Numerical Simulation of Aero-Acoustic Noise Prediction of Radiator fan in free field

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Abstract: Axial fan is used inside radiator for heat dissipation of coolant which in turns cools engine and also provide air circulation through generator. This axial fan is major source of noise generation inside the generator. Now a day's Gensets have to meet noise limits in different parts of the world, such as Central Pollution Control Board of India's CPCB-II norms in India or European Commission's CE Regulation (2000/14/EC). For developing gensets to meet such noise limits, it is important to develop more accurate noise levels prediction tools before manufacturing gensets. The simulation work discussed in this manuscript is carried out at Cummins India Ltd.'s Technology Center at Pune and experimental work and technology algorithm development work is done at Cummins Power Generation's head quarter in Minneapolis, MN, USA.. To address the experimental challenges, a numerical analysis of noise prediction by using Ffowcs Williams and Hawkings method was performed. In this study aero-acoustic noise prediction of radiator fan is done by using suitable computational fluid dynamics tool like ANSYS. Steady state simulation shows acceptable results of tonal noise at blade pass frequency and its harmonics at radial receiver location. This shows that current methodology is suitable for given problem and results are acceptable to initialize transient state. Validation of numerical results of Sound Pressure Level (dBA) value at certain distance from fan is proposed with experimental testing results.

Keywords: Axial fan, Aero-acoustics, Tonal noise, Far field noise.

Nomenclature

BPF	Blade pass Frequency, Hz
c	Sound speed, m/s
f	Frequency, Hz
N	Rotating speed, RPM
SPL	Sound Pressure Level, dB
PSD	Power spectral density, Watt/ Hz
OASPL	Overall Sound Pressure Level, dB
Prms	Root mean square pressure, Pa
Pref	Reference pressure = $2*10^{-5}$ Pa
	Components of flow velocity v(x, t)
	Specific viscosity, m^2/s
	Shear stress, N/m^2
	Light hill stress tensor
	Kronecker delta
	Friction velocity, m/s
	First cell height, m

I. INTRODUCTION

A typical electrical power generator set uses axial fan to pass air through radiator core made up of tube and fin structure containing liquid coolant, which is cooled by convection phenomenon. This radiator fan is major source of airborne noise and structure borne noise generation inside the generator. Noise generated by axial fan depends on shape of blade [1, 2]. Air borne noise is created by axial fan due to vortex shedding [3, 4], self generated noise (turbulent or laminar boundary layers, boundary layer separation) [5], turbulent inflow [6], blade passing noise, and due to separation at trailing edges. First step to set forward for this is to calculate accurately noise generated by each source region and their propagation path to receiver location. To avoid costly experimental testing, a numerical analysis of noise propagation through radiator fan is proposed. Researchers have proposed many methods for analysis of sound propagated through air. Ffowcs Williams and Hawkings (FW-H) method is used to track sound transmission from source to receiver location [7] called as acoustic analogy. Computational aero-acoustic (CAA) method is more precise over acoustic analogy modeling in which entire domain of interest containing source along with receiver location is resolved [8, 9]. But FW-H method is more appropriate since it requires only source simulation therefore less computational cost. Acoustic analogy method separates the field of calculation into an aerodynamic part and aero-acoustic part. Pressure fluctuations created from source are calculated by the resolution of Navier-Stokes equations. FW-H model is used to calculate propagation of these pressure fluctuations to far-field. The advantage of this methodology is that acoustic pressure fluctuations are directly connected to the fluctuations of the instantaneous aerodynamic parameters. Thus, the unsteady character of the various parameters is entirely restored. In this research numerical technique is studied to obtain dominating source region of fan, noise propagation from source to sink, overall sound pressure level at receiver location. Validation of this numerical analysis results at receiver locations are proposed with experimental test results. Experimental testing for given radiator fan is performed at Cummins' Acoustical Technology Centre (ATC) Fridley, Minneapolis, MN, USA.

II. AERO-ACOUSTIC SIMULATION

Lighthill [10] in 1952, developed a theory, which determines that sound radiated by turbulent flow in a fluid without solid boundaries has quadrupole characteristics. Lighthill's theory was extended by Curle [11] shortly afterwards to a flow where stationary solid objects are present. According to Curle, a sound wave radiated by a flow in the presence of a solid object is the sum of the Lighthill's quadrupole sound and sound wave generated due to flow over acoustic source surfaces which has dipole characteristics. Curle also proved that the strength of this dipole sources is proportional to the total force per unit area on the surface.

Ffowcs Williams and Hawkings [12] developed more general than Curles equation and describes flow around a solid object, which moves at an arbitrary speed. Ffowcs Williams and Hawkings (FW-H) equation contains a monopole term, which depends on the velocity of the object with respect to a stationary observer. Also main conclusion of Curle's theory that the sound radiated from stationary object has dipole characterics remains unchanged in the Ffowcs Williams and Hawkings theory, and for an immoveable object the FW-H equation reduces to Curles equation.

A. GOVERNING EQUATION

The Lighthill's aeroacoustics analogy mainly introduces a quadrupole-type acoustic source term associated with turbulence stress, which allows the calculations of acoustic source data in a considered control volume inside a CFD computation domain. Lighthill's equation can be derived from the mass and momentum Conservation Equations. Here density perturbations (ρ') with respect to the ambient condition (ρ_0): i.e., $\rho' = \rho - \rho_0$ are considered. Lighthill's equation in the form of an inhomogeneous wave equation is written as [10];

$$\frac{\partial^2 \rho'}{\partial t^2} - c^2 \nabla^2 \rho' = \frac{\partial^2 T_{ij}}{\partial x_i \partial x_j}$$

Where c is the speed of sound and T_{ij} is the Lighthill stress tensor defined as [10];

$$T_{ij} = \rho v_i v_j + \sigma_{ij} - c^2 \rho' \delta_{ij}$$

The Lighthill's aeroacoustic analogy describes sound generated by the turbulence stress of fluid flow without considering any foreign bodies submerged in a fluid medium. Using free-space Green's function, solution to the Lighthill's equation is obtained i.e. [10],

$$\rho'(x,t) = \frac{1}{4\pi c^2} \frac{\partial^2}{\partial x_i \partial x_j} \iiint \frac{T_{ij}\left(y, t - \frac{|x-y|}{c}\right)}{|x-y|} d^3y$$

Where **x** and **y** are the vectors coordinates pointing receiver and source, respectively. Time *t* is defined at the receiver location where as $\tau = (t - |\mathbf{x} - \mathbf{y}|/c)$ represents the time, at the source location. Using Green's theorem we can construct combined integral equation of the effect of sources, propagation, boundary conditions and initial conditions in a simple formula. In CFD Fluent when a system of interest includes an arbitrarily moving solid body submerged in a fluid medium, the FfowcsWilliams-Hawkings (FW-H) aero-acoustic analogy is the most commonly used for the aeroacoustic analysis of system. Simplified FW-H equation can be obtained by using a free-space Green's function: i.e. [12];

$$4\pi\rho(x,t) = \frac{\partial^2}{\partial x_i \partial x_j} \iiint \frac{T_{ij}}{r|1-M_r|} d^3y - \frac{\partial}{\partial x_i} \iint \frac{\rho v_i (v_i - v_j) + \sigma_{ij}}{|1-M_r|} n_j d^2y + \frac{\partial}{\partial t} \iint \frac{\rho v_i (v_i - v_j) + \rho_0 u_i}{|1-M_r|} n_i d^2y$$

Where n_i is the unit normal vector to the source surface, M_r is the Mach number of the source surface velocity component along the direction of the radiation vector, r: i.e., $M_r = Miri$. In Eq., the first volume integral, that is a quadrupole-type, shows sound source due to the turbulence stresses inside the permeable source region. Second and third surface integrals i.e. dipole- and monopole-types shows the force from the control surface to the fluid medium and the mass flow through the permeable surface, respectively. In the current aero-acoustic analyses, the source surfaces are defined only over fan blades without considering any permeable surfaces representing certain control volume. Therefore, the acoustic source does not consist of all the aforementioned source types. The acoustic pressures at predefined receiver locations are evaluated by Equation once the source data (i.e., fluid flow velocities and pressures) are obtained from transient CFD computations.

B. ACOUSTIC GRID GENERATION

Fluid volume is extracted for required domain of analysis, figure 1 shows schematic of fluid volume domain.

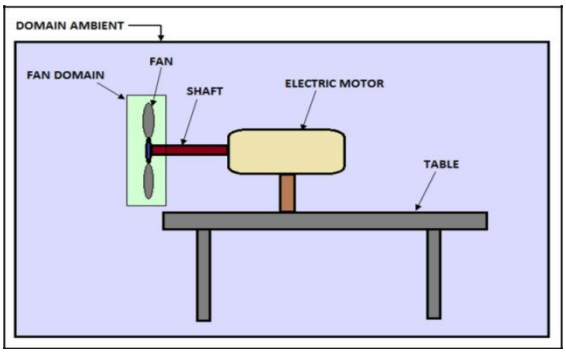

Fig.1:Schematic Fluid Volume Extraction In Ansys Showing Fan Domain And Ambient Domain

In aeroacoustic analysis of fan, mesh sizing of domain is maintained considering frequency range of interest also blade pass frequency, which creates tonal noise when tip of fan blade profile cuts the air at particular location repeatedly in one revolution. To capture all effects along with boundary layer phenomenon accurately over fan blade profile, fine grid around all critical surfaces of fan is ensured. This is done by considering first cell height y_{wall} to capture Boundary layer phenomenon, which can be find out using y^+ dimensionless number;

$$y_{wall} = \frac{\nu y^+}{u_\tau}$$

In aero-acoustic analysis discretization should satisfy both the flow and acoustic CFL condition, which is a limitation on the ratio between the time and length scale.

$$Flow\ CFL = \frac{v_i\,\Delta t}{\Delta x_i} \leq 1$$

$$Acoustic\ CFL = \frac{c\,\Delta t}{\Delta x_i} \leq 1$$

Where shows time step sizing and gives minimum length scale. Generated grid over blade profile for given fan is shown in fig2.

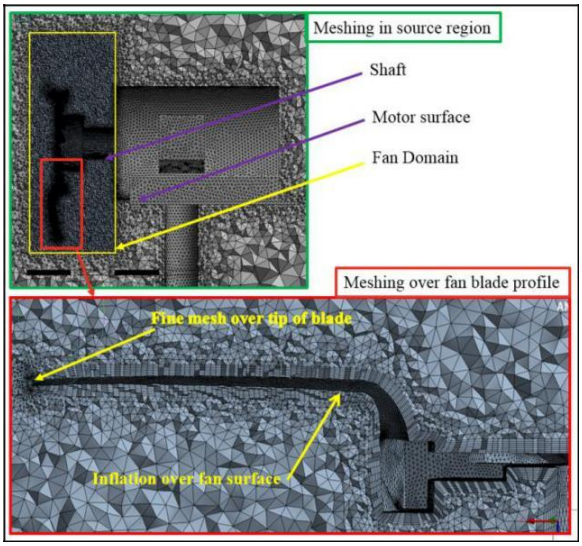

Fig. 2: Mesh Generated In Fan Domain
And Ambient Domain.

C. TURBULENCE MODELING

k-ε model is used for steady flow calculations along with acoustic FW-H model. Delayed detached eddy simulation (DDES) is used for unsteady flow calculations. Detached Eddy Simulation (DES) is hybrid formulation that switches between Reynolds average Navier Stokes (RANS) and Large Eddy Simulation (LES) methods based on the grid resolution provided. By this formulation, the wall boundary layers are entirely covered by the RANS model and the free shear flows away from walls are computed in LES mode. The main purpose of this model is to run RANS mode attached flow regions, and to switch to LES mode in detached regions away from the walls. This switching process is activated by grid limiter which depends on grid resolution. This model allows user to reduce high computing costs of covering the wall boundary layers in LES mode. But this model can leads to Grid-Induced separation (GIS) due to activation of DES limiter inside attached boundary layers by grid refinement. Hence modification to DES model is done such that slow transaction will occur while shifting to LES from RANS. To achieve this modification of numerical scheme is done such that it will not suppress formulation of resolved turbulence as the flow separates from the wall.

Atmospheric pressure and temperature are specified at the inlet boundary and outlet boundary. There is no requirement to provide NRBC i.e. non reflective boundary condition to this since FW-H model does not account for reflection. No-slip conditions are used over fan wall surfaces. The sliding mesh technique was applied to the interfaces in order to allow the unsteady interactions between fan domain and ambient domain.

In the discretization, the time-dependent term is discretized by the second-order implicit scheme which provides higher accuracy for rotating or swirling flows. Using Green-gauss node based as a Gradient will increase the accuracy and give better interpolation between two adjacent grids. The momentum term is discretized by the second-order bounded central-differencing scheme. The pressure-velocity coupling is calculated through SIMPLEC algorithm. It allows higher Under Relaxation Factor (URF) to use and gives faster convergence than we get from default SIMPLE method. Rest all the methods are kept second order to get the better results and interpolation between the adjacent grids. Under Relaxation Factor (URF) given is 1 to all the parameters. Convergence criterion on the scaled residuals and sum of the fluxes are maintained less than 10^{-4} for a given variable in all the cells. For unsteady simulation, time step is chosen based on maximum frequency (f_{max}) of interest to be resolved i.e. $\Delta t = 1/ (f_{max} *10) = 0.00005$ s, which is smaller than time step size depending on rotating speed of fan i.e. $\Delta t = 60/360N$ (where N denotes rotating speed of fan).

II. RESULTS AND DISCUSSION

When the solution is converged then we have the pressure fluctuation values (Pressure-Time History Data) at source surfaces. These source values are propagated at the receiver end using FW-H model. Most of the sound sources produce a very disordered and random waveform of pressure versus time. Such a wave has no periodic component, but by Fourier analysis resulting waveform may be represented as a collection of waves of all frequencies.

$$F(x) = \frac{a_0}{2} + \sum_{n=1}^{\infty} [a_n \cos(nx) + b_n \sin(nx)]$$

Here, F(x) is function of time (x) and a_0 stands for pressure amplitude. The received time content signal will be broken up into sine waves such as to describe its frequency content. After obtaining pressure amplitude versus frequency data, calculations of various results like Power spectral density (PSD), Sound Pressure Level (SPL), Overall Sound Pressure Level (OASPL) etc. can be done.

$$\text{Here PSD} = \frac{P_{rms}^2}{\rho_0 C_0^2} \text{Watt/Hz},$$

$$\text{SPL} = 10 log_{10}(PSD_i / P_{ref}^2) \text{ (dB)}.$$

Using Fast Fourier Transformation (FFT) qualitative analysis such as auto pruning, windowing, truncation and normalization of FFT plots at all receiver location can be done. In steady state analysis fan does not rotate for 1 second of time which shows downward shift of results giving normalized values. This normalization effect is added to obtained original results such as;

$SPL_{WN} = SPL_{W/O\ N} - 10\log_{10}(\Delta f)$.

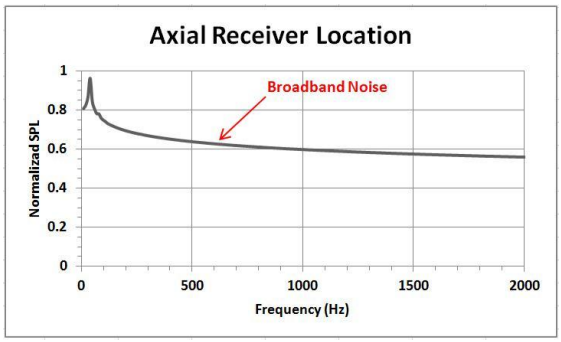

Fig. 3: Normalized Spl Vs Frequency
Fft Plot At Axial Receiver Location

In both the axial direction receiver locations as the effect of blade pass frequency is not captured in steady state analysis, only declining continuous broadband noise spectrum is seen as shown in figure 3.

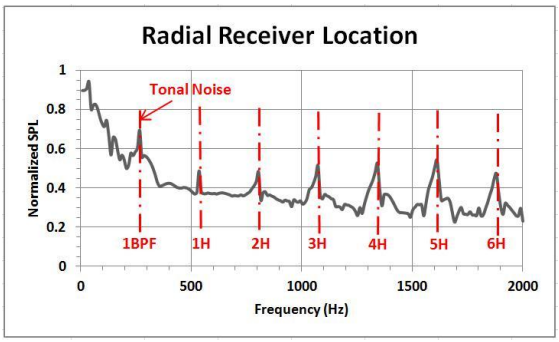

Fig. 4: Normalized Spl Vs Frequency
Fft Plot At Radial Receiver Location

At radial receiver locations we are able to see picks at blade passing frequency (BPF) and its harmonics as shown in fig. 4. This means that effect blade passing noise at radial receiver locations is captured in steady state analysis. Source strength is not captured in steady state analysis since fan is not actually rotating hence steady state plots are only for quick check out, that where we are getting BPF tonal noise. These tonal noises are seen at first BPF and its harmonics, resolution of which is important, since they dominate the overall sound pressure level. This results show that simulation methodology used for given fan is suitable and we can use this steady state simulation results to initialize transient simulation.

IV. CONCLUSION

Steady state simulation results are analyzed at required receiver locations. At Axial receiver locations effect of tonal noise or blade passing noise is not captured in steady state analysis, it only shows broad-band noise spectrum. Radial receiver location shows tonal noise at BPF and its harmonics, which dominates overall sound pressure level. Steady state results are suitable to initialize transient simulation for obtaining source strength data and sound propagation in domain, this will help to reduce simulation time and cost.

ACKNOWLEDGEMENT

It gives me immense gratitude to acknowledge the support and help of my manager at Cummins Power Generation Technologies **Mr. Abhishek Kakde** for his esteemed guidance and support, especially through difficult times. I am grateful for getting an opportunity to work with **Dr. Shashikant More** whose valuable comments and suggestions broadened my vision and guided me to succeed in this work. I would like to offer my special thanks to **Mr. Himanshu Agrawal** for sharing his valuable knowledge during my work at Cummins and enthusiastic support while exploring new tasks. I also place on record, my sense of gratitude to one and all who, directly or indirectly, have lent their helping hand in this venture. Lastly, I would like to thank my parents for their blessings and strength that helped me a lot to complete this task successfully.

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