts to scale the mean square pressure on either outer or inner variables. These attempts have been only partially successful. One of the important second order statistics used to describe the wall pressure is the single point wall pressure spectrum, $\Phi(\omega)$. This describes the frequency distribution of the energy in the wall pressure at a single point. The mean square pressure can be recovered by integrating $\Phi(\omega)$ over all frequencies. A

review of these models can be found in Miller, Gallman, and Moeller [16].

The single point wall pressure spectrum is a topic of intense interest. Attempts to model the TBL with curve fits based on outer, inner or mixed variables met with limited success. Recent curve fittings by Goody [17] and Smol'yakov [18] have shown that a full frequency collapse requires mixed parameter scaling. This is because the low frequency content is outer variable controlled and the high frequency content is inner variable controlled. So a single inner or outer variable scaling does not collapse the full frequency range. Because of this inner/outer dependence, a simple mean square collapse on either inner or outer variables is clearly impossible.

There have been several attempts to relate the turbulent velocity fluctuations in the boundary layer to the resulting wall pressure. Recently three separate groups have used Reynolds averaged Navier-Stokes (RANS) and computational fluid dynamics (CFD) to develop mean velocity profiles. Lee, Farabee, and Blake [19], Peltier and Hambric [20], and Remmler, Christophe, Anthoine, and Moreau [21] have made meaningful predictions of the single point spectrum using these numerical techniques.

The single point wall pressure spectrum provides a description of the temporal content of boundary layer wall pressure. This is necessary for predicting vibration response but is not sufficient. It is also necessary to know the spatial distribution of energy as well as the temporal to predict the vibration. The statistics used to describe the space time content can be either in the space-time domain or the wavenumber-frequency domain. The space time statistic is the cross-correlation function. The space time statistic can be Fourier transformed in time and space to arrive at a wavenumber-frequency spectrum. The two statistics each contain the same information. Each has advantages and disadvantages. A recent review of these statistics can be found in Graham [22].

There are many space-time statistical models for the wall pressure. Some of these models are reviewed Miller, Gallman, and Moeller [16]. One of the earliest models is one proposed by Corcos [23]. Efimtsov [24] did an update to the Corcos model. Chase [25], Mellen [26, 27], Smol'yakov and Tkachenko [28] all proposed other models. The Corcos [23], Efimtsov [24] and Chase [25] model predictions are compared to the Finnveden, Birgersson, Ross, and Kremer [15] experiment in the following sections.

SEA POWER INPUT DUE TO TBL LOADING

SEA models predict the diffusion of energy through a system. One of the inputs required to predict the system response is the input power. The TBL is one source that is used in SEA system models of aircraft. The SEA model divides the aircraft into many substructures. These substructures are called subsystems. Subsystems only interact at their boundaries. The TBL forms over the outer skin of the

aircraft. The assumption is made that the TBL is spatially homogenous over the extent of each subsystem. This allows a local wavenumber -frequency spectrum to be calculated for each exterior subsystem.

Equation (1) from LeVitte [29] relates the subsystem properties and flow properties to predict the local input power for the SEA model:

$$\Pi_{inj}(\omega) = 2S_{pp}(\omega)\Delta\omega\frac{\pi S}{\rho_s}n_1(\omega)J^2(\omega) \quad (1)$$

where:

$\Pi_{inj}(\omega)$ is the input power of interest

$S_{pp}(\omega)$ is the single point wall pressure spectrum

$n_1(\omega)$ is the modal density

S is the plate area

ρ_s is the surface mass density of the structure

$J^2(\omega)$ is the joint acceptance.

The modal joint acceptance can be calculated either in the space-time or wavenumber-frequency domains. Equation (2) gives the joint acceptance in terms of the wavenumber-frequency domain. It describes how the structural modes interact with the TBL wall pressures.

$$J_{mn}^2(\omega) = \frac{1}{(2\pi S)^2} \int_{-\infty}^{\infty} \int_{-\infty}^{\infty} S_{pp}(k_x, k_y, \omega) |f_{mn}(k_x, k_y)|^2 dk_x dk_y \quad (2)$$

Here, the joint acceptance $J_{mn}^2(\omega)$ is the frequency averaged modal joint acceptance.

where:

$S_{pp}(k_x, k_y, \omega)$ is the wavenumber-frequency spectrum

$|f_{mn}(k_x, k_y)|^2$ is the wavenumber filter shape for the m, n mode.

The TBL input power is calculated and applied to the SEA system model.

EXAMPLES AND DETAILED DISCUSSION

There are a few studies in the open literature that can be used as a basis for discussion purposes. One is by Finnveden, Birgersson, Ross, and Kremer [15] and it is the one that is used here. It is a clamped aluminum plate tested at 80, 100 and 120 m/s. The plate was 76.8 cm long and 32.8 cm wide and flush mounted in the bottom wall of a wind tunnel. The plate was 1.6 mm thick.

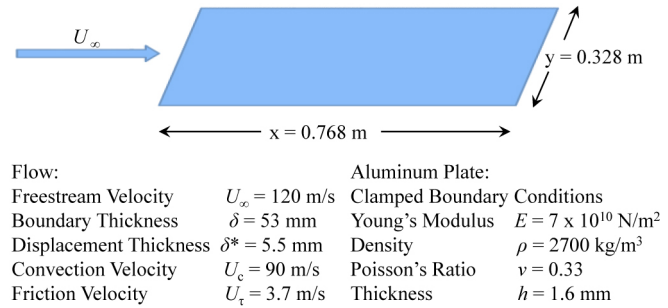


Figure 1. Finnveden, Birgersson, Ross, and Kremer [15] plate experiment.

The Finnveden, Birgersson, Ross, and Kremer [15] experiment shown in Figure 1 was conducted in air. The wind tunnel was designed to be isolated from acoustic noise by enclosing it in a reverberation room. The tunnel was a closed loop system with the inlet was in an anechoic room and the exhaust was in an adjoining semi anechoic room. The test section was in a connected reverberation room. The boundary layer was allowed to naturally evolve on the bottom wall of the test section. Extensive wall pressure and boundary layer measurements were made. The plate damping loss factors were measured. The plate response to TBL excitation was measured using light weight accelerometers.

The experiment was designed to examine the performance of two wavenumber-frequency wall pressure models, namely the Corcos [23] and Chase [25] models. There are three speeds of interest in this problem. The first is the bending wave speed in the plate. The second is the speed of sound, C_a . The third speed is the convection velocity, U_c , which the speed of the dominant energy in the TBL. Finnveden, Birgersson, Ross, and Kremer [15] found the convection velocity to be

$$U_c = 0.75U_{\infty} \quad (3)$$

The units of a wavenumber are radians per unit length and a wavenumber is generally defined as 2π divided by the wavelength at a fixed frequency.

There are three wavenumbers that are important for this problem. The first is the plate free wavenumber given by

$$k_p = \sqrt[4]{\frac{\rho_p h \omega^2}{D}} \quad (4)$$

where the bending rigidity is given by

$$D = \sqrt{\frac{Eh^3}{12(1-\nu^2)}} \quad (5)$$

The acoustic wavenumber is simply the frequency divided by the speed of sound.

$$k_a = \frac{\omega}{C_a} \quad (6)$$

And the convective wavenumber is

$$k_c = \frac{\omega}{U_c} = \frac{\omega}{0.75U_\infty} \quad (7)$$

When the plate free wavenumber equals the convective wavenumber, the TBL and plate strongly interact. The wavenumbers important to this problem are plotted in [Figure 2](#).

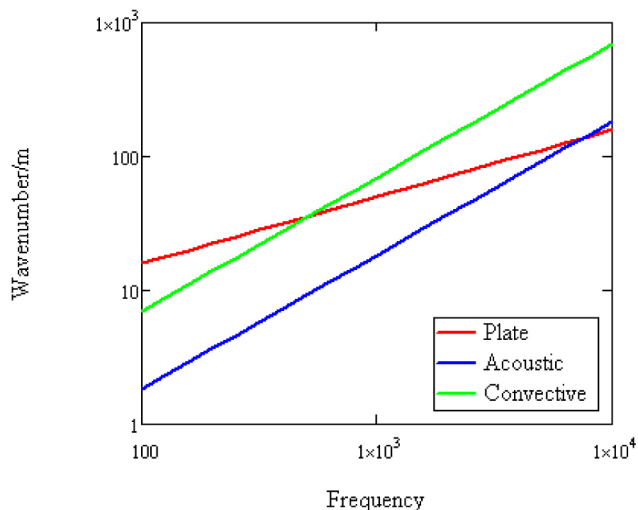


Figure 2. Wavenumbers.

The plate free wavenumber gives a characteristic length scale for the plate motion. The plate boundary conditions also play a role in the plate response. Finnveden, Birgersson, Ross, and Kremer [15] modeled the plate boundary conditions as clamped. A fully clamped plate does not have an analytical solution. A simply supported plate has an analytical solution. The boundary conditions do play a role in the plate response to both acoustic and TBL response. The finite plate has discrete resonant frequencies. The equation of motion for a thin plate is given by:

$$\rho_p h \frac{\partial^2 w}{\partial t^2} + D \nabla^2 w = F \quad (8)$$

A rectangular plate of length 'a' and width 'b', simply supported plate results in sinusoidal mode shapes.

$$w(x, y) = \sum_{m,n=1}^{\infty, \infty} a_{m,n} \sin\left(\frac{m\pi x}{a}\right) \sin\left(\frac{n\pi y}{b}\right) \quad (9)$$

Each mode pair m, n represents a unique resonance. The resonant frequency will be:

$$\omega_{m,n} = \sqrt{\frac{D}{\rho_p h} \left[\left(\frac{m\pi}{a} \right)^2 + \left(\frac{n\pi}{b} \right)^2 \right]} \quad (10)$$

A plate resonance occurs whenever a half wavelength fits in both directions simultaneously. So modes occur in m half wavelengths longitudinally and n half wavelengths transverse to the flow.

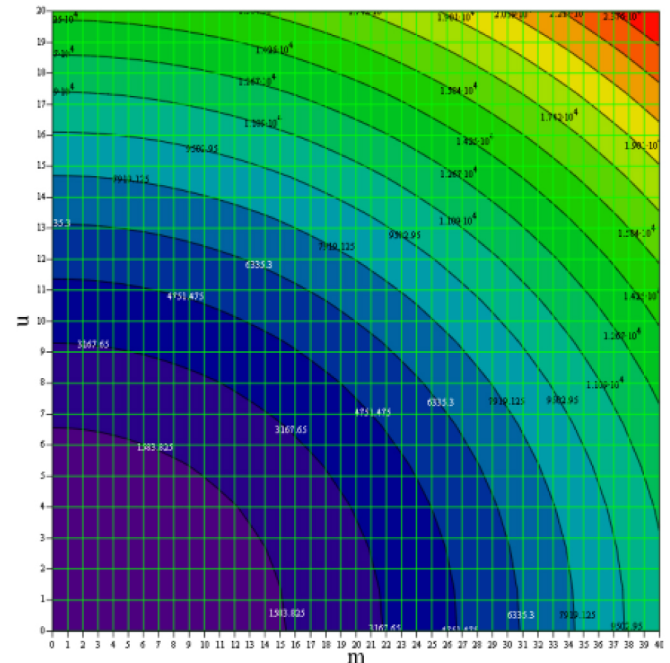


Figure 3. Plate resonant frequencies.

In [Figure 3](#), each lattice point represents a mode. The resonant frequencies are cross plotted on the graph. One of the important parameters in SEA is the modal density. The modal density is the total number of modes in a frequency band divided by the bandwidth. Lyon [30] computed the modal density for plates and showed it was constant with frequency. The total mode counts are shown in [Figure 4](#) for