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Modeling fan noise with time-domain CFD within the product design cycle

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ABSTRACT

In this work, we present the results of a new fan noise estimation process, based on time-domain, Lattice-Boltzmann computational fluid dynamics. This approach, coupled with modern advances in high performance computing, has brought the accurate modeling of (installed) fan noise spectra well within the short design cycles typical of modern product programs.

We demonstrate the accuracy of the computational method through comparison with data from three axial fans operated in a mocked-up engine compartment. We extend those results to a design variation that demonstrates a reduction of total fan noise. Finally, we explore process cost, including computational requirements and the total time from CAD design to acoustical results.

1. INTRODUCTION

In controlling sound emissions from off-highway machines and equipment, noise from air-moving devices typically predominates over other sources. These can include engine cooling fans, blades for mowing or chopping, and blowers for material handling or HVAC. For example, introduction of EPA Final Tier 4 (EU Stage IIIB) regulations on diesel-powered equipment has increased cooling requirements up to 20% over Tier 3. Constrained from increasing area for cooling packages, fans are now forced to move more air in the same footprint, resulting in increased radiated noise.

Lacking reliable, affordable acoustical models, fan system designers have not had specific, useful guidance in designing for low noise. While the Fan Laws [1, 2] help establish levels of broadband, turbulent noise, they are ineffective at predicting the powerful Blade Passage Frequency (BPF) tones. These tones are dominant noise sources, and they are typically exacerbated by installation effects. As a result, noise reduction from such air-moving devices waits until prototypes are built; at that point little flexibility remains for potential design changes.

Computational technologies have evolved over the last decade for acoustical modeling of aero-acoustical systems where rotating blades are the acoustical sources [3, 4]. Until very recently,

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while these models have proven sufficiently accurate, they have been too expensive for the typical industrial environment and too slow to impact products within compressed design cycles.

In this paper, we outline an approach to estimating the acoustical radiation from rotating axial-flow fans directly from a Computational Fluid Dynamics (CFD) calculation. Leveraging a time-domain Lattice-Boltzmann method for computing flow from fans, and affordable High-Performance Computing (HPC) hardware, we successfully demonstrate our approach on a full-scale mock-up of a cooling system and engine compartment. We have shown that the entire process, from CAD geometry to acoustical spectra, can be accomplished within one business week using commonly-available HPC resources.

Section 2 outlines the computational approach. Section 3 describes a set of laboratory measurements used to correlate the computational approach to a real-world application. Section 4 demonstrates the correlated model results and briefly explores one design variation capable of reducing overall sound levels.

2. COMPUTATIONAL APPROACH

A. Using Computational Fluid Dynamics to support Aero-Acoustics

Two main approaches to handle Computational Aero-Acoustics (CAA) problems have emerged over the past decade.

In the first approach, a sequential, one-way (uncoupled) calculation begins with a time-domain CFD computation from which aero-acoustical source terms are derived. In a first step these source terms are estimated from the fluctuating pressure, density and velocities using the Lighthill's [5] or the Ffowcs-Williams and Hawking's [6] acoustic analogies. In a second step, the reconstructed source terms are propagated to the far-field using either a spectral method or time domain algorithms [7]. The most common CFD algorithms in use for low Mach number acoustical source derivation are Large Eddy Simulation (LES) and Dynamic Eddy Simulation (DES) [8]. Common acoustical propagators include integral methods, Boundary Element Methods (BEM) and Finite Element Methods (FEM). In the latter case, the volume mesh must be bounded by Infinite Elements or some other exterior matching layer scheme, in order to accurately estimate far-field radiation. This approach has been successful for rotating fans in the past [9, 10], and work in this area continues.

In the second approach, acoustical pressure fluctuations are obtained directly from the CFD domain, without reference to an aero-acoustical analogy or an acoustical propagator. Accurate estimation of aero-acoustical quantities directly from Computational Fluid Dynamics (CFD) that are 3 to 5 orders of magnitude smaller than the fluctuating flow variables is extremely challenging. The direct CFD approach has become more accessible with the advent of new solvers, 64-bit word-lengths in computation, and massively-parallel processor technology at affordable prices. As with the sequential approach, most Finite Volume and Finite Element CFD codes make use of LES/DES in solving the Navier-Stokes equations.

In the present work, we make use of the second, direct approach to applying CFD to CAA. However, instead of solving the Navier-Stokes equations, we employ the Lattice-Boltzmann Method (LBM) to solve the flow field in the time domain. This method is outlined briefly in the next Section, 2.B.

Regardless of the CFD method one selects for CAA, the parametric demands on these codes required to effectively propagate acoustical quantities are substantial; they will be discussed in Section 2.C. below.

B. Application of the Lattice-Boltzmann Method to Direct Solution of Aero-acoustics

The CFD/CAA code discussed in this paper is based on the Lattice-Boltzmann method (LBM). With it, the time-varying, unsteady flow is modeled, along with the corresponding flow-induced noise source generation and radiation [11]-[16]. Lattice-based methods are explicit, transient and compressible, and are an alternative to traditional CFD methods based on the discretization of the Navier-Stokes equations and derived variations.

Fundamentally, LBM tracks the advection and collisions of fluid "particles". Since the average number of particles in a representative volume of fluid far exceeds the computational power required to track them individually, the particles are grouped into an integer number of discrete directions i. The computation follows the particle distribution function f_i, which represents the number of particles per unit of volume (referred to as a "voxel") at a specific time and location and moving with velocity c_i. As in statistical physics, the flow variables such as density and velocity are determined by taking the appropriate moments, *i.e.* summations over the set of discrete directions of the particle distribution function. LBM has been validated across many aeroacoustics applications, including automotive wind-noise [17], HVAC system noise [18], and acoustical propagation [19]. Further details on use of LBM in aeroacoustics can be found in the cited references.

C. Performance Requirements in an Industrial Setting

In setting out to accomplish this work, our goals go well beyond simply proving technical feasibility. Instead, we require that the entire process, from CAD geometry to Sound Power Level spectra, be available well within the typical product design cycle. In the off-highway machine industry, the primary design cycle can be as brief as several weeks. We set a goal of completing the end-to-end process in one business week or less. We further required that the computations can be run affordably using HPC (computer cluster) technology commonly available to typical OEM's. In this Section, we outline the parametric requirements for estimating sound radiation from rotating fans, and describe how our selected approach meets these requirements while satisfying our stringent performance goals.

Estimation of acoustical radiation from steady-state turbulent flow is a well-established technology [20, 21]; it is used to estimate aircraft fuselage and automotive body turbulent boundary layer-induced noise. Rotating systems, however, are much more involved. Instead of imposing static flow field, the rotating element is itself the prime mover of the fluid. This requires that the code be capable of supporting a rotating mesh, the surrounding static mesh, and an interface layer between these meshes. Further, the code must accommodate the transfer of flow variables across that interface at each time step. This crucial capability is often the Achilles heel for CFD approaches to CAA; data must be passed from rotating to non-rotating meshes with sufficient accuracy that no artifacts arise at the level of the acoustical fluctuations.

The choices of grid size and time step are likewise crucial, and comprise the fundamental expense of the computations. The mesh surrounding the fan and in its near proximity must be fine enough to support the small turbulence length scales of interest. In the CFD code [23], Variable Resolution (VR) meshing strategy is employed, where the grid size changes by a factor of two for adjacent resolution regions.

Achieving frequency resolutions in the acoustical spectra sufficient to identify blade tones, say 4 Hz, requires computation of a significant fraction of 1 second's worth of data. Even more time steps are needed if overlap-averaging techniques are to be employed to reduce both the spectral amplitude spread (confidence interval) and spectral leakage (via windowing).

Combining the large grid sizes with the number of time steps presents a significant computational hurdle.

3. CORRELATION EXPERIMENTS

To correlate the computational process developed for this work, we selected a system that is representative of a real-world application. A mock-up of the engine compartment of a large hydraulic excavator was built and tested in a Hemi-Anechoic Chamber. The geometry of the mocked-up system was then translated to CAD and used directly to build the CFD/CAA model. The mock-up, acoustical measurements, and fans that were tested are described in this Section.

A. Engine Compartment Mock-up and Acoustical Measurements

A full-scale mock-up of a large hydraulic excavator's engine compartment was built using 16 mm (5/8 inch) plywood, maintaining the interior dimensions and inlet/outlet areas, as shown schematically in Figure 1. An adjustable panel on the intake was used to obtain correct flow and pressure rise for the system. The cooling package consists of an engine coolant radiator, a charge-air cooler, a hydraulic oil cooler, a fuel cooler, and an air-conditioning condenser. Behind the coolers, the fan was mounted within its production shroud, as shown in Figure 2. The fan was driven by a hydraulic motor, whose supply was maintained outside the chamber. Sound radiation from the motor consisted of several very narrow harmonic tones, which were easily excluded from the noise measurement spectra.

Five microphones were placed around the system. They were not intended for a sound power measurement; their placements were intended to provide key indicators of the fan noise. Microphone 1 had line-of-sight to the fan axis through the intake port and the coolers.

Figure 1. CAD image of the engine compartment mock-up. Air enters from the right and exits out the top.

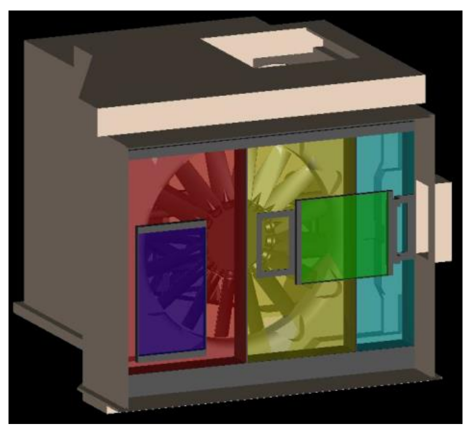


Figure 2. CAD image of the cooling package and fan mounted within the engine compartment mock-up.

B. Engine Cooling Fans and their Sound Spectra

Three very different fans were tested in the mockup. Each was a current production model from a different manufacturer. Key parameters for the fans are listed in **Table 1**. All of the fans were 965.2 mm in diameter. Each was immersed in the shroud per manufacturer's specifications. Each fan's speed was then selected to provide similar pumping capacity.

Significant differences in the blade numbers and airfoil designs of each fan contributed to dissimilar radiated sound characteristics for each fan. These differences are important in evaluating the efficacy of the modeling process.

Table 1. Parametric description of the three fans.

Parameter	Unit	FAN #1	FAN #2	FAN #3
Number of Blades	#	16	9	8
Blade speed	RPM	1550	1170	1220
BPF	Hz	413	176	163
Blade Tip Velocity	m/s	78	59	62

4. THE COMPUTATIONAL AERO-ACOUSTICS MODEL

A. Parameters for CFD Solution

Each fan was modeled directly from the manufacturer's CAD geometry and placed into the model shroud with the recommended immersion. **Table 2** lists the parameters used in the CFD model for each run, along with some information on the computational effort. The CFD grid required approximately 200M voxels. The finest grid resolution was Δx=0.625mm, resulting in 1540 voxels around the fan blades and 20 voxels across the tip gap clearance.

The simulation time step was $\Delta t=1.024\text{x}10^{-6}$ sec, and the total physical time modeled was T=1.3 sec, in order to obtain a smooth 4 Hz narrowband spectrum. This time corresponds to roughly 33 complete fan rotations, and the model ran in about 110 hours of "wall-clock" time (about 4-1/2 days). For design studies and optimization runs, for which only 1/3-Octave band spectra are required, the computational cost can be reduced by as much as 75%. Added to the one day required for model preparation, the total effort falls within our "one business week" criterion.

Table 2. Computational parameters and performance measures for the CAA model with each fan.

Parameter	Unit	FAN #1	FAN #2	FAN #3
Man-Hours (Pre/Post)	Hr/Hr	16/8	8/2	8/2
Wall Clock Time (for 4 Hz Bandwidth)	Hr	103	103	112
Wall Clock Time (for 1/3rd Octave Band)	Hr	26	26	28
Total Voxels	(M)	193	194	195
Average Spectral Error	dB	1.3	2.5	0.1

B. Comparison with Experimental Results

The results are shown in Figures 3-5 for each fan in terms of A-weighted, narrow-band Sound Pressure Levels. FAN #1 exhibited a very strong non-harmonic feature just below its BPF, and it was well-captured by the model. For FAN #2, the model introduced a non-existent tone at 352 Hz whose nature is still under investigation. The model also tended to underestimate the sound level between 200 and 1000 Hz. FAN #3 was the most highly tonal, and this was captured well by the model. However, in the measurement, only odd harmonics were prevalent; in the model, all harmonics through the 12th are strong. For FAN #1 and FAN #3, the overall shapes of the spectra are well-captured.

C. Design Alternatives

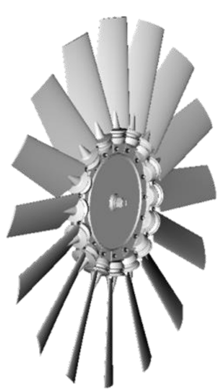
The fundamental goal of modeling fan noise within the design cycle is best exercised by examining design alternatives. The parameter space for axial fan systems is far too large to explore this way, but flow visualization indicated some key interactions.

The deep insets in the shroud used as structural mounting points create stagnations; long eddies peel away and interact with the blade tips. In the model, we closed over those indentations and smoothed out the corners and edges of the shroud. This increased the mass flow rate by ~ 8%. By reducing the fan speed until the mass flow rate was equal to the baseline design, the predicted Overall Sound Pressure Level (OASPL) was reduced by 1.7dB. The baseline shroud upstream velocity magnitude is shown in Figure 6; the smoothed shroud design in Figure 7 highlights the differences.

5. CONCLUSIONS

By combining modern CAD tools, a geometrically-flexible, time-domain CFD code (Lattice-Boltzmann method) capable of generating and tracking acoustical quantities within the flow volume, and computing resources available to OEM's, we have successfully modeled the sound radiated from large axial-flow fans *in situ* within one business week. The solutions proved true to the overall shape of the sound spectra (capturing turbulent structures and scales) and estimating Blade Passage Frequency tones to within a few decibels. There are discrepancies between the models and the measured spectra; this will require additional investigation. Likewise, other types of fans and installations need to be modeled and correlated.

Because it is often the Blade Passage tone and its harmonics that dominate the overall sound levels of off-highway equipment, the long-term goal of this effort is to predict blade passage tone levels to within 2 dB, and to track changes to those tones with design variations. More work remains to be done to achieve this challenging objective. However, we believe we are well on the way to achieving the ability to accurately estimate sound spectra from installed fans to a degree of accuracy sufficient to guide design choices. Furthermore, the technology is now available to do so within the design cycle itself, an important criterion if these practices are to be accepted in industrial settings.



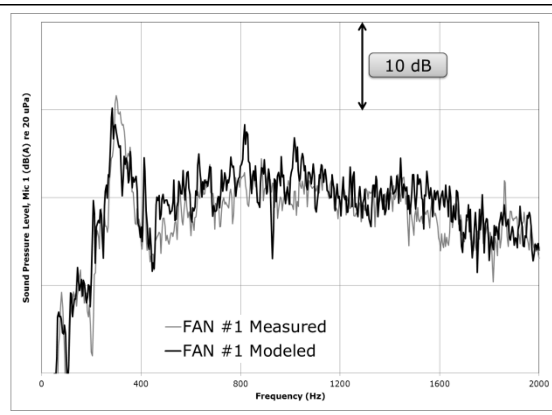
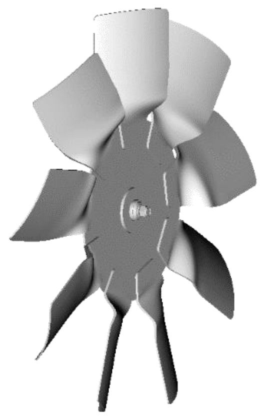


Figure 3. Mic 1 SPL vs. frequency for FAN #1.



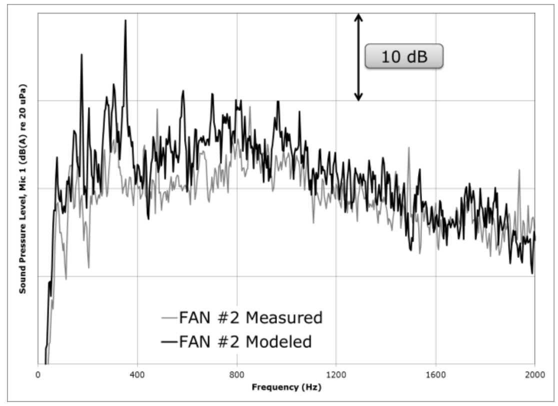
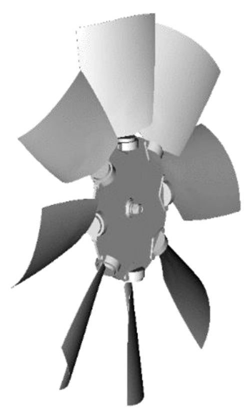


Figure 4. Mic 1 SPL vs. frequency for FAN #2.



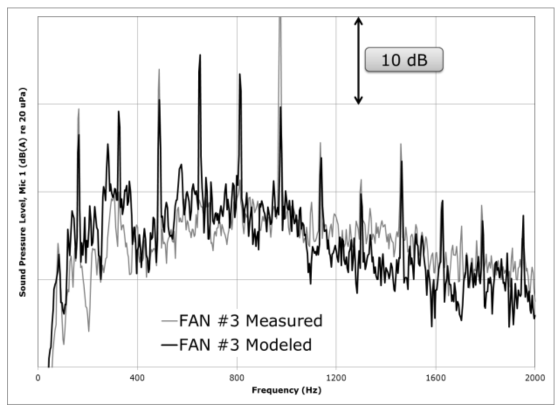


Figure 5. Mic 1 SPL vs. frequency for FAN #3.

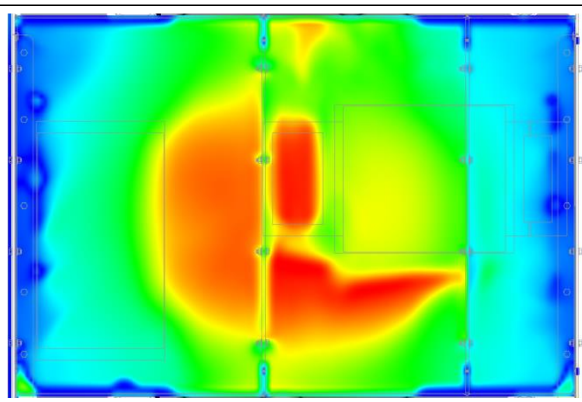


Figure 6. Flow velocity magnitude looking into the cooling package from upstream. Baseline shroud design

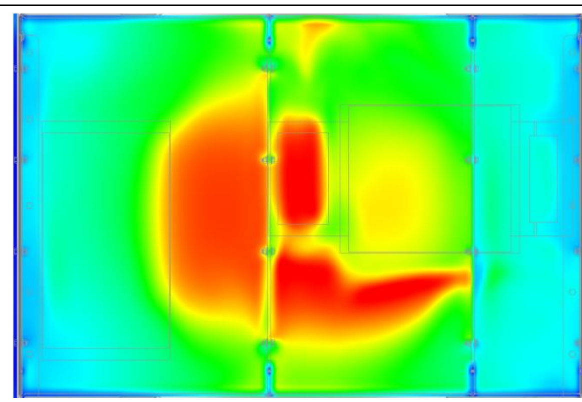


Figure 7. Flow velocity magnitude looking into the cooling package from upstream. Smoothed fan shroud.

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