inter.noise 2000
The 29th International Congress and Exhibition on Noise Control Engineering
27-30 August 2000, Nice, FRANCE

I-INCE Classification: 7.6

NOISE PREDICTION FOR ROTATING MACHINERY USING CAA

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Keywords:
COMPUTATIONAL AEROACOUSTICS, ROTATING MACHINERY, INDUSTRIAL NOISE, LES AND ALE

ABSTRACT
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 aerospace, automotive, and heavy industry machinery such as cooling fans, exhaust muffler, and air handling system. In this paper, a comprehensive finite element methodology is developed to predict the compressible flow performance of rotating machinery, and to quantify the source strength and sound pressure levels at any location in the system. The acoustic and flow performances are predicted simultaneously using a computational aero-acoustic technique combining transient flow analysis and noise propagation. The calculated sound power is validated against the measured sound power data.

1 - INTRODUCTION

The prediction of sound induced by turbulent flow has been studied extensively since the pioneering acoustic analogy of Lighthill [1]. This analogy is limited to knowing the source terms a *priori* requiring prohibitively extensive experimental data collection. It is, however, numerically possible to evaluate any particular *exact* source term, within the limits of computational accuracy, and predict the radiated acoustic field. A reasonable amount of theoretical and computational research is done in the area of vortex structure and related sound generation [2-8]. The recent advances in high performance digital computers have allowed some investigators to utilize direct numerical simulation (DNS) of the Navier-Stokes equations, whereby the entire flow field is resolved, including the small scales associated with turbulent fluctuations [2].

While DNS simulations offer the potential to provide highly accurate computational predictions of flow-generated vortices and acoustic pressure, the application of DNS is limited to simple flows with relatively low Reynolds number till date as the procedure is extremely computational intensive. Several other alternative research efforts include both singular and non-linear perturbation approaches [3], [4], lattice gas analysis [5] and the low Mach number stochastic model for expansion about incompressible flow (EIF) [6]. Practical implementation of noise calculation has also been reported for fan noise prediction [7], [8]. There are several commercially available software in the market that address the issue of noise. 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 rotating machinery, and to quantify the source strength and sound pressure levels at any location in the system. The acoustic and flow performances of the subject stator-rotor 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 (1.1m radius). The calculated sound power levels compare favorably with the measured sound power data per AMCA 300-96 code.

2 - THEORETICAL DEVELOPMENT

The computational aero-acoustics (CAA) technique utilized in this paper involves analysis of two computational components simultaneously. First, computational fluid dynamics (CFD) is used for analyzing three dimensional flow structures in time domain and calculating the corresponding unsteady pressure fluctuations. Flow induced noise (FIN) is then 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.

For accurately describing the moving boundaries in a stator-rotor system, 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 $\mathbf{u}$ and a grid velocity $\mathbf{w}$ describing its arbitrary movement. The corresponding ALE formulation of unsteady classical conservation laws are written as:

Mass:

$$\partial\rho/\partial t + [(\mathbf{u}-\mathbf{w})\cdot\nabla]\,\rho + \rho\nabla\cdot\mathbf{u} = 0 \tag{1a}$$

Momentum:

$$\partial\mathbf{u}/\partial t + [(\mathbf{u}-\mathbf{w})\cdot\nabla]\,\mathbf{u} - \nabla\cdot\sigma/\rho = 0 \tag{1b}$$

Energy:

$$\partial\rho e/\partial t + [(\mathbf{u}-\mathbf{w})\cdot\nabla]\,\rho e + (\rho e + p)\,\nabla\cdot\mathbf{u} = 0 \tag{1c}$$

Here, the density is denoted as ρ, e is the mass specific total energy, p is the static pressure, and σ is the stress tensor.

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. In the current CAA procedure, LES is used for capturing the macro turbulence structure, while for micro-turbulence Smagorinsky's sub-grid scale dissipative model is utilized for computational efficiency.

The CFD simulation predicts acoustic pressure signals at selected locations in the computational domain and at its boundaries. These signals are then analyzed into a frequency domain using fast Fourier transform (FFT). The convergence of the fluctuating terms is first ascertained in terms of global sound pressure level (SPL) and third octave spectra, by analyzing their evolution for different time windows of the signals.

The transformed averaged velocity distribution on the envelope of the computational domain and the Rayleigh integral equation are used to determine the acoustic pressure at a field point [9]. The 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 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:

$$W = 0.5\int P_{rms}^2/\,(\rho c)\,\mathrm{dS} \tag{2}$$

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. Table 1 highlights the important features of the commercial CAA software [10].

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

Table 1: CAA software features.

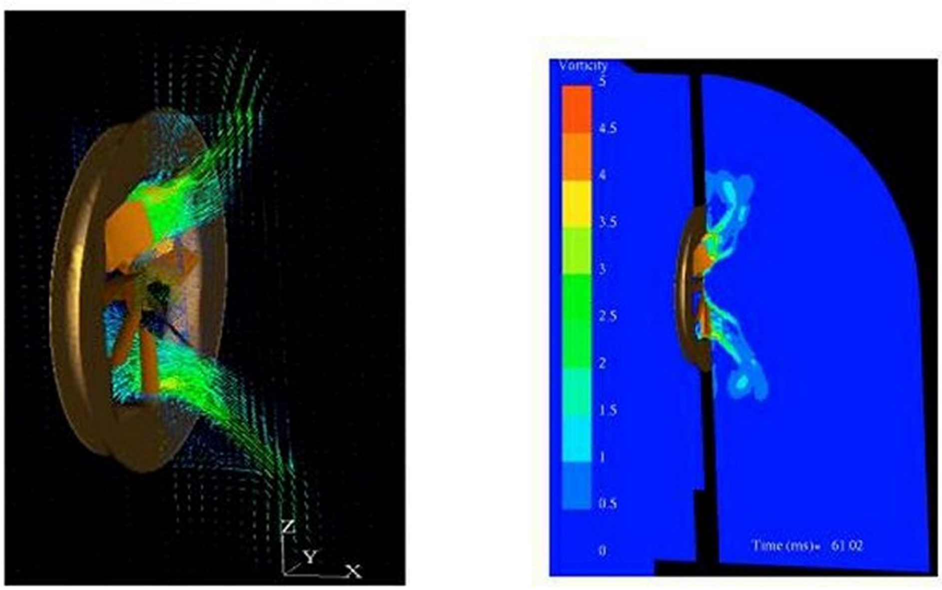
3 - RESULTS AND DISCUSSION

CAA validated the noise levels for axial flow fan at three different flow rates, 3075, 5084 and 5685 cfm. The testing room set up and dimensions are according to AMCA 300-96 code. The 3D finite element model of the 7-blade axial flow fan and the test chamber was built from the given Pro-Engineer geometry. 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. 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. A fine mesh of 200,000 hexahedral finite elements is applied near the rotating fan blades to capture the boundary layer detail of the CFD solution resolution for this frequency range.

Computational results after 61.02ms is documented here. The velocity vectors and the vorticity contours shown in Figures 1a and 1b respectively, confirm the fan exit plane recirculation indicating the flow induced noise source. Fourier transformed sound pressure levels (SPL) at 1.1 m cross-section downstream of the fan are plotted in the frequency domain, Figure 2a. Corresponding fan noise analysis solutions extracted from CAA postprocessor documented in Table 2 agree *reasonably* with the test data. These validation results, plotted in bar chart Figure 2b, show that the error is reasonably small for low flow rates (3075 and 5084 cfm) where the computational mesh successfully captures the flow vortices and corresponding pressure fluctuations for these flow rates. However, the error grows as the flow rate increases to 5685 cfm. This due to the fact that the mesh is not fine enough to resolve the micro-turbulence structure even using the SGS model. A better mesh may be required.

Fan Speed (RPM)	Flow rates (CFM)	Test (dBA)	Analysis (dBA)	Difference (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

Table 2: Fan noise simulation results.

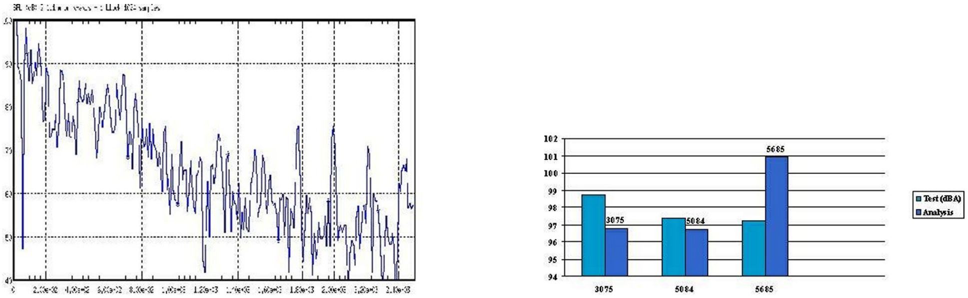


(a): Velocity vectors. **(b):** Vorticity contours.

Figure 1: CFD simulation results after 61.02 ms.

4 - CONCLUSIONS

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 necessary to optimize the fan for flow and noise. Although



(a): SPL. **(b):** Sound power.

Figure 2: CAA analysis results: (a) sound pressure levels in third octave spectra at 1.1 m downstream of the fan; (b) sound power validation with test data (x-axis cfm, y-axis dBA).

current results confirm CAA application to fan, the methodology is general enough to be extended to any rotating machinery. Proving that, however, requires additional computational effort.

ACKNOWLEDGEMENTS

The authors thankfully acknowledge the support of Gopi Somayajula of CNH Corporation and Dimitri Nicolopoulos of MCube Corporation for making this work successful.

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