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Total Noise Analysis of a Directional Drill

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ABSTRACT
Off-highway machine and equipment manufacturers are working to lower their products' total noise level to meet local noise regulations, reduce neighborhood disturbance, reduce operator fatigue and facilitate improved jobsite communication. Total radiated noise includes contributions from different noise sources on the equipment including cooling fans, engines, transmissions, mufflers and hydraulic systems. Noise from these sources is often attenuated by the design of hood and chassis openings, silencers, acoustic insulation and louvers. An accurate simulation tool capable of total noise analysis is valuable in the product design and optimization processes.

In this paper, we present the results of a methodology to predict and optimize total noise for a horizontal directional drill. The simulation tool, PowerFLOW, is a flow and acoustic solver based on the Lattice Boltzmann Method. Within the methodology, contributions from different noise sources and absorption from insulation and hood louvers are modeled in the solver as is air flow and cooling system performance. The computational results were correlated with test data. An optimized design was then created based on analysis of simulation results and tested. This approach, coupled with high performance cloud computing, delivers accurate and complete noise analysis well within the short design periods typical of modern engineering project arcs.

1 INTRODUCTION
The noise radiated from directional drills and similar machines originates from a number of sources: engine surface vibration, exhaust systems, air intakes, cooling fans, transmission gearing and hydraulic systems. Noise control treatments for such equipment must be designed for good performance both at the operator location, for less operator fatigue and in the far field, for reduced neighborhood and jobsite disturbance. Noise can be attenuated by the design of hood and chassis openings and the addition of silencers, acoustic insulation and louvers. Early co-simulation of airflow and noise performance permits a larger range of design configurations to be assessed and is a valuable tool in bringing new designs to market rapidly. Catching and correcting cooling and noise issues of a given design early in the product cycle allows for better design tradeoffs before building expensive prototypes and eliminates the need for a major redesign in the middle of product validation, which often can lead to two or three month delays. Because hood openings are critical for cooling and acoustic performance, the ability to analyze both in a single simulation is a valuable resource in reducing design assessment turnaround time and project schedule risk.

This paper describes the application of a transient and compressible CFD solver, based on the Lattice Boltzmann Method (LBM), to simulate flow and acoustics simultaneously in the underhood and immediate exterior regions of a directional drill. Noise generated by the rotation of the cooling fan was simulated directly in LBM, while other sources of airborne noise were produced by virtual speakers in the model. The general approach was previously described in its application to power generators[1]. The capabilities of this CFD solver, PowerFLOW, to simulate acoustic propagation have been described in detail[2], as has the accuracy with which it directly simulates fan noise[3,4]. Acoustic absorption materials are represented in the simulation as an equivalent volume of acoustically porous material[5]. In order to predict operator ear sound and total radiated sound power, propagation to the far field was carried out with an acoustic analogy approach based on the Ffowcs-Williams and Hawkings method (FW-H)[6].

After initial calibration of the virtual speakers and verification of total radiated noise on an existing machine, an optimized design was then created based on the analysis of simulation results. Tests on the new design verified that significant improvements in noise performance were achieved. The next sections describe the directional drill under study, the total noise analysis process, baseline and optimized results with physical validation, and finally, conclusions.

2 DIRECTIONAL DRILL NOISE

Figure 1 shows the simulation model representation of the horizontal directional drill. The machine is about 3.5 m long and 1.5 m tall. Cooling for the underhood components is performed by an engine driven fan pulling in air though the grills located on its top, bottom and left side. Cooling air flow exits through the louvers located at the top and back of the engine bay.

Figure 1: *Directional drill, showing upstream (left side) and downstream (rear & top) openings and louvers*

Figure 2 shows the underhood components of the model including the engine and cooling fan. Engine exhaust is routed through the DOC after-treatment to the exhaust stack. The hood is partially covered by foam, which serves as a noise absorber and insulator. The cooling package has a pusher fan and two heat exchanger cores for hydraulic fluid and engine coolant, respectively.

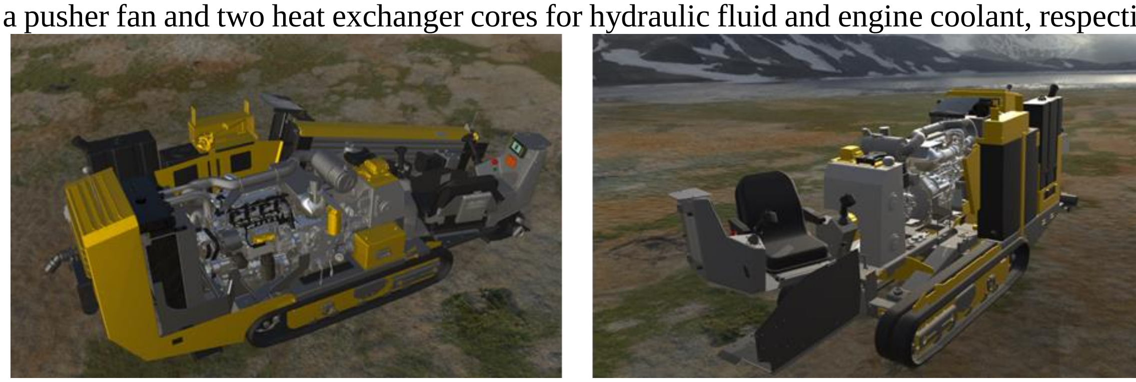

Figure 2: *Directional drill, showing interior, engine, cooling package, exhaust and other underhood components*

3 TOTAL NOISE ANALYSIS

To capture the total noise of the directional drill, it is important to include all major noise sources. Based on previous test experiences for this specific model, it is known that there is strong noise radiating from the engine, hydraulic pump and transmission. Significant noise is also generated by the cooling fan pushing air through heat exchanger cores. The combined underhood noise is attenuated by acoustic insulation panels while EAT (exhaust aftertreatment) exhaust noise is unattenuated and radiated outside the hood.

Cooling fan noise is simulated directly in LBM, providing source contributions and sound directivity with cooling flow. Exhaust, engine and other underhood component noise was calibrated with test data from component suppliers and the OEM, by using virtual speakers. This made it possible to have all significant noise sources and acoustic treatments modeled in one simulation. The numerical approach for predicting the total noise of an off-highway machine can be described in three steps, with details provided in the following subsections:

1. Acoustic (no flow) simulations calibrate sub-component noise sources to component or machine acoustic test conditions. Virtual speakers were used to model engine and other underhood noise sources. Exhaust noise was modeled by a virtual speaker for this directional drill, but it can also be simulated with the combination of test inputs of exhaust jet flow (flow rate, temperature, muffler internal structure, etc.) and an upstream virtual speaker.

2. Cooling fan noise simulation, where rotating fan geometry acts directly on the underhood air, creating flow and noise from turbulence and blade wake impingement on surfaces.

3. Total noise simulation on the machine, with the combination of all virtual speakers and cooling fan, with machine sound power level calculated based on ISO 3744.

3.1 Sub-Component Virtual Speaker Calibration

Engine noise sources were calibrated to the noise spectrum provided by the engine vendor. In the lab test, five microphones were placed 1 m away from the test engine unit in a hemi-anechoic chamber, as shown in Fig. 3 (left). In the simulation, an array of 5 speakers was placed on engine surfaces above a ground plane, as shown in Fig. 3 (center, right). Speaker transient boundary conditions were adjusted to match the engine radiated sound level. Results are shown in Fig. 4. There was good agreement with the engine noise test, with A-weighted overall SPL of individual and averaged microphones within 0.5 dB and close matching to the shape of the noise spectrum.

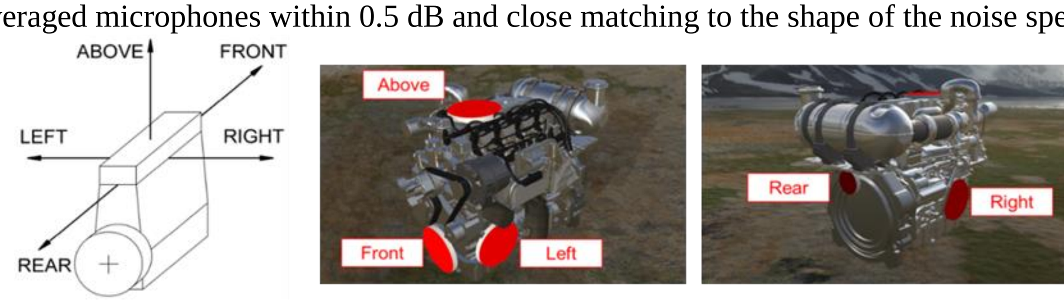

Figure 3: *Virtual speakers on five sides of the engine*

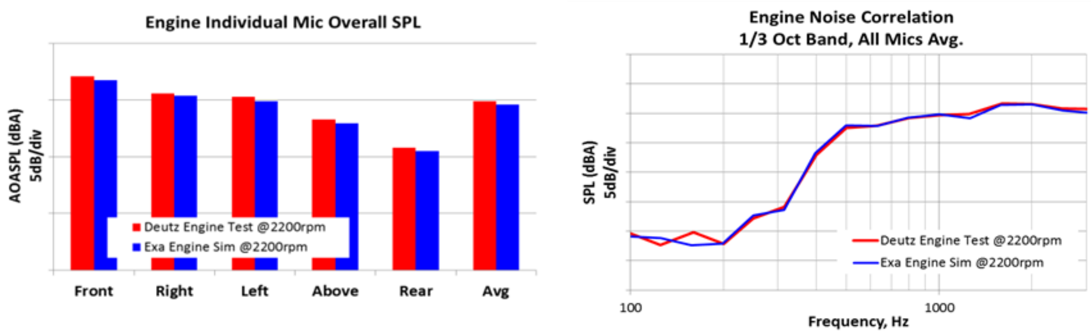

Figure 4: *Engine noise correlation with test microphones (red: lab test, blue: simulation). Individual and averaged microphone A-weighted SPL (left) and averaged 1/3 Octave band result (right)*

Similarly, a virtual speaker was placed at the end of the exhaust pipe, inside the exhaust stack on the hood to recreate exhaust noise, as shown in Fig. 5 (left). Heavy blankets were placed on top of the tested machine to reduce contamination from other sources. The fan was removed to eliminate fan noise. Three microphones were placed 0.5 m away at 45 degree intervals from the exhaust jet direction, as shown in Fig. 5 (center). Simulation agreed well with the averaged test spectrum after source calibration, as shown in Fig. 5 (right). A-weighted overall exhaust SPL is within 0.5 dB.

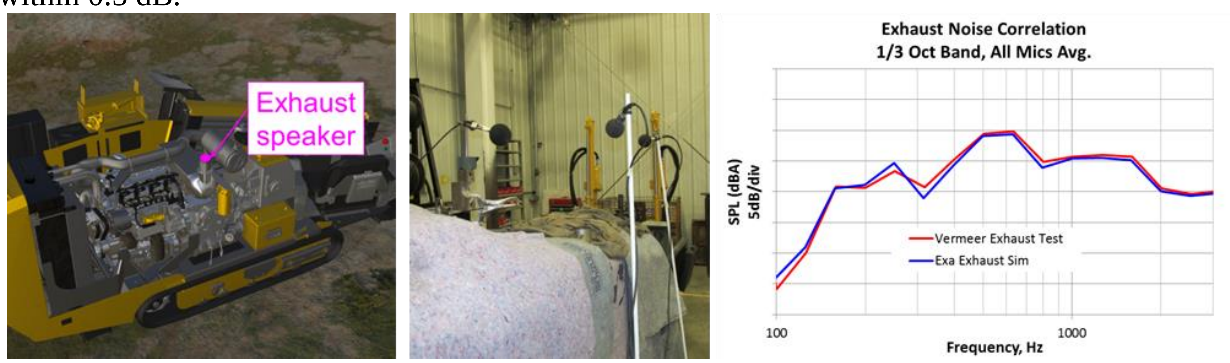

Figure 5: *Exhaust virtual speaker setup (left), covered test machine with microphones (center) and spectra (right)*

The remaining underhood speaker calibration was slightly different from the engine and exhaust noise calibration, as there was not isolated noise test data for other individual underhood parts or isolated fan noise test data. As a result, this calibration was made after the cooling fan noise simulation was finished. Simulated noise contributions from the cooling fan were removed from the total noise test data by power subtraction. Five additional underhood virtual speakers were added into the model, as shown in Fig. 6 (left, center), along with the previously calibrated engine and exhaust speakers. All acoustic insulation and louvers were included in the simulation model to correlate with test data. Figure 6 (right) shows good sound power level correlation between test and simulation results with the fan-less configuration.

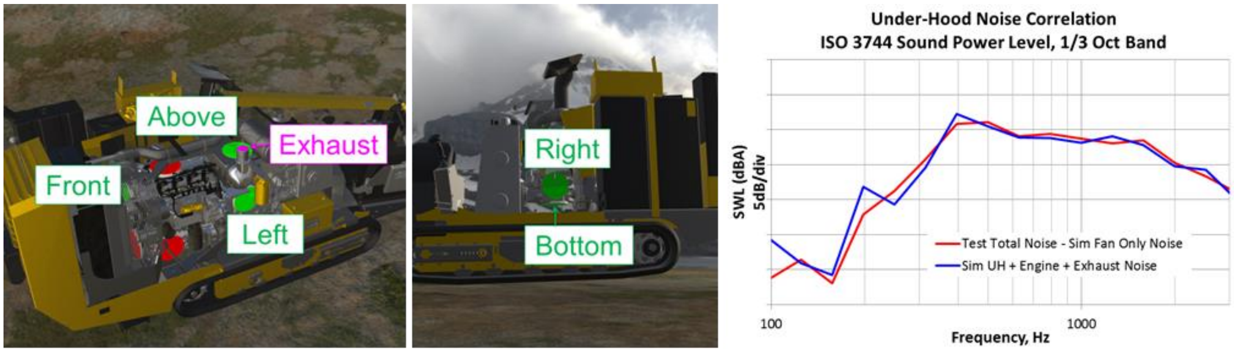

Figure 6: *Five underhood speakers (in green, at left and center), along with engine (red) and exhaust (pink) speakers; with sound power level in 1/3 Octave bands (at right). Simulated cooling fan noise has been subtracted from test (red curve) and compared to simulation with underhood, engine and exhaust speakers (blue curve)*

Combining the cooling fan noise simulation with all calibrated engine, exhaust and other underhood virtual speakers in the full machine, including acoustic insulation and louvers, the propagation and attenuation of the resulting total noise field was captured by the computational method. This method was validated with comparisons to both the baseline and optimized final design full machine noise test data, as shown in the results section.

4 NUMERICAL SETUP

The entire test machine is modeled in the simulation with as much production detail as available. The cooling fan was prepared directly from the manufacturer's CAD geometry. All rotating parts were handled using a Local Reference Frame (LRF) domain, which is separated from the fixed outer domain by a closed interface. The outer domain grid is fixed with respect to the ground while the inner domain grid is physically rotating with the cooling fan. As implemented, the sliding mesh LRF scheme for fan performance and noise has been validated in multiple studies[3,4]. The flow induced noise contribution is directly simulated during the transient flow simulation. Twelve simulation microphones are defined by following ISO 3744 along a hemisphere with a 10 m radius, as shown in Fig. 7.

The simulation domain is contained within a very large cubic box, which includes an anechoic sponge zone on the boundary to represent non-reflecting boundary conditions. All solid surfaces including floor/ground are defined as rigid walls and noise insulation foam is modeled as an acoustic porous medium, with acoustic absorption properties obtained from the insulation vendor. The two heat exchanger cores present in the test, the radiator and oil cooler, are modeled in Exa's 1-D heat exchanger software, PowerCOOL, using supplier provided cooling air-flow resistance, heat transfer coefficients, flow and heat rates[7,8]. Time domain convergence of the cooling flow and noise is reached after four fan revolutions and the flow and acoustic recordings start at this time.The full test machine simulation domain is composed of 180M voxels (fluid cells) with the finest grid resolution near the fan blade tip. A permeable FW-H sampling surface box covers the whole machine. The transient flow properties computed directly with LBM (density, pressure and velocity) are collected and used as input for the FW-H post processing solver to propagate sound signals to the 10 m far field microphones. Total physical time modeled was 0.41 sec, in order to obtain a smooth 16 Hz narrowband spectrum. This time corresponds to 18 complete fan rotations. The model ran for 40 hours of wall-clock time on ExaCLOUD utilizing 336 cores. Underhood virtual speaker simulations were run using the same input CAD. Without fan rotation, the simulation domain is composed of 110M voxels and requires only 4 hours on the same 336 core cluster. Added to the one day required for baseline preparation, the total simulation turnaround time is under one week. This allowed more alternative design solutions to be evaluated than would be possible with physical prototypes and lab tests in a fixed project timeline.

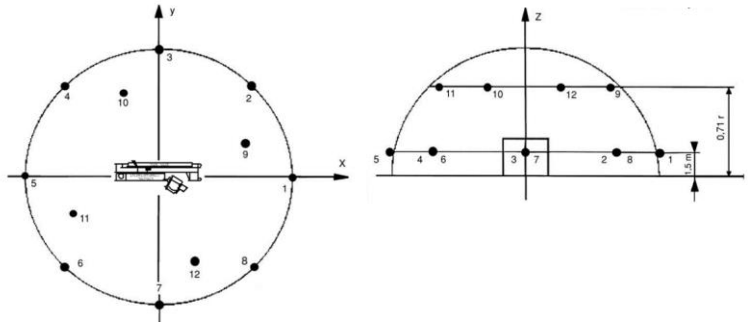

Figure 7: *Twelve test microphones located on a 10 m radius hemisphere based on ISO 3744*

5 RESULTS AND DISCUSSION

5.1 Baseline Correlation

Test machine sound power level was calculated from a 12 microphone SPL based on ISO 3744. The total noise analysis simulation combines all noise sources and allows direct comparison with the test data. Baseline experiments were completed with background noise 30 dB below the measured levels. A Quest SoundPro SE/DL sound level meter was used for overall A-weighted SPL. The test directional drill was running at rated engine speed without load and sitting on a concrete sound pad in the operating configuration.

Figure 8 shows A-weighted overall SPL for 12 individual microphones and sound power levels in 16 Hz spectrum, with test in red and simulation in blue. Most microphones are within 2 dB, except microphones 5 and 8, and the sound power levels are within 0.2 dB overall. The good agreement of the individual microphones indicates accurate directivity prediction. The narrow band spectrum shows the capability of capturing details of the machine noise signature.

To analyze the noise field, the baseline design SPL distribution in the centerline vertical plane and horizontal plane are shown with 500 Hz octave bands dB maps in Fig 11 (left). These results suggest that the downstream front and top openings are the main noise propagation outlets. Under machine sound reflection and propagation was also characterized. Better hood noise insulation in key areas and the redesign of hood acoustic louver openings are recommended based on the simulation results with considerations made to minimize impact to cooling airflow performance based off the simultaneous cooling simulation.

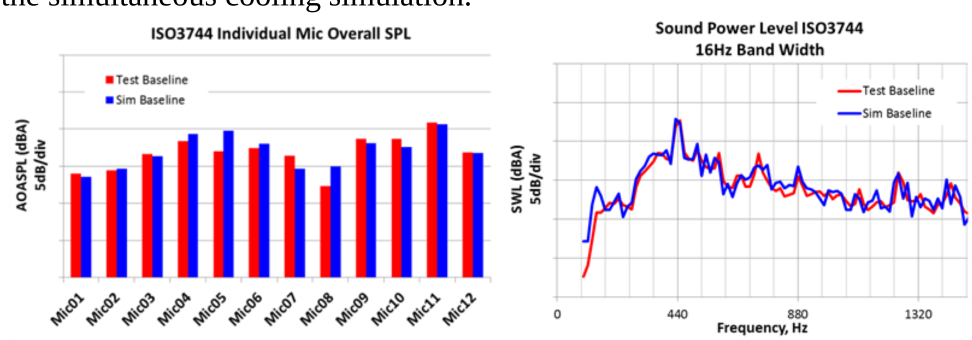

Figure 8: *Baseline noise correlation (red test, blue simulation). A-weighted overall SPL of individual microphones (left) and ISO 3744 sound power level spectrum in 16 Hz bandwidth (right)*

5.2 Design Optimization

Once an explanation for the dominant noise sources and paths was identified from the baseline simulation, the directional drill design engineers were able to provide design alternatives and evaluate them simultaneously for cooling and noise reduction. Design alternatives included multiple fan and shroud geometries, engine enclosure vent placement and size, as well as foam thickness and location. Having the ability to evaluate multiple designs resulted in being able to implement designs that had the least impact on total manufacturing cost and increase to the physical size of the machine.

The final optimized design was tested and simulated. All virtual speakers remained the same as calibrated from the baseline design, while other changes in the optimized design test were updated in the simulation. These changes were in the fan, shroud, underhood component, hood and hood insulation geometry. Figure 9 shows A-weighted overall SPL and sound power level spectrum, with test in red and simulation in blue. Most microphones are within 2 dB and the sound power levels are within 0.3 dB overall. The good agreement of the individual microphones and sound power narrow band spectrum confirmed the accuracy of this methodology.

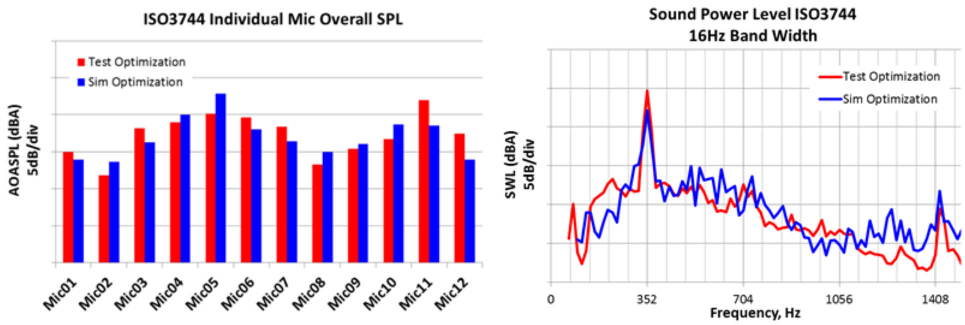

Figure 9: *Optimization design noise correlation (red test, blue simulation). A-weighted overall SPL of individual microphones (left) and ISO 3744 sound power level spectrum in 16 Hz bandwidth (right)*

For deltas between individual ISO 3744 microphones, shown in Fig. 10 (left), most changes in the simulation predictions are within 2 dB of the test data. Changes of ISO 3744 sound power level between test data and simulation are shown in Fig. 10 (right). The test data shows a 4.3 dB noise reduction, while the simulation prediction is 4.7 dB.

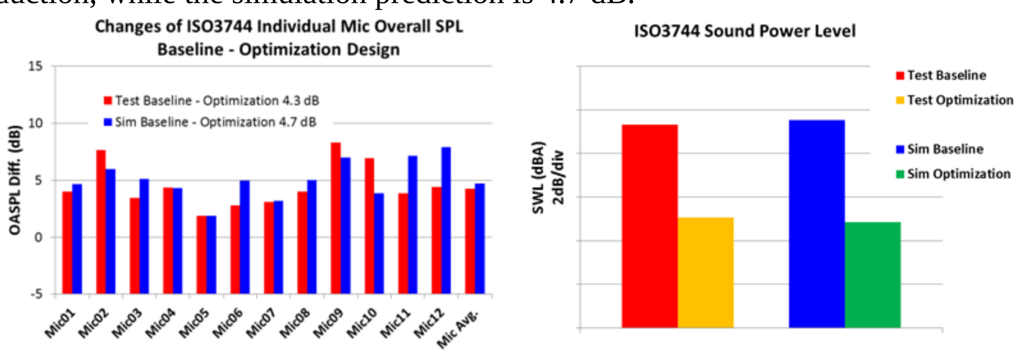

Figure 10: *A-weighted overall SPL of individual microphones (left), ISO 3744 sound power level of test and simulation between baseline and optimization design (right)*

Figure 11 (right) shows 500 Hz octave band dB maps for the optimization design, which includes the BPF tone for both baseline and optimization designs. Compared to the baseline results (left), the optimization design with updated hood, hood insulation, exhaust stack, fan and shroud effectively reduced system total noise that propagates towards far field microphones.

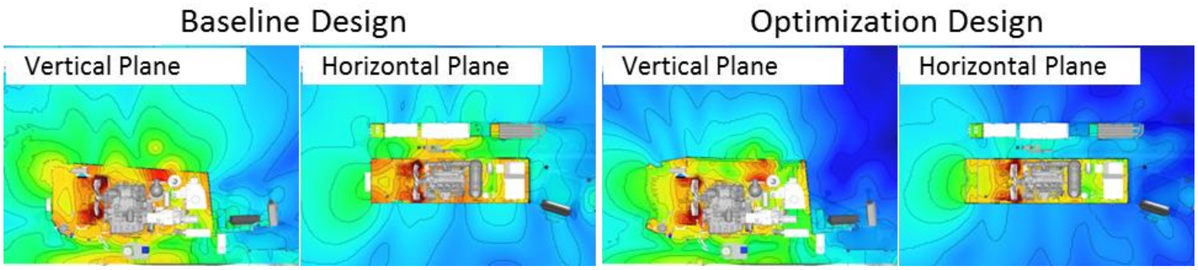

Figure 11: *500 Hz octave band dB maps at centerline vertical and horizontal planes (60 dB scale red-blue). Baseline design result (left) and optimization design result (right).*

6 CONCLUSIONS

This paper has described the application of simultaneous flow and acoustic simulation to a directional drill. Baseline correlation was achieved for total radiated noise from noise sources of engine, exhaust, cooling fan and other underhood components. Combining unsteady cooling flow and acoustics with a CFD solver based on the Lattice Boltzmann Method and the high performance computing resources of ExaCLOUD, total noise was simulated in a short turnaround time. The simulation result can provide more physical insights about interior cooling flow and noise generation mechanisms to provide powerful design guidance and optimization.

A new design was proposed based on the total noise analysis and subsequently tested. The accuracy of the methodology was demonstrated with experiments on both baseline and final, optimized designs. Overall A-weighted sound power levels were within 0.3 dB based on ISO 3744 and overall A-weighted SPLs of most microphones were within 2 dB. The simulations captured both the overall shape of the sound spectra and tones within a few decibels. In addition to the total noise prediction, the simulation predicted cooling flow performance, allowing tradeoffs between acoustic and cooling performance to be assessed. This method has drastically reduced the duration of the product design cycle and has improved performance while minimizing manufacturing costs, all important criteria for these practices to be accepted in industry.

7 ACKNOWLEDGEMENTS

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