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1999-01-2817

Designing Axial Flow Fan for Flow and Noise

Subrata Roy
Kettering University

Phillip Cho
Case Corporation

Fred Périé
Mecalog

Reprinted From: Noise and Vibration for Off-Highway Applications (SP-1473)

SAE The Engineering Society For Advancing Mobility Land Sea Air and Space®
INTERNATIONAL

International Off-Highway & Powerplant Congress & Exposition
Indianapolis, Indiana
September 13-15, 1999

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ISSN 0148-7191
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Printed in USA

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Designing Axial Flow Fan for Flow and Noise

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ABSTRACT

A comprehensive finite element methodology is developed to predict the compressible flow performance of a non-symmetric 7-blade axial flow fan, and to quantify the source strength and sound pressure levels at any location in the system. The acoustic and flow performances of the fan are predicted simultaneously using a computational aero-acoustic technique combining transient flow analysis and noise propagation. The calculated sound power levels compare favorably with the measured sound power data per AMCA 300-96 code.

INTRODUCTION

Due to stringent government regulations and competitive pressures, meeting noise goals are as important design criteria as meeting the flow rate and pressure head requirements. Although experimental sound measurement is quite straightforward, analyzing the origin of noise remains elusive. The challenge is to predict the flow induced noise for major noise contributors of agricultural and construction machinery such as cooling fans, exhaust muffler, and air handling system.

There are several commercially available software in the market that address the noise generated due to fluid-structure interaction. However, the choice is very limited for predicting flow induced noise and subsequent fluid-structure interaction. In this paper, a comprehensive finite element methodology is developed to predict the compressible flow performance of a non-symmetric 7-blade axial flow fan, and to quantify the source strength and sound pressure levels at any location in the system.

The acoustic and flow performances of the subject fan are predicted simultaneously using technique combining transient flow analysis and noise propagation. Corresponding flow induced noise solution is directly available at the boundary of the domain. The sound power level of the subject fan is then calculated by integrating over the hemisphere (1m radius). The calculated sound power levels compare favorably with the measured sound power data per AMCA 300-96 code.

THEORETICAL DEVELOPMENT

The computational aero-acoustics (CAA) technique utilized in this paper involves analysis of two computational components simultaneously.

a. Computational fluid dynamics (CFD) is used for analyzing three dimensional flow structures in time domain and calculating the corresponding unsteady pressure fluctuations.
b. Flow induced noise is determined in frequency domain using Fast Fourier Transform (FFT) for calculating the sound pressure level and the integrated sound power of the system based on the domain boundary unsteady pressure fluctuations.

COMPUTATIONAL FLUID DYNAMICS – As direct Navier-Stokes solutions remain impractical for industrial problems, the technical framework requires solving the compressible Navier-Stokes equation with some turbulence closure model. In this paper, the effect of turbulence is modeled with large scale eddy (LES) simulation. The computational method includes arbitrary Lagrangian Eulerian formulation (ALE), sliding non-conforming interfaces and silent radiation boundaries.

<u>ALE Formulation</u> – For accurately describing the moving boundaries, physical quantities are described at fixed points in space in the Eulerian description and following specific material particles in the Lagrangian description. In this case, each point in space has a material velocity **u** and a grid velocity **w** describing its arbitrary movement. In this context, classical conservation laws are written as:

Mass: $\partial\rho/\partial t + [(\mathbf{u}-\mathbf{w})\bullet\nabla]\rho + \rho\nabla\bullet\mathbf{u} = 0$ (1a)
Momentum: $\partial u/\partial t + [(\mathbf{u}-\mathbf{w})\bullet\nabla]\mathbf{u} - \nabla\bullet\sigma/\rho = 0$ (1b)
Energy: $\partial\rho e/\partial t + [(\mathbf{u}-\mathbf{w})\bullet\nabla]\rho e + (\rho e+p)\nabla\bullet\mathbf{u} = 0$ (1c)

Note that for **u**=**w**, equations (1a)-(1c) reduces to a Lagrangian formulation, while **w**=0 describes the Eulerian case. A single formulation is thus able to describe the evolution of physical variables in the laboratory reference frame and in a grid with any arbitrary movement (e.g. rotation). The coupling between the rotating grid and the fixed grid is done using a simple finite element interpolation scheme.

SUPG scheme – The unsteady pressure fluctuations in the computational domain are computed utilizing explicit time integration, including non-diffusive streamline upwind Petrov-Galerkin (SUPG) formulation for momentum advection and Large Eddy Simulation (LES) for turbulence.

FLOW INDUCED NOISE (FIN) ANALYSIS – Acoustic pressure signals are predicted at selected locations in the computational domain and at its boundaries. These signals are then analyzed into a frequency domain using FFT. The convergence of the fluctuating terms is first ascertained in terms of global SPL and third octave spectra, by analyzing their evolution for different time windows of the signals.

Acoustic pressure signals at node points mapping the outer hemisphere are then used to compute the source strength or sound power by integrating over the surface:

$$W=0.5 \int P^2_{rms}/(\rho c)\, dS \qquad (2)$$

where ρc is the characteristic impedance of flow medium, P_{rms} is the predicted root mean square acoustic pressure at field points, and S is the surface area that encloses the computational domain. For this case study, a hemisphere is used as the surface area ($4\pi r^2$) of integration.

The acoustic velocity or acoustic pressure distribution and the Raleigh integral equation can also be used to determine the acoustic field outside the computational domain. After quasi-stationary modes of oscillations are achieved, time domain velocity signals on the envelope of the computational domain are transformed into frequency domain using FFT. The transformed averaged velocity distribution on the envelope of the computational domain [2] and the Rayleigh Integral equation are used to determine the acoustic pressure at a field point [3].

$$P(\mathbf{r},t)=(j\omega\rho/2\pi)\exp(j\omega t) \int_S v_n(\mathbf{r}_s)\exp(-jkR)/R\, dS \qquad (3)$$

where $v_n(\mathbf{r}_s)$ is the averaged velocity distribution, exp(-jkR)/R is the Green's function that satisfies the wave equation and the far-field radiation condition. The Rayleigh Integral can strictly be applied only to infinitely extended planar surfaces, but it provides a good estimate for plane vibrating surfaces of dimensions large compared to the acoustic wavelength. Hence, care must be taken such that the dimensions of computational domain are large compared to the acoustic wavelength.

Acoustic Test Procedure – Air Movement and Control Association, Inc (AMCA) 300-96 Standard is used to measure the sound power of the axial flow fan in a wind tunnel. AMCA 300-96 standard is a reverberation room measurement technique and uses the following procedure. First, a reference sound source with known source strength is positioned at the fan location and the spatially averaged sound pressure level is measured downstream of the fan, which is in a reverberation room. Second, the spatially averaged sound pressure level of th axial flow fan is measured at the fan speed. Then, the sound power of the fan at each operating point is determined by the following relationship,

$$L_{w\ fan} = L_{p\ fan} + (L_{w\ ref} - L_{p\ ref})(4) \qquad (4)$$

where $L_{p\ fan}$ is the measured fan sound pressure level, $L_{w\ ref}$ is the known reference source sound power level, and $L_{p\ ref}$ is the measured sound pressure level of the reference source. Because the sound pressure levels are measured only downstream of the fan, a correction factor of 3 dB is added to the calculated sound power to account for the noise radiated into the upstream of the fan [5].

For this case study, the fan speed was 1900 revolutions per minute (rpm). Sound power levels calculated for three operating points were used for comparison with the predicted results from CFD.

The following table highlights the important features of the developed commercial CAA software [4].

Table 1. CAA software features

- Accurate time domain signal
- Compressible fluid flow, coupled solid interaction
- ALE-subcycling, interface sliding mesh
- k-ε, LES, non-diffusive momentum
- non reflecting boundaries

RESULTS AND DISCUSSION

CAA validated the noise levels for axial flow fan at three different flow rates, 3075, 5084 and 5685 cfm. The test was conducted following the AMCA 300-96 standard. The 3D finite element model of the 7-blade axial flow fan and the test chamber as shown in Figure 1 and 2 were built from the given Pro-Engineer [6] geometry. For the 50-2500 Hz frequency domain, a model consisting of 600,000 nodes is used to predict the sound pressure level at 1m downstream from the subject fan. The testing room set up and dimensions are according to AMCA 300-96 code. The following assumptions were made for the simulation:

- Unsteady turbulent wakes are the primary noise source.
- Fluid-structure interaction is neglected due to the high impedance mismatch (e.g. at the fan blades and

at the shroud). This assumption might not, however, be verified in the upstream channel. Nevertheless the low frequencies involved in the upstream cavities are of less interest when considering A-weighting.

- Boundary layer noise is not simulated by the LES methodology which by design does not take into account turbulent micro-scales.

Computational results are given in a tabular form in Table 2. The velocity vectors after 61.02 ms are shown in Figure 3. CFD simulation results for vorticity contours after 61.02 ms are plotted in Figure 4. The fan noise analysis solutions extracted from CAA postprocessor documented in Table 2 agree reasonably with the test data.

Table 2. Fan Noise Simulation Results

Fan RPM	Flow rates (CFM)	Test (dBA)	Analysis (dBA)	Diff (dBA)
1900	3075	98.7	96.8	1.9
1900	5084	97.4	96.7	0.7
1900	5685	97.2	100.9	-3.7

CONCLUSION

The developed CAA technique showed significant promise for flow induced noise analysis. Predicted sound power level (dBA) for the axial flow fan at three flow rates correlated favorably with the test data. Further design sensitivity study is being conducted to optimize the fan for flow and noise.

ACKNOWLEDGEMENTS

This work was a collaboration between Case Corporation, USA and Mecalog, France (Radioss-CFD). The authors thankfully acknowledge the support of Gopi Somayajula and Kasi Sethupathi of Case Corporation and Dimitri Nicolopoulos of Radioss Consulting Corporation for making this work successful.

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CONTACT

Subrata Roy was born in India where he did his undergraduate schooling in Mechanical Engineering at Jadavpur University, Calcutta. He came to the US for earning his MS and PhD in Engineering Science with specialization in Computational Fluid Dynamics. He then worked as a technical consultant for Boeing, the US Army and Fuisz. He pioneered the flow induced noise technology at Case Corporation where he served as a specialist. Currently he is a member of the faculty of Mechanical Engineering Department at Kettering University. Web: http://cfdlab.engr.utk.edu/~roy. E-mail: sroy@kettering.edu.

Fred Périé graduated from Ecole Polytechnique in France. He has been a developer of simulation techniques for 15 years. He joined Mecalog in 87 where he developed RADIOSS CFD focusing on algorithms dedicated to fluid-structure problems and to aeroacoustics.

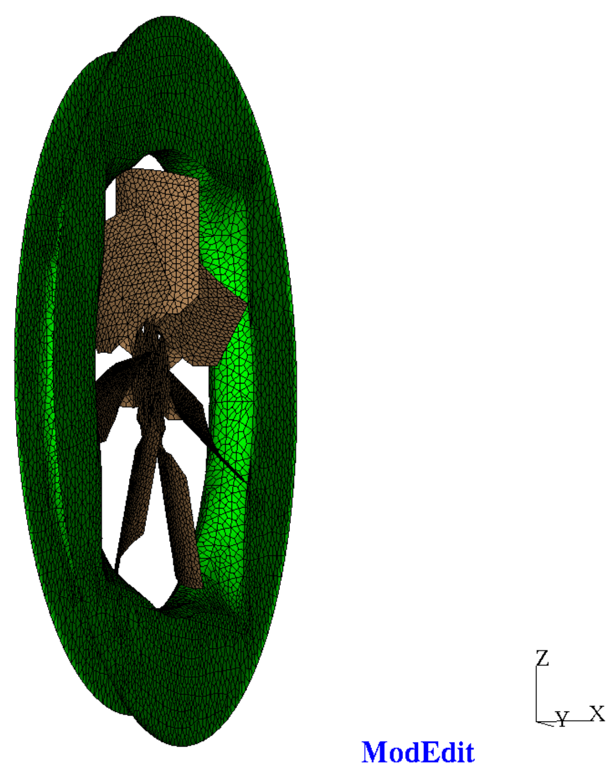

Figure 1. Fan blade and shroud mesh.

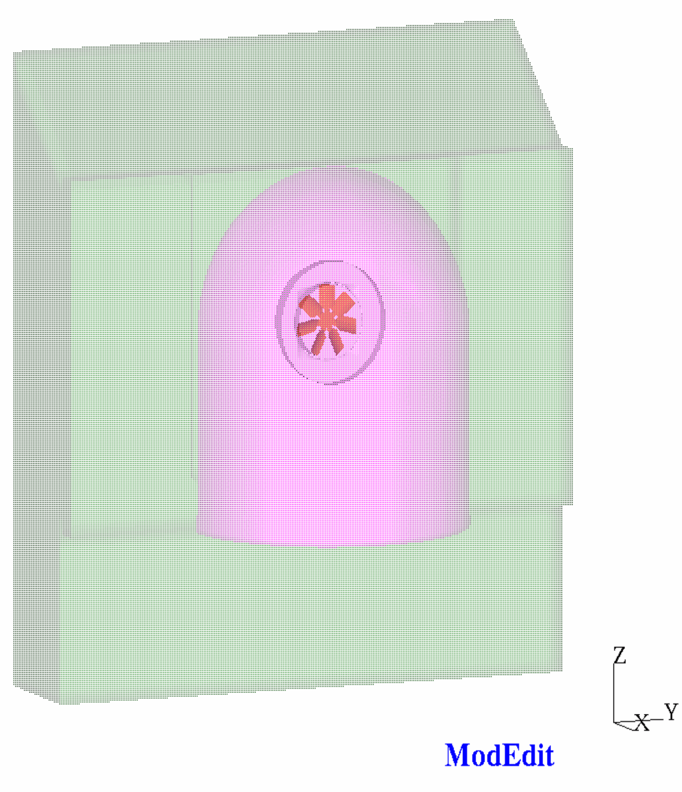

Figure 2. Mesh details for the test chamber.

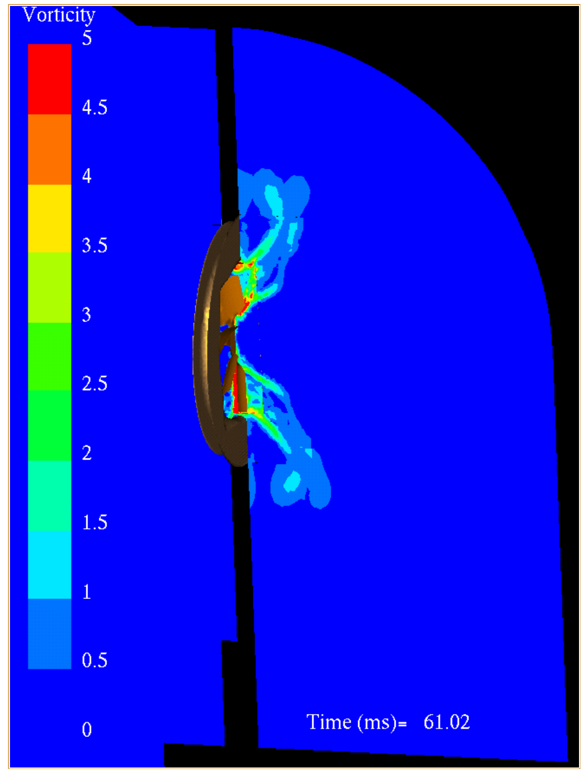

Figure 4. Vorticity contours after 61.02 ms.

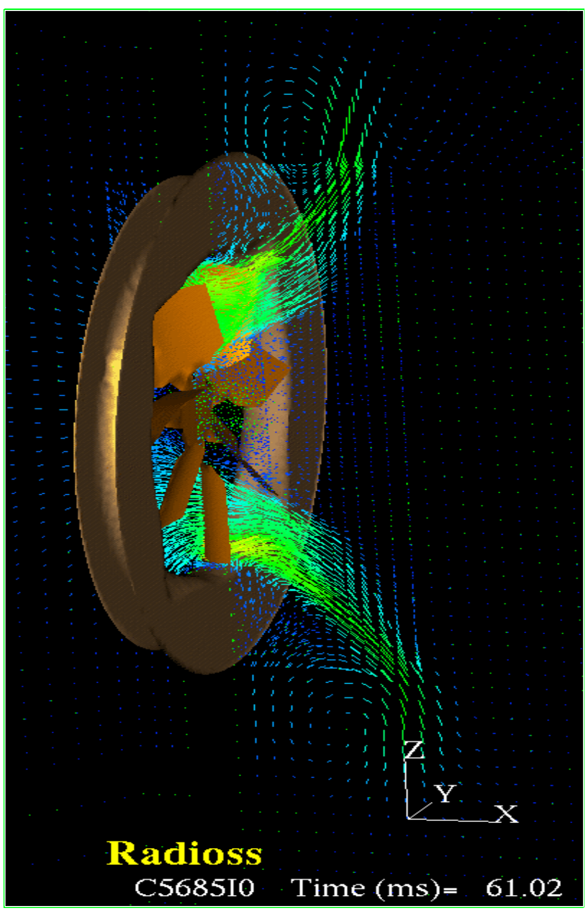

Figure 3. Velocity vectors after 61.02 ms.