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Simulating Nonlinear Stator Noise for Active Control

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1. INTRODUCTION

A major source of noise from modern turbofan engines is the noise produced from vortical gusts (rotor wakes) impinging on a stationary blade row.[1,2] Analytical techniques for predicting sound levels from this interaction have been developed,[3–7] but were based on linearizing the flow equations and simplified geometries. Numerous techniques have been investigated in the past to numerically simulate the so-called "rotor wake/stator interaction" effect of realistic blade geometries. Most of those techniques required strong assumptions about the flow and the more successful approaches were often limited to linearized single frequency calculations[8–9] which avoided dissipation and dispersion errors. And time-accurate codes have met with mixed success on flat plate cascades with uniform flows.[10–14] Full fan configuration prediction codes often require:

- blade acoustic source definition,
- linearizing the flow equations,
- and the use of linear theory for acoustically post-processing RANS solutions.[15–17]

A recent full fan configuration that was based upon first principles only (solves thin-layer, Navier-Stokes equations) was not able to successfully simulate the generation and propagation of acoustic signals due to its inherent dissipation and dispersion.[18]

Building on these past efforts, NASA Glenn Research Center has developed a Broadband Analysis Stator Simulator (BASS) code[19] which for the first time successfully simulates the full nonlinear, time-accurate, acoustic response of an unrolled section (2D cascade) at a radial location of a modern stator vane configuration. Our simulation assumes inviscid and ideal flow, but it solves for the entire flow field using a truly time-accurate methodology that enables the simultaneous calculation of the entire range of frequency response up to at least 3 times the Blade Passage Frequency (BPF). Initial comparisons with established theory show good agreement.[20]

This encouraging development not only suggests that the prospect of accurate 3D calculation of rotor-stator interaction noise is within reach, it also hints at the possibility of rigorous investigation of other aspects of the turbomachinery noise problem. In particular, it seems possible to study active noise control strategies using a numerical code (like the BASS code). Now, this is not a new idea in that, at least at low frequencies, linearized inviscid flow solvers[21] have been used to investigate active noise control via surface mounted actuators. However, from a numerical, as well as a practical point of view, one of the shortcomings of current active noise control approaches is their frequency limitation. For problems of engineering interest, control frequencies in excess of 1 kHz are typically needed to eliminate (or at least mitigate) unwanted noise.

To circumvent this problem, it is proposed to use acoustic resonance produced at a low frequency to exert control at high frequencies. The production of high frequencies from low frequency resonance is accomplished through nonlinear processes that don't come into play at low signal amplitudes. However, at sufficiently high signal amplitudes, inherent nonlinearities in the equations of motion lead to production of high frequency fluctuations from low frequency fluctuations. Such high amplitude fluctuations are present at or near acoustic resonance where favorable conditions lead to a positive feedback type build up of acoustic amplitudes to extremely high levels. By careful tailoring of the conditions, it may be possible to cancel undesirable high frequency signals through production of a low frequency resonance. The advantage of such a method over the conventional approaches is that is it usually easier to produce controlled signals of desired characteristics (amplitude and phase) at lower frequencies than it is at high frequencies.

This paper outlines an idea to achieve this type of active control with on-blade actuators. First, a simple analytical example demonstrates how a simple actuator could be used to nonlinearly control sound. And second, results from a realistic geometry demonstrate high amplitude vortical gusts produce nonlinear harmonics that can destructively interfere with otherwise unattenuating (propagating) acoustical modes. We focus on the 2 BPF acoustic response here to simplify the presentation and to show that nonlinear cancellation is effective on realistic blade rows, but similar simulations for higher frequencies have also been successfully completed.

2. ACTIVE NOISE CONTROL APPROACH

Nonlinear acoustics began with Leonard Euler when he developed finite-amplitude sound equations (Euler Equations) which we write in curvilinear coordinates as:

$$Q_t + \xi_t Q_\xi + \eta_t Q_\eta + \xi_x E_\xi + \eta_x E_\eta + \xi_y F_\xi + \eta_y F_\eta = 0$$

$$\{Q\} = \begin{Bmatrix} \rho \\ \rho u \\ \rho v \\ E \end{Bmatrix}, \{E\} = \begin{Bmatrix} \rho u \\ \rho u^2 + p \\ \rho uv \\ u(E+p) \end{Bmatrix}, \{F\} = \begin{Bmatrix} \rho v \\ \rho uv \\ \rho v^2 + p \\ v(E+p) \end{Bmatrix}, \begin{matrix} \xi = \xi(x,y,t); \eta = \eta(x,y,t) \\ p = p(x,y,t), u = u(x,y,t), v = v(x,y,t) \\ p = (\gamma - 1)\left(E - \frac{\rho}{2}\left(u^2 + v^2\right)\right) \end{matrix}$$

At an inviscid surface, the impermeability (no-flow) condition must be satisfied and its higher time derivatives:[22]

$$\frac{\partial^\alpha (\vec{V} \bullet \vec{\eta})}{\partial t^\alpha} = \frac{\partial^\alpha (\eta_t + \eta_x u + \eta_y v)}{\partial t^\alpha} = 0$$

Using only those equations, it is possible to investigate a simple nonlinear noise control strategy. Approximate an airfoil actuator as a pulsating circle so the metric functions are:

$$\eta(x,y) = x^2 + y^2 + A\cos(\omega t + \phi) + 4; \xi(x,y) = y/x$$

$$\eta_x = 2x, \eta_y = 2y, \eta_t = -A\omega\sin(\omega t + \phi), \xi_x = -y\big/x^2, \xi_y = 1\big/x, \xi_t = 0$$

At the surface of the actuator notice that the time derivatives of pressure are:

$$p_t = -p_\eta \overset{(=-\eta_t)}{\underline{(v\eta_y + u\eta_x)}} - \gamma p(v_\eta \eta_y + v_\xi \xi_y + u_\eta \eta_x + u_\xi \xi_x) + p_\xi (v\xi_y + u\xi_x)$$
$$= -p_\eta \underline{(-A\omega \sin(\omega t + \phi))} - \cdots$$

This result follows from letting $\alpha = 0$ in the no-flow boundary condition. Notice that the p_n term increases the effective actuator frequency and enables higher frequency control with a lower frequency actuator. And similarly, the second time derivative of pressure is:

$$p_{tt} = -p_\eta(-\eta_{tt}) + p_{\xi t}(-\eta_t)\cdots$$

The p_{nt} term provides additional frequency boosting. Clearly, an effective control strategy needs to adjust the phase and amplitude of these higher order pressure derivatives and nonlinear acoustics plays an important role.

Further understanding of nonlinear frequency response is gained by expanding the dependent variables in a complex Fourier series in, ω, the dominant excitation frequency:[23–24]

$$\rho(x,y,t) = \sum_{n=-\infty}^{\infty} \rho_n(x,y) e^{i\omega nt}, u(x,y,t) = \sum_{n=-\infty}^{\infty} u_n(x,y) e^{i\omega nt}$$
$$v(x,y,t) = \sum_{n=-\infty}^{\infty} v_n(x,y) e^{i\omega nt}, p(x,y,t) = \sum_{n=-\infty}^{\infty} p_n(x,y) e^{i\omega nt}$$

Next, substitute these series into the Euler equations and evaluate the Taylor series of pressure to find the expected response of the flow field:

$$p(x,y,t+\Delta t) = p(x,y,t) + p_t(x,y,t)\Delta t + p_{tt}(x,y,t)\Delta t^2 + \cdots p_N(x,y,t)\Delta t^N$$

If only one input frequency is assumed present in the flow variables, ω, then not only is the original frequency, ω, present in the response, but the following frequencies are naturally occurring as well:

- $2\omega, 3\omega$ up to $(N+1)\omega$

Spatial wavenumbers are also amplified by nonlinear effects and these naturally occurring high frequency/wavenumber signals have the potential for enabling active acoustic mode control with low frequency devices. An example of significant nonlinear mode cancellation in a realistic blade row is shown in the next section.

3. NONLINEAR MODE PRODUCTION AND CANCELLATION

Our initial tests on a 2D realistic stator row suggest nonlinear effects can produce acoustic mode cancellation. We simulate a 54 blade loaded cascade for its acoustic response from a 22 blade rotor as described in figure 1. Due to periodicity we only need to solve one-half of the fan. Our mode orders are also reported as half of the full fan stage.

When all three gust frequencies are provided as input, we notice constructive interference on circumferential mode, m = –5, at the inflow and destructive interference at the outflow as shown in figure 2. The mechanism of this interference is shown in figures 3, 4, 5 in which the 2 BPF response from independently applied 1, 2 and 3 BPF gusts. Notice that the amplitude of the nonlinearly produced 2 BPF acoustic response from the 1 and 3 BPF gusts is of the same relative order as the 2 BPF acoustic response from the expected (from linear theory) 2 BPF gust input despite the fairly large differences in the amplitudes of the gust inputs (fig. 1).

The beating behavior seen at the outflow can only be produced by three physical mechanisms:

- First, when two wavetrains of the same frequency travel along the same line in opposite directions, standing waves are formed (reflections from the outflow)
- Second, when two wavetrains of nearly the same wavenumber/frequency travel through the same region (interference in time).
- Third, when two wavetrains of the same wavenumber change their relative phase very slowly in comparison to the wavenumber

By adjusting the gust amplitudes downward by a factor of four we find

- the beating phenomena in the outflow of figure 2 vanishes.
- at 2 BPF, the mode m = –5 propagates without interference.
- nonlinear mode production ceases to be a significant factor in the simulation

A reflecting outflow would still produce beating at the lower amplitude. And the sound levels in the outflow region are in the linear range and outside the wakes the flow is nearly uniform. The wave equation may therefore be applied and for a particular Fourier acoustic signal: $p = p_n \cos(k_x x + k_y y - wt)$, we know given $\omega = 2BPF$ and $k_y = -\frac{5\pi}{9}$, and mean velocity (U,V), and speed of sound, a, that the wavenumber in the axial direction must be:[5]

$$k_x = \frac{U(\omega - k_y V) \pm \sqrt{k^2{}_y (U^2 + V^2 - a^2) + 2k_y \omega V + \omega^2}}{U^2 - a^2}$$

With these assumptions, the amplitude of the $2BPF$, mode $k_y = -\frac{5\pi}{9}$, shown in figure 2, must have the same axial wavenumber whether produced nonlinearly or not. However, since the flow is not uniform in the outflow, the wave equation need not apply.

The third physical mechanism is the acoustical modes change phase periodically due to nonlinear mode production in the blade passage. For example, if the expected linear response from a $2BPF$ gust is described at the outflow by, $p_1 = \cos(k_x x + k_y y - wt)$, and a second response at $2BPF$ is produced from the $1BPF$ gust with a phase that varies axially described by, $p_2 = A\cos(k_x x + k_y y - wt + \phi(x))$, then the linear supposition of these two waves at a particular frequency ($\omega = 2BPF$) and circumferential mode order (m = –5) is:

$$p(x) = p_1 + p_2 = A\left(2\cos\left\{-\frac{\phi(x)}{2}\right\}\cos\left\{k_x x + \frac{\phi(x)}{2}\right\}\right)$$

If the phase change, $\phi(x)$, is small compared to the axial wavenumber, k_x then beating would be observed.

Clearly more study is required, but it is hoped some of the complexity encountered in active control systems[25] can be avoided since a phase relationship exists between the modes as they propagate through the same nonlinear mean flow.

4. CONCLUSION

The importance of nonlinear effects in flow and noise simulation has encouraged the development of multiple solution techniques[26] that either extend single-frequency analysis or use models to describe nonlinear interaction. One clear advantage of our simulation approach is the ability to isolate specific gust and modal behaviors so nonlinear effects can be quantified, understood, and hopefully controlled. The mathematics of nonlinear acoustics is quite complicated, particularly when realistic geometries are included and the goal of low-noise fan design requires a better understanding of the sound mechanisms that analytical techniques are not likely to provide. Our new simulator provides an opportunity to experiment with noise control strategies as new ideas develop.

This paper is only an overview of our approach and recent 2D results. We have briefly shown an example of nonlinear noise cancellation. The intent of this work is to demonstrate that nonlinear effects on realistic geometries can be a tool for fan noise reduction. The development of a robust control strategy is left for later work. Our work is an ongoing effort to produce a tool for 3D acoustically damped fan design.

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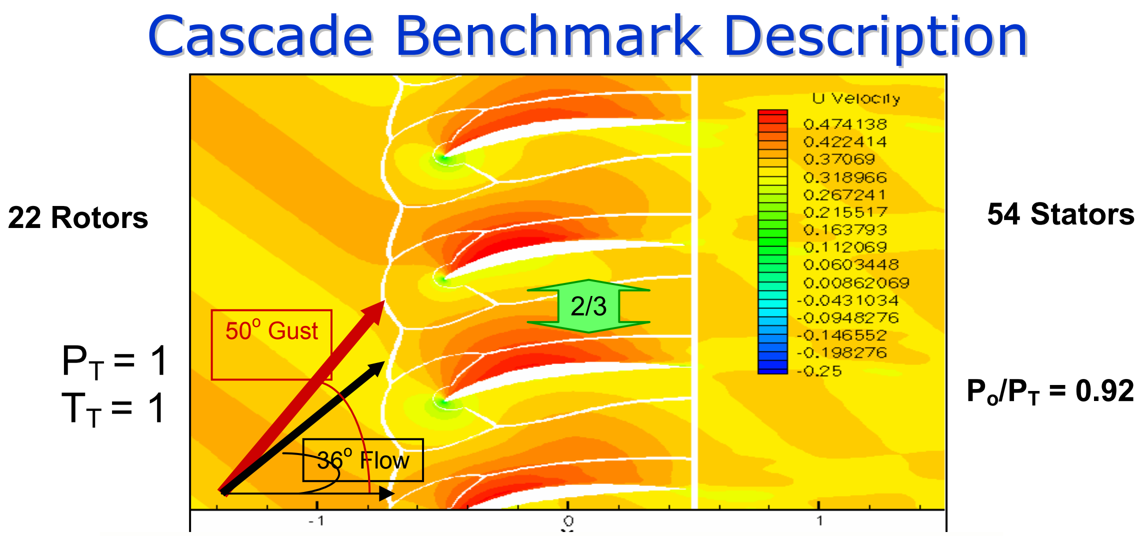

$$\vec{u}'_g = \left\{ \sum_{n=1}^{3} a_n \cos(n(k_y y - \omega t) + \varphi_n) \right\} \hat{e}_\beta$$

$$a_1 = 0.03, a_2 = 0.003, a_3 = 0.0007, \varphi_1 = 0, \varphi_2 = \tfrac{-7\pi}{5}, \varphi_3 = \tfrac{-\pi}{2}, \omega = \tfrac{3\pi}{4}, k_y = \tfrac{11\pi}{9}$$

Figure 1. —Description of Cascade Benchmark Problem.

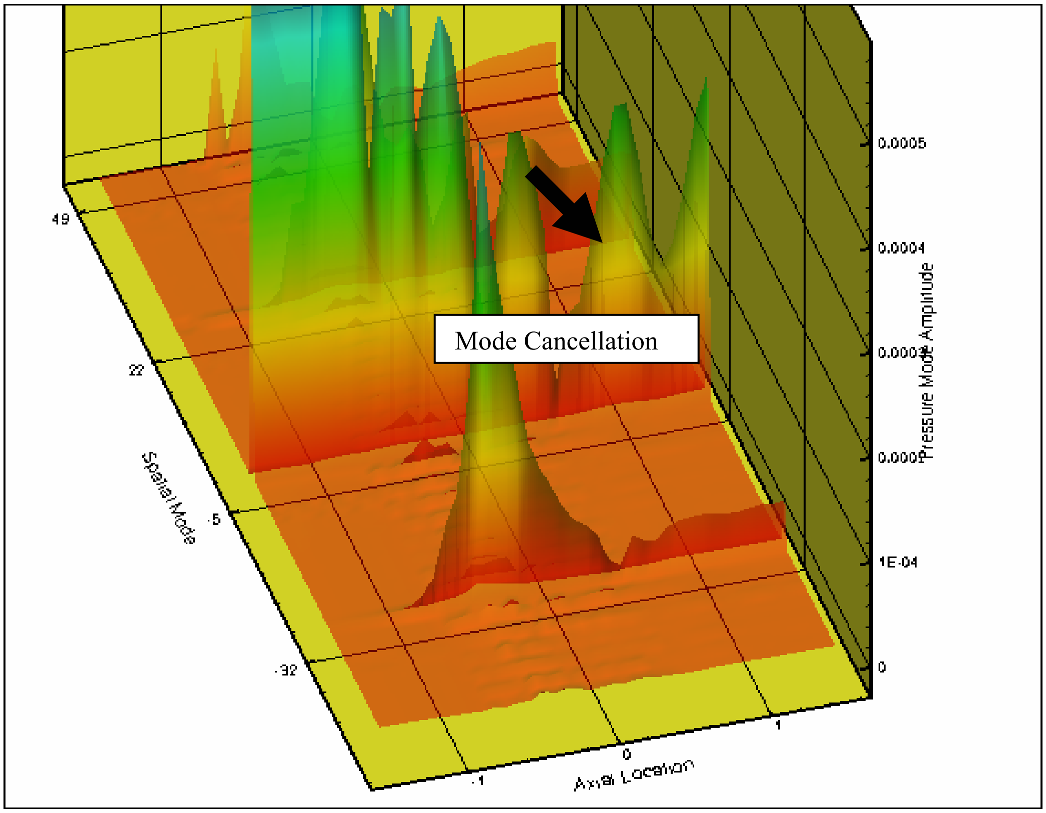

Figure 2.—Nonlinear Destructive Interference at 2 BPF, m = –5, from x = .5 to 1.5. Beating phenomena vanishes as nonlinear effects are decreased.

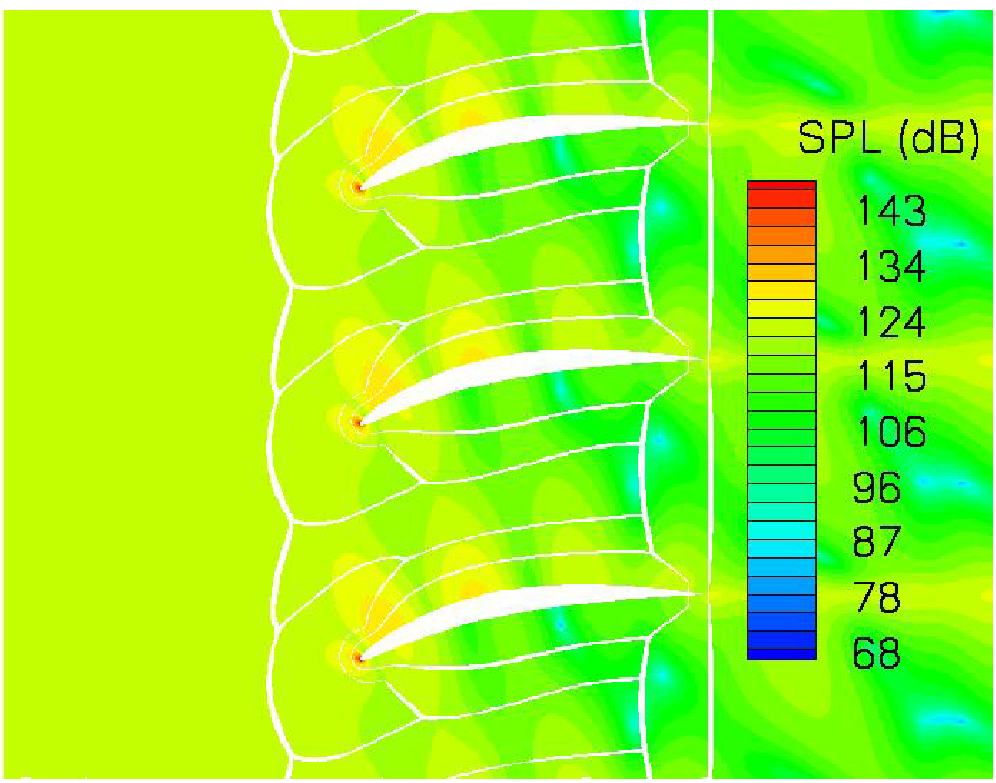

Figure 3.—1 BPF Gust—2 BPF Response.

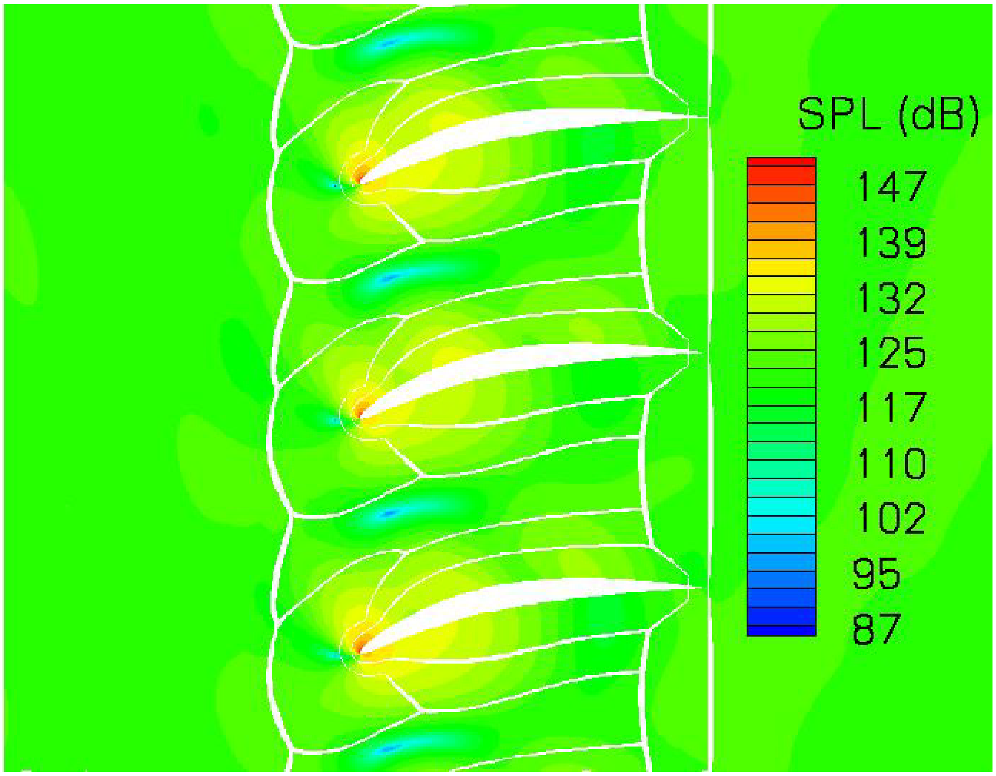

Figure 4.—2 BPF Gust—2 BPF Response.

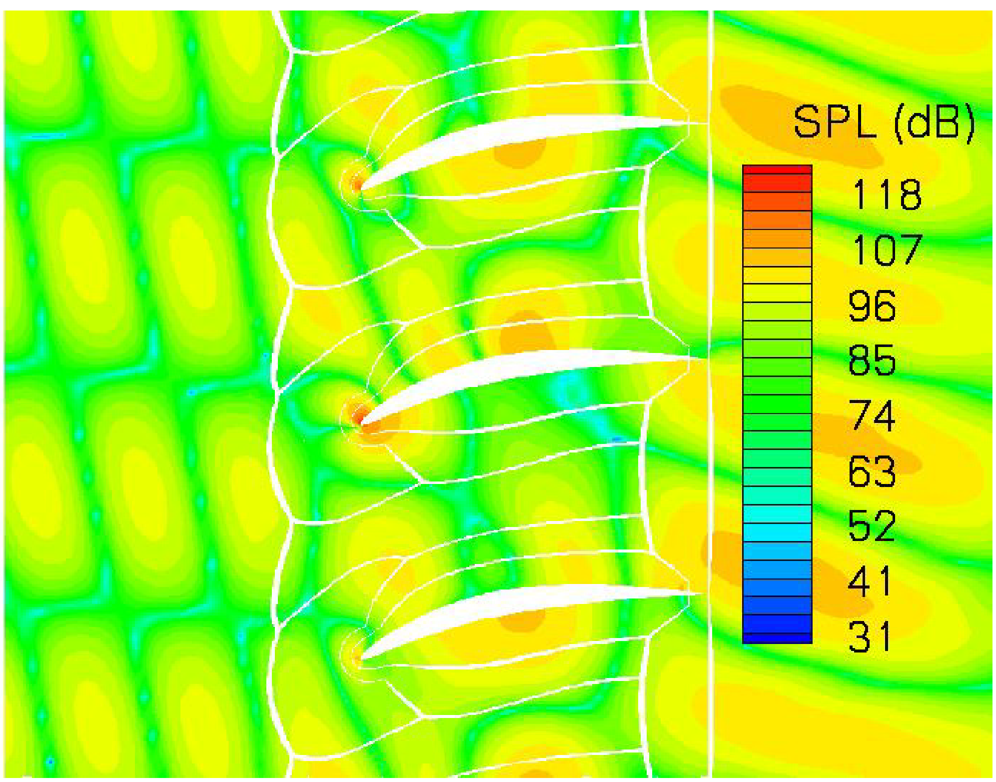

Figure 5.—3 BPF Gust—2 BPF Response.

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13. ABSTRACT ***(Maximum 200 words)***

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