Fort Lauderdale, Florida
NOISE-CON 2014
2014 September 8-10

Total Noise Analysis for Containerized Power Generator

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ABSTRACT

In order to meet stringent noise requirements from customers, containerized power generators require design assessment of total radiated noise. Total noise level is a superposition of contributions from all noise sources: exhaust, engine, generator and cooling fan; with attenuation from silencers, acoustic insulation materials and the design of airflow openings. Simulation analysis tools can be a valuable resource in reducing design turnaround time.

The objectives of this study were to validate a methodology to predict the total noise for a containerized power generator by coupling a Lattice Boltzmann Method (LBM) based flow and acoustic solver with a Statistical Energy Analysis (SEA) structural acoustic model. Cooling fan noise, engine/generator noise, exhaust noise and acoustic material absorption were modeled within the LBM approach. The SEA approach handled noise transmission through walls. Validation of the methodology was performed on a 1500 kilowatt containerized generator, with results within 1 decibel of overall test levels at 7 meters from the walls. Although total noise prediction was the main objective, the tradeoff between cooling airflow and noise was also considered in assessing the impact of design variations.

1. INTRODUCTION

Noise performance of portable power generation equipment is an important design element, especially as operational concerns must be met in a commercial environment. For containerized power generators intended for the construction and entertainment industries, reduced noise can be a significant factor when selecting a generator. Reliability is perhaps an even more critical concern, as noise control features must not result in an operational environment that compromises dependable life. Manufacturers need to be able to provide very quiet operation with adequate cooling for the generator system.

The noise radiated from containerized power generators originates from a number of sources: engine surfaces, exhaust, intake, cooling fan and generator[1]. Noise control treatments for power generation systems are designed for good performance in both sound absorption and for sound transmission[2]. Simulation of airflow and noise performance can be a valuable tool in bringing new designs of power generation equipment to market, potentially identifying both cooling and noise issues of a given design to permit tradeoffs before building expensive prototypes. While traditional noise control design of container enclosures addresses only acoustic performance, the direct interaction of cooling air openings with both flow and noise performance suggests that consideration of both simultaneously could be a valuable resource in reducing design turnaround time. Flow simulation is commonly addressed using computational fluid dynamics (CFD) tools, which may be steady or transient, incompressible or compressible, depending on the needs of the system being predicted. Simulation of noise transmission may use the techniques of finite element analysis, boundary element analysis or statistical energy analysis (SEA), with the choice of tools dependent mainly on the frequency range and size of the system.

This paper describes the application of a CFD solver, based on the Lattice Boltzmann Method (LBM), to simulate both transient flow and acoustics simultaneously in and around a 1500 kW containerized power generator. All important noise sources contributing to radiated sound at 7 meters distance from the container sides were incorporated in the analysis. While noise of the rotating cooling fan was simulated directly in LBM, other sources of airborne noise were produced by virtual speakers in the LBM model. Propagation to the far field was carried out with an acoustic analogy approach. Noise transmission through acoustically treated and untreated walls of the container was simulated with an SEA modeling tool. Other available noise performance measures, such as acoustic absorption, sound transmission loss and silencer insertion loss were applied at appropriate stages in the modeling. Results from the total noise analysis are compared to experimental noise measurements on early production units, verifying the predictive capability.

2. CONTAINERIZED POWER GENERATOR

A. Model Description

Figure 1 shows a CAD model representation of the 1500 kW power generator used in this study. The longest dimension of the trailer is about 12 meters. Ventilation for the power generator is performed by an engine-driven fan pulling in air through the louvers located on its left and right sides. Cooling air flow exits through the grill located at the rear roof.

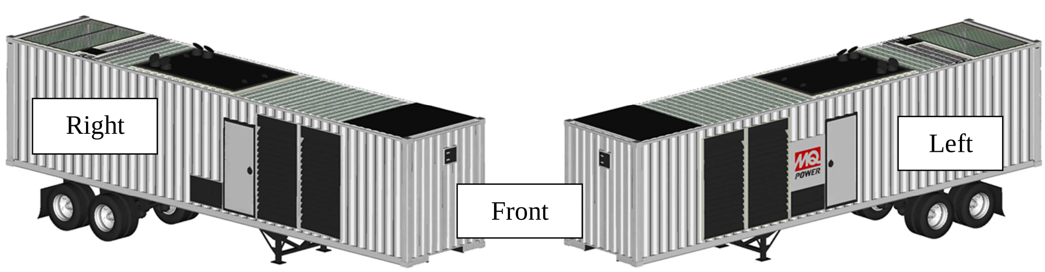

Figure 1: 1500 kW power generator enclosure, showing right, front, left, and top sides, with cooling inlet louvers on sides and exit grill on rear ceiling

Figure 2 shows the inside view of the model. The generator, with a sixteen cylinder diesel power source, is located centrally between acoustically-treated louvers (left and right), serving as ventilation inlets. Engine exhaust flow is routed from turbocharger outlets into a large, ceiling-mounted silencer with four outlets. Trailer walls and ceiling are partially covered by acoustic panels serving as noise absorbers and insulators. The cooling package is comprised of a pusher fan and two cores.

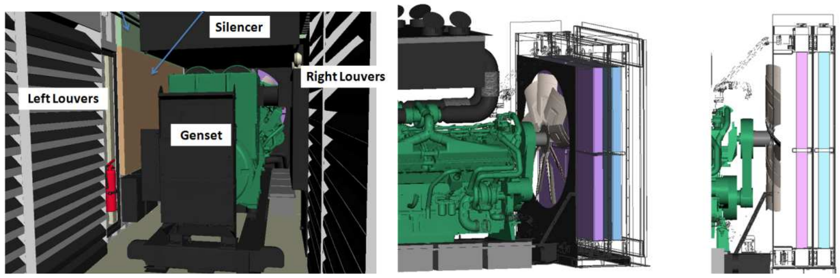

Figure 2: 1500 kW power generator, showing interior, generator, exhaust silencer and cooling package

B. Noise Sources and Transmission Paths

To predict the total noise of the power generator, it is important to incorporate all of the contributing noise sources and paths. First of all, there is noise radiated from the engine and generator surfaces, to which is added the noise generated by the rotating fan pushing air to cooling cores. Intake air filters are just above the engine-generator coupling, contributing additional engine noise. The combined noise level of the engine, generator and fan is attenuated by absorbing panels, partially transmitted to the outside air through the inlet louvers and exit grill, and also through wall transmission with further blocking by the acoustic insulation panels. Exhaust system openings radiate additional noise outside.

The desire was to predict fan noise directly with the LBM solver, much as described by Pérot et al[3], providing realistic source location and sound directivity with flow. With the available noise performance test data for the engine and unsilenced exhaust from the manufacturer[4], it was possible to calibrate radiated sound power for these sources using an array of virtual speakers for the engine and transient mass flow for exhaust. This made it possible to have the cooling fan noise, engine/generator noise, exhaust noise and acoustic material absorption[5] modeled within the LBM approach, providing airborne transmission through cooling and exhaust openings, while the SEA approach handled noise transmission through walls. Steps detailing this approach to total noise simulation are explained in the next section.

3. TOTAL NOISE ANALYSIS

A. Computational Approach

The computational approach to predicting total noise of the containerized power generator can be described in 3 steps, as shown in Figure 3.

Figure 3: Steps to predicting total noise

Step 1 is an acoustic (no flow) simulation, in generator noise test condition, to calibrate the engine sound power. Step 2 consists of adding the enclosure with absorption properties of the louvers and acoustic panels. In this LBM simulation, engine sound is generated by the calibrated speakers and fan noise is produced directly by the rotating blades, now with all *in situ* flow obstructions. Fluctuating quantities are captured in virtual measurement surfaces outside the openings, allowing post-CFD computation of the far field noise at 7 meters from the container wall in step 3, using an integral extrapolation method based on the Ffowcs Williams and Hawkings (FW-H) acoustic analogy approach[6]. Contributions from exhaust and wall transmission may also be included in step 3.

The following subsections explain in detail the engine and exhaust calibration process, the acoustic absorption modeling and the wall transmission modeling with SEA. For further details on the fan noise simulation, including rotating geometry, please refer to reference[3].

1. Engine and exhaust calibration

The calibration process estimates boundary conditions for the engine and the exhaust that will deliver the sound power spectrum provided by the generator vendor in free field conditions[4]. The chosen approach for engine noise was to place virtual speakers on various surfaces of the engine and to adjust the speaker velocity as a transient boundary condition to match the engine radiated sound power data. Figure 4 shows the geometry of generator in the engine test configuration and the results of the calibration compared to vendor noise data[4].

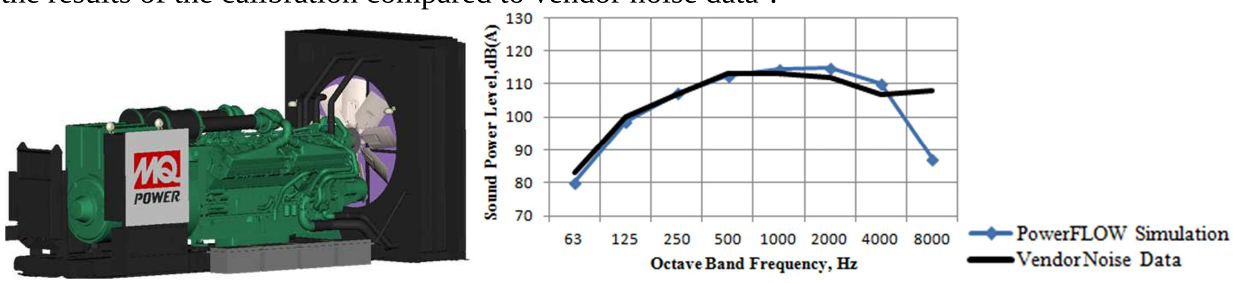

Figure 4: Model used for the engine calibration (left), Sound power level calibration spectrum (right)

In the sound test configuration, the generator is placed on a reflecting hard surface, with isolation mounts to reduce structureborne noise, exhaust routed away to large mufflers and cooling handled remotely, that is, the fan noise is not present. Radiated sound power was measured in test and simulation in accordance with ISO 3744 and ISO 8528-10, as applicable. After the simulation calibration process, there was good agreement with engine noise except at high frequency. With the overall sound power level only 1 dB low in simulation, the discrepancy at high frequency on the final results was minor. While very high frequencies can be simulated in LBM using smaller fluid cells, or voxels, doing so for such a large system would result in excessive turnaround time. As such, the speaker signal corresponding to this sound power level was used as boundary condition for the engine noise contribution in the final prediction simulation.

The exhaust calibration was slightly different than the engine calibration. Instead of using virtual speakers as boundary conditions, the exhaust mean mass flow was supplemented by transient mass flow as a boundary condition, calibrated to match the test sound data for the exhaust alone, obtained per ISO 6798 Annex A. Figure 5 shows the model used for the exhaust calibration (left) and the corresponding sound power results (right).

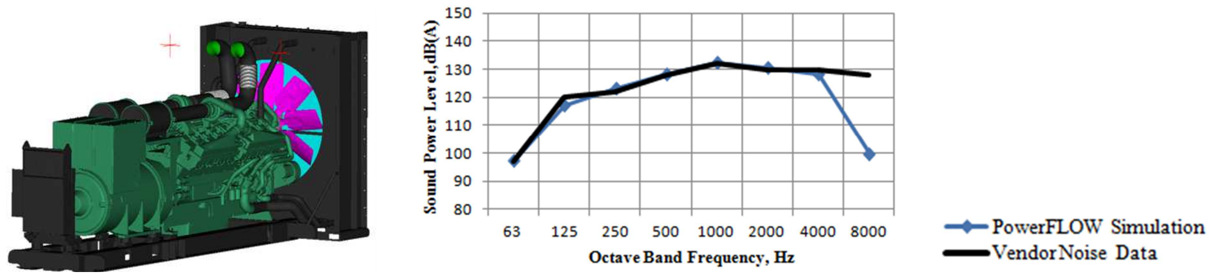

Figure 5: Simulation model used for exhaust calibration (left), Sound power level calibration spectrum (right)

Again, the discrepancy at high frequency was acceptable, as the difference in overall A-weighted sound power level between the test data and LBM result was only 1 dB. The transient mass flow boundary condition was then attenuated by including the silencer acoustic insertion loss. The silencer attenuation curve was provided by the vendor. The calibrated and silencer-attenuated mass flow transient boundary condition was then divided linearly between the four exhaust openings at the top of the silencer in a separate LBM simulation with the outside of the trailer enclosure. SPL at 7 m microphones was extrapolated from simulation measurement surfaces around the exhaust openings using the FW-H method, as described later in section 4 for the cooling airflow openings.

2. Wall transmission model (SEA)

Statistical Energy Analysis is a framework of methods that simulate mid and high frequency dynamics by considering statistical ensembles of mode groups and the dynamical energy exchange between them[7]. Both structural and acoustic mode groups, or subsystems, are included, using a common set of power and energy variables. Spatial and frequency band averaging is applied in estimating response amplitudes from kinetic or potential energy. Coupling between subsystems can be estimated from wave-based measures, such as Transmission Loss (TL) or radiation efficiency. Power inputs to subsystems in frequency bands form the excitation for the SEA model.

The SEA model of the containerized power generator, comprising interior acoustics, steel enclosure panels, acoustical insulation and coupling to exterior acoustics, was developed using the commercial software SEAM, from Cambridge Collaborative, Inc.[8]. The trailer interior acoustics were divided into three axial subsystems, with absorption coefficients and areas of the acoustical panels providing dissipation. Direct coupling to the exterior was provided by available TL tables provided by vendors of the inlet louvers and the acoustical panels. Analytical coupling built into the software was used for transmission through the untreated steel panels. Excitation of the model was provided to the central interior acoustic space, using free field sound power of the generator, including that of the fan, which was also available from the manufacturer[4]. Because SEA is not particularly suited to model exterior acoustic spreading, special nearfield exterior subsystems were applied around the enclosure to collect and report transmitted sound power from the interior. Care was exercised in applying sufficient damping, in the form of coupling to far-field acoustic sink subsystems, so that no significant sound power was reflected back to the container subsystems. These radiated sound powers were extrapolated to the 7 m microphone locations using a simple, free-field spreading model, assuming the power originated at a point at the center of the panel or opening. Only the contributions from SEA wall transmission were considered in the final simulations levels, as noise from the openings and exhaust was handled by the LBM solver, in conjunction with FW-H far-field extrapolation.

4. NUMERICAL SETUP

The primary simulation model is of the airborne sound transmission through inlet louvers and exit grill, using the LBM solver. With boundary conditions determined for the engine and exhaust, the full model was assembled including the enclosure, absorbing materials and openings to the exterior. Figure 6 shows the distribution of the voxel size (fluid grid size) inside and outside of the container.

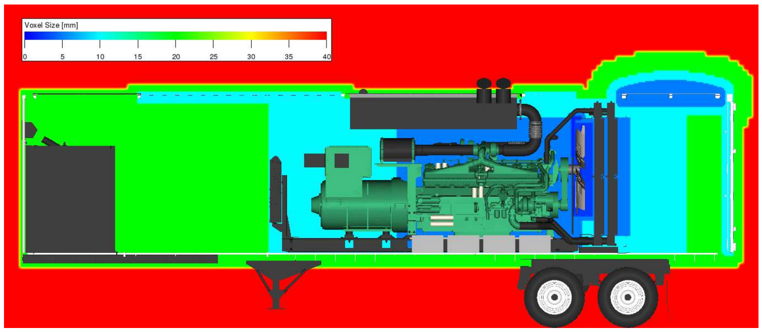

Figure 6: Voxel size distribution inside and outside of the container

The finest resolution of 1.25 mm was allocated to the space surrounding the fan, while louvers and cooling exit openings had the second finest resolution of 2.5 mm. A 5 mm resolution was distributed to cover the entire cooling package, as well as part of the engine where the virtual speakers were located. The rest of the engine bay was resolved with a 10 mm grid, while the whole inside of the container had a 20 mm resolution. Outside the container and towards the location of the microphones, the finest grid resolution was 40 mm and coarsest was 160 mm. Acoustic absorption properties of the interior panels and louver treatments were simulated using porous media features[5]. Two different absorbing panel surfaces were simulated, with one including an impermeable facing and the other a perforated facing. Louver absorption consisted of of fibers behind perforated metal.

There were four microphones each located 7 m from the trailer surface. The microphone name refers to its position relative to the container: front microphone, right microphone, left microphone and rear microphone.

Exterior fluid resolution that would be fine enough to allow sound wave propagation up to ~5000 Hz would result in a computationally expensive simulation. Instead, the Ffowcs Williams–Hawkings (FW-H) analogy was used to compute the sound pressure level at the microphone locations. Figure 7 shows the sampling surfaces, placed at the openings of the container, where flow properties computed directly with LBM (transient density, pressure and velocity components) were collected and used as input for the FW-H solver in post-processing calculations.

Figure 7: Ffowcs Williams–Hawkings (FW-H) sampling surfaces (green)

5. RESULTS AND DISCUSSION

Sound pressure levels (SPL) were calculated by summing power contributions from three simulations at the microphone locations: noise transmitted through cooling airflow openings (LBM + FW-H), noise transmitted from exhaust openings (LBM + FW-H) and noise transmitted through container walls (SEA + spreading). Exhaust could have been included in the main opening simulation, but it was separated in order to allow diagnostic images without the exhaust component. This total noise process allowed comparison of the spectrum contributions of the different sources/paths. The strongest path was from the louvers, with wall transmission becoming significant below 250 Hz and above 2000 Hz.

Experiments were completed well after the simulations, with background noise 30 dB below the measured levels. Obstructions and the electrical load bank were located at least 3 enclosure lengths away from the microphones in 3 of 4 directions. Due to site limitations, reflective walls and a corrugated steel canopy were within 2 enclosure lengths from the rear microphone. A Larson Davis model LxT1 sound level meter was used for overall A-weighted SPL, as well as a spectrum of SPL at the left microphone position. Table 1 shows the experimental SPL, simulation SPL and dB difference for each of four microphone locations and for a power average of the four levels. While the level at the rear position is underpredicted by 4 dB, the power average of the four locations is accurately simulated.

Table 1: Overall A-weighted sound pressures at 7 m microphones

	Front Mic.	Right Mic.	Rear Mic.	Left Mic.	Power Average
Test SPL (dBA)	73.4	76.8	77.5	77.0	76.4
Simulation SPL (dBA)	73.8	77.8	73.5	77.8	76.2
Sim-Test (dBA)	+0.4	+1.0	-4.0	+0.8	-0.2

Third octave bands from the experiment were combined into octave bands for comparison to the simulation results at the left microphone, as shown in Figure 9. The largest spectrum discrepancy occurs at the 63 Hz octave band. This could result from use of free field source levels, where the actual source sound power may be enhanced by reflections inside the enclosure.

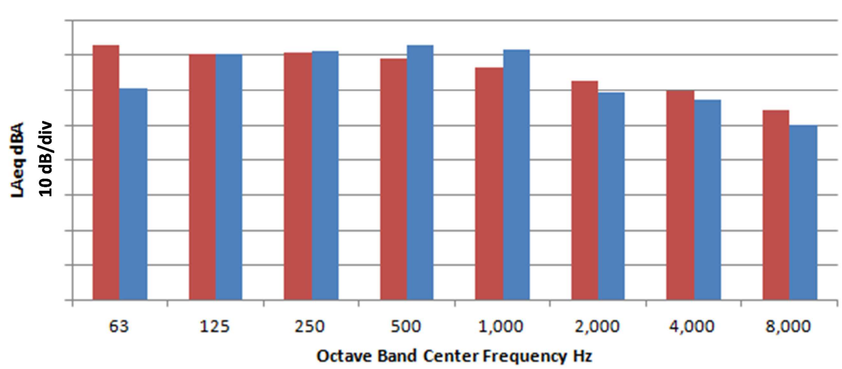

Figure 9: A-weighted SPL at left side 7 m microphone (red experiment, blue simulation)

As an example of the diagnostic images available with LBM simulation, SPL distribution in the centerline plane is shown in Figure 10 for octave bands (dB Maps) with generator virtual speakers and the fan sources active. The 50 dB scale from blue to red shows that the container provides good attenuation, especially for higher frequencies.

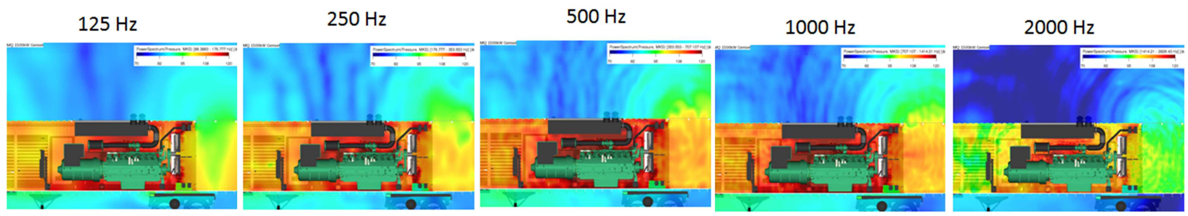

Figure 10: SPL distribution in simulation at centerline planes and in octave bands

6. CONCLUSIONS

This paper has described the application of simultaneous flow and acoustic simulation to a 1500 kW containerized power generator. Design assessment was accomplished for total radiated noise from a superposition of contributions from noise sources: exhaust, engine, generator and cooling fan; with attenuation from silencers, acoustic insulation materials and the design of airflow openings. Methodology to predict the total noise for a containerized power generator was developed by combining a flow and acoustic solver based on the Lattice Boltzmann Method (LBM) with a Statistical Energy Analysis (SEA) structural acoustic model. The LBM model included cooling fan noise, engine/generator noise, exhaust noise and acoustic material absorption, while the SEA approach handled noise transmission through walls. Validation of the methodology was demonstrated with experiments on initial production units of the new design. Overall A-weighted SPL results were within 1 decibel of overall test levels at 7 m from the walls for three of four microphone locations compared. Test results in the rear location may have been affected by proximity to reflective surfaces. Overall, the power average of the four locations was within 0.2 dB. In addition to the total noise prediction, the simulation provided cooling airflow performance, allowing tradeoffs with noise in assessing the impact of design variations.

ACKNOWLEDGEMENTS

The authors wish to thank Adrien Mann, Franck Perot and Paul Spangler at Exa for valuable contributions to this study.

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