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Noise and Sound Quality Optimization of Agricultural Machine Cab

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ABSTRACT

For the development of a self-propelled crop spraying machine, a hybrid experimental and analytical Source-Path-Contribution (SPC) approach is utilized by a leading agricultural equipment manufacturer. The objective is to predict noise and sound quality in the cab before prototypes are assembled, so that dB(A) and SQ targets can be assessed early on and better specifications sent to suppliers to achieve these vehicle-level targets. The experimental SPC task is conducted on the current crop sprayer model, which has the same cab but different engine, transmission and hydraulics than the new model. A hybrid FE-SEA model of the current cab is developed and run at load cases derived from test data. The SEA approach is needed to evaluate the effect of cab acoustic treatments, which are not accounted for in the SPC experimental model. Contributions to in-cab noise for the current sprayer are estimated from both experimental and analytical SPC. Acoustic and structural loads to the cab in the new model are estimated from measurements on hardware components and from supplier provided data. The hybrid FE-SEA model is then updated with the new loads and re-run to predict total in-cab noise and contributions from each source in the new model, which is then used to identify the best countermeasures for noise reduction and SQ improvement.

INTRODUCTION AND OVERALL APPROACH

A leading agricultural farm equipment OEM was in the process of redesigning one of their self-propelled crop spraying machines. The new model was planned to have a different engine and exhaust system to meet the Tier 4i, Stage 3B Diesel emissions requirements as well as major changes to both air intake, hydraulic and chassis systems. The goal of the design team was to identify and correct early on potential issues of the new machine regarding operator noise exposure requirements, such as those in the EU directive [1] , and perceived Sound Quality. In order to achieve this objective, a CAE model of the new machine was developed and used to predict interior noise and sound quality performance prior to developing a working prototype. The current production sprayer was used as a reference and was tested and modeled to validate the prediction process.

At the beginning of the project, it was understood that while the newly designed crop spraying machine would see major changes to engine, exhaust, air intake, hydraulic and chassis systems, the cab would remain basically unchanged. Therefore, in the new crop sprayer, only the loads to the cab would be different and the goal of the project was to predict the response at the interior of the same cab due to the new loads. The current production crop sprayer could be decomposed into loads and cab and the cab response to the existing loads modeled and correlated to test data. The overall project approach is shown in Figure 1.

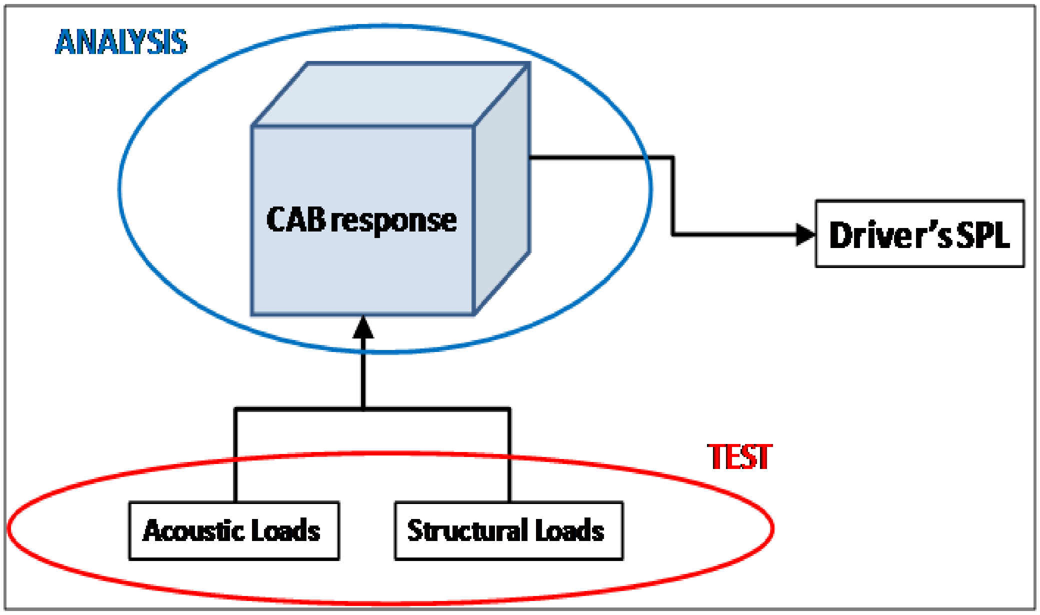

Figure 1. Project Approach

Once the experimental and analytical models of the current production machine were developed and validated, the CAE model of the cab was excited with the new loads, to predict the change in noise level and sound quality performance at the driver's ear. The steps of this hybrid test-CAE approach are summarized in Figure 2.

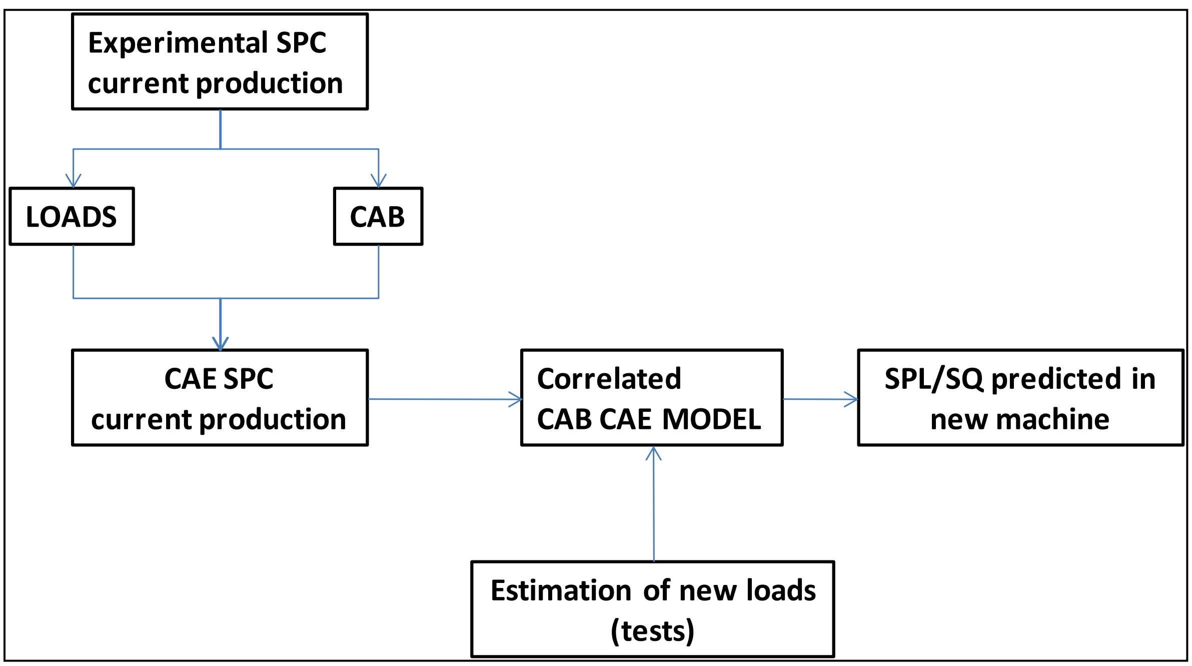

Figure 2. Project steps to build the hybrid test-CAE model of the crop sprayer

TESTS OF CURRENT PRODUCTION CROP SPRAYER

The crop spraying machine was tested with two objectives in mind:

- To develop a Source-Path-Contribution (SPC) model of operator's noise [2], to understand the dominant contributions to the A-weighted SPL at driver's ear and identify key areas/subsystems in need of improvement to achieve the desired dB(A) target (objective #1)
- To assess the Sound Quality at the operator's position, to understand the features of the interior noise that affect the perceived image of the crop spraying machine and establish a Sound Quality target (objective #2).

THE SOURCE-PATH-CONTRIBUTION (SPC) MODEL

The SPC approach builds a model of the vehicle by quantifying sources (acoustic and structural) and the energy transfer paths from sources to receiver. This is a well documented process that can be summarized as follows

$$P_{receiver,acoustic} = \sum[(P/Q) \cdot Q] \quad \text{(eq. 1)}$$
$$P_{receiver,structure} = \sum[(P/F) * F] \quad \text{(eq. 2)}$$

$$P_{receiver} = P_{receiver,acoustic} + P_{receiver,structureborne} \quad \text{(eq. 3)}$$

where Q is the source strength of each acoustic source, F is the Force injected into the structure from each structural source of vibration, P/Q and P/F are Frequency Response Functions measured with an acoustic artificial source (P/Q) and with a shaker (P/F) between each source and the receiver. The total noise at the receiver is due to the contribution from all acoustic and all structural sources and paths. The sources are characterized by measurement of their operating levels, while the paths are characterized by FRF measured under artificial excitation conditions.

For this project, the sources considered in building the experimental SPC model were:

- Acoustic: engine, exhaust tailpipe, exhaust shell, intake snorkel, hydraulic pumps, steering pump
- Structural: cab mounts (3)

The computation of the structureborne paths in (eq. 2) can be done in two different ways. The first method, generally referred to as the Mount Stiffness Method, estimates the Force injected into the structure by measurement of the displacement across the mount (in operating test) and of the mount dynamic stiffness (on a lab fixture for mount testing). The second method, referred to as the Impedance Matrix Method, estimates the Force from measurements of operating accelerations and of the (A/F) matrix by artificial excitation tests.

For this project, the mount stiffness method was used based on the interest of the vehicle design team to quantify the cab mount performance and to use it in their CAE model. Measuring the dynamic stiffness outside of the vehicle system allows the design team to specify desired dynamic mount performance which can be qualified independent of the vehicle system, and compared across vehicle platforms.

For the operating conditions, the crop sprayer was tested at several representative conditions, including:

- WOT at 7.25 kph which is equivalent to the European Union operating measurement standard

- 2nd gear 2100rpm the most typical operating condition
- 2nd gear 2100 rpm with heavy steering maneuvers
- Slow acceleration from 0 to 7.25 kph at engine low idle
- Stationary engine speed sweeps from idle to WOT
- Stationary engine steady state measurements at 500 rpm increments
- Moving engine speed sweep with constant gear setting
- Test with and without the AC on

Figure 3 shows the frequency spectra in both 1/3rd octave spectra (up to 4kHz) and narrowband (up to 1.4 kHz) format for the EEC condition (WOT @ 7.25 kph). In the 1/3rd octave spectra one can clearly see that the frequency range of 100 – 2500 Hz contains most of the energy that contributes to the A-weighted Sound Pressure Level, with the 125 Hz band having the highest impact. From the narrowband spectrum, it can be seen that the level of the 125 Hz band is driven by the 3rd engine order at 116 Hz.

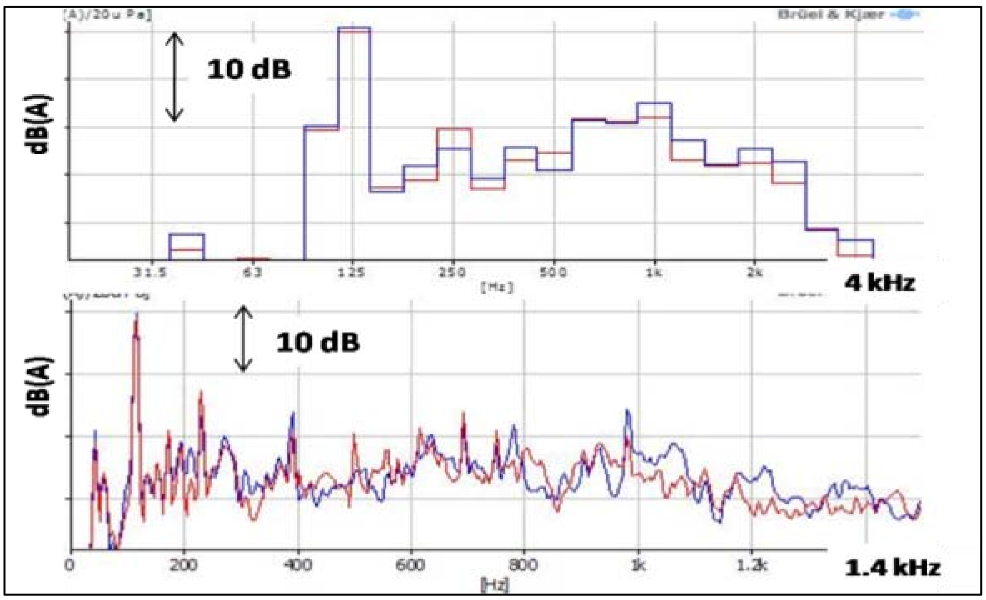

Figure 3. Baseline frequency spectra at driver's SPL

After the 125 Hz band the next highest contribution bands are between 630 and 1000 Hz. This frequency range in the narrow band spectra, while containing some noticeable peaks, is best described as broadband. During the initial study a series of "what-if" studies were conducted to evaluate the effectiveness of reducing the 3rd engine order and broad band response in order to reduce the overall A-weighted SPL. Two of the scenarios are shown in Figure 4. The amount of dB reduction shown in the figure refers to the reduction of the overall A-weighted SPL. One can see that by reducing the 3rd engine order by 9 dB an overall reduction of 4 dB can be expected, likewise by reducing the broadband level (630-1000 Hz) by 5 dB an additional 2 dB reduction can be expected in the overall SPL. These scenarios help to define the focus of the subsequent tests and design of countermeasures.

Figure 4. What-if scenarios to identify effect of possible countermeasures

At the core of the SPC model are the Transfer Functions from the indicator microphones (placed in the proximity of the sources) to the sources, and from the sources to the receiver. There are two general types of Transfer Functions used in this model, purely acoustic and vibro-acoustic. The acoustic transfer functions consist of two sets including the local transfer functions (LTF) which account for the paths between the indicator microphones and the sources and the source-receiver transfer functions that define the paths from the source location to the receiver. Both transfer functions can be measured simultaneously, if a volume velocity source (VVS) is placed at the source location. Figure 5 shows both a local TF, left, and source-receiver TF, right.

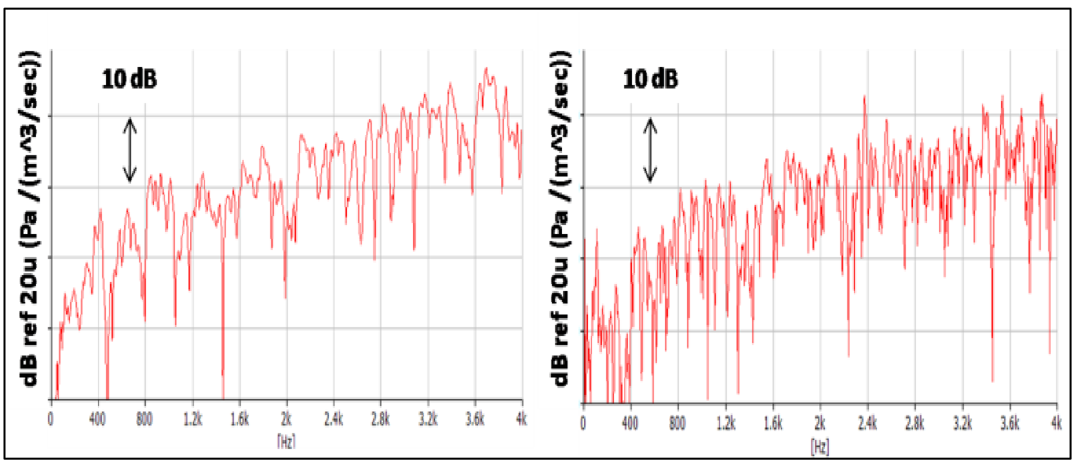

Figure 5: Source-receiver TF, left, local TF (LTF), right,
measured using a VVS at the source location.

The vibro-acoustic TFs can be measured in a similar manner, but rather than using a VVS, a shaker or impact hammer is used to excite the structure through a force transducer. A picture of a test set-up using a shaker input is shown in Figure 6, where the cab is suspended and the shaker excites the cab at the rear mount in the Z-direction. This test is repeated for 3 axes at each cab mount. An example of one of the measured vibro-acoustic TFs is shown in figure 7.

Figure 6: Test set-up using a shaker input to measure the cab transfer functions (P/F).

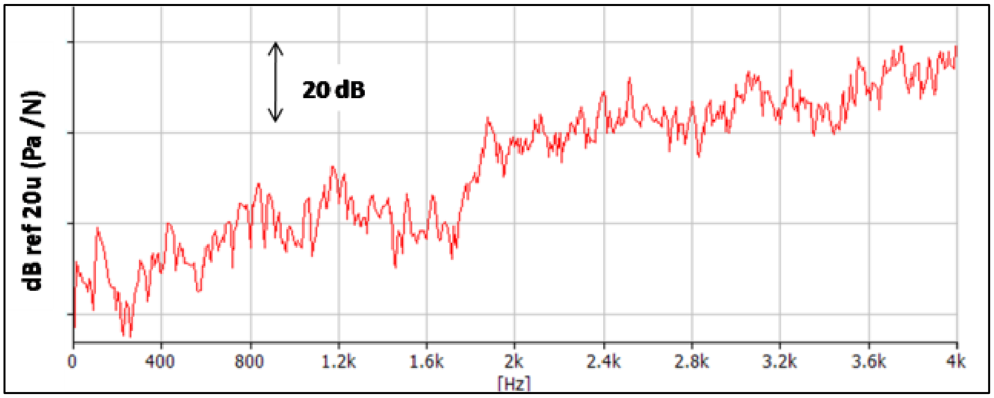

Figure 7: Example of measured P/F function

The last piece of data required to compute the contribution from each cab mount is the dynamic stiffness of the cab mounts. Each cab mount was then tested by an external lab and the resulting stiffness curves used in the SPC model. The measured dynamic stiffness curve for one of the front cab mounts, cylindrical bushing type mount, is shown as an example in figure 8.

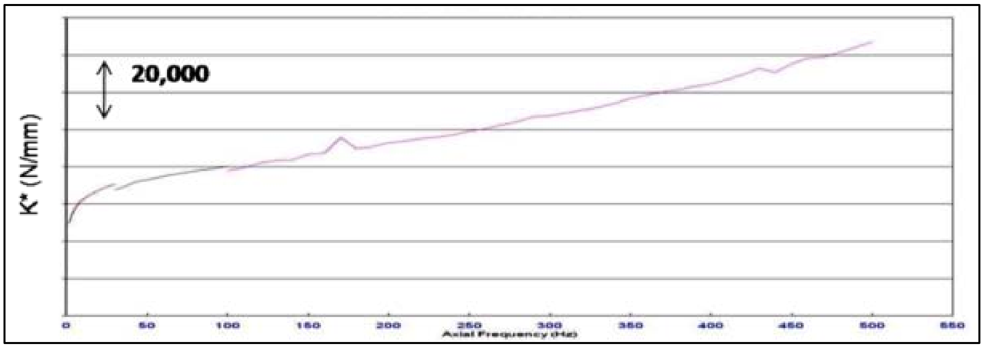

Figure 8: Dynamic stiffness curve for front cab mount

The overall result of the model is shown in Figure 9, where estimated structureborne (light gray solid curve) and airborne contributions (dashed) are compared to measured total noise (bold line).

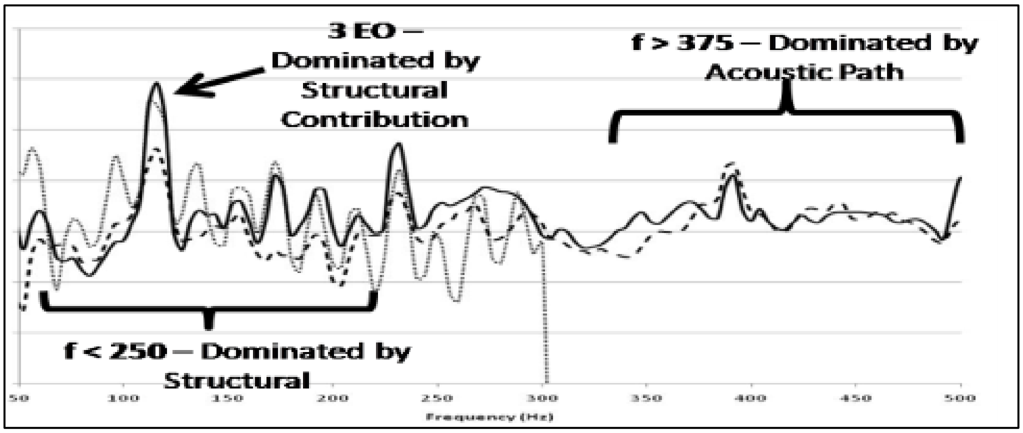

Figure 9. Measured noise spectrum (bold) and estimated structureborne (light gray solid) and airborne contribution (dashed)

As shown in figure 9, the operator's ear SPL is dominated by the structural sources (cab mounts) below 250 Hz, and by the acoustic sources at frequencies over 375 Hz. It was determined that the structural contribution at frequencies below 250 Hz, including the 3rd engine order, was dominated by the front mounts in the vertical direction, and that the main acoustic source at frequencies over 375 Hz was the engine radiated noise.

ASSESSMENT OF THE SOUND QUALITY OF THE CROP SPRAYING MACHINE

In conjunction with the SPC model development, the vehicle Sound Quality was evaluated.   The goal of the Sound Quality portion of this project was to ensure that the approach taken to reduce the overall A-weighted SPL in the cab did not amplify negative aspects of the vehicle acoustic signature, as measured at the driver's position. It is understood that while an overall reduction in SPL would be considered an improvement, if this reduction increased the prominence of the hydraulic pump tones, which may be considered unpleasant, or completely eliminated the engine orders and half orders, which are associated with power, the overall subjective impression of the change would be negative. For this reason the prominent features of the production level vehicle were identified, and compared to the expected changes based on the predicted new model driver's position response, from the SPC model.

The dominant features affecting sound quality were identified to be:

- Loudness, or Amplitude of the sound, measured by dB(A) or the psychoacoustic metric Loudness
- Tonality, or amount of narrow-band frequency components (tones), clearly noticeable above the background noise. It is measured by the psychoacoustic metric Tonality or by Tone-to-Noise Ratio or Prominence Ratio or more simply by comparing levels in adjacent third-octave bands.
- Sharpness, or amount of broadband noise in the higher frequencies, compared to the low frequency content. Measured by the psychoacoustic metric Sharpness or by simpler spectral balance metrics (i.e. level at high frequency vs. level at low frequencies).

For each of these features, preliminary sound quality targets were developed and used as reference during the development of the new machine.

CAE MODEL OF CURRENT PRODUCTION

The frequency range of interest was 100 Hz to 1 KHz, and the noise is due to a combination of structure-borne vibration transmitted through the cab structure from the mounts and to airborne noise from the engine, exhaust, intake, etc. SEA models are routinely applied to the prediction of airborne noise in cabs from 200 Hz to 10 KHz [4,5], however, SEA models of structure-borne sound in cabs in the 100 Hz-1 KHz frequency range have proven more difficult for three reasons.

1. The vehicle's structure is not well represented at lower frequencies by the idealized standard set of SEA structural subsystems such as flat panels, curved shells and cylinders.
2. The locations where external structure-borne excitations are applied are typically very complex (engine mount, cab mounts etc.) and usually the SEA model does not contain enough detail of those regions to correctly estimate the input power.
3. The stiff parts of the vehicle, primarily the rails and pillars of the rollover protection structure, transmit coherent energy throughout the cab structure.

The first two difficulties can be overcome with an SEA model by updating the model properties such as modal density, coupling loss factor, or power input obtained from tests or from local FE/BEM models. The third difficulty cannot be overcome within the standard SEA framework. Therefore a hybrid FE-SEA model was developed for this application. The stiff parts of the structure, principally the cab frame is retained as an FE structure and is necessary to model the response of the system in the low frequency range ($f < 250$ Hz), while the SEA model covered the frequency range for $f > 250$ Hz. The hybrid FE-SEA model is shown in Figure 10.

Starting from the CAD model, a FE model of the cab structure was created which allowed non-essential local design details to be removed and the geometry to be cleaned. The mesh density was selected based on the wavelength of flexural waves in the different components the cab structure. Once the FE model was created, the panels were converted to SEA panels and then acoustic cavities of the cab, ducts, etc. created. The trim and noise control treatments were then added, and acoustic loads were defined on each of the panels and forces applied at the mount locations to create a baseline model.

The analysis approach was to model the current production cab using the acoustic and structural vibration loads that were measured, and validate the model by comparison to the measured cab interior levels during typical operating conditions. The acoustic loads for the SEA panel were defined from the measured sound pressures on the cab exterior, via microphones mounted near the exterior surfaces. The accelerations on the cab mounts were measured via accelerometers mounted on the active (frame) and passive (cab) side of the cab mounts and the passive side measured accelerations were used to estimate the forces applied to the cab frame. Figure 11 shows the CAE model of the cab with the locations of its loads. The SEA semi-infinite fluid applies fluid impedance to the SEA panels and also increases the SEA panel damping by adding the radiation loss from the panels to the exterior.

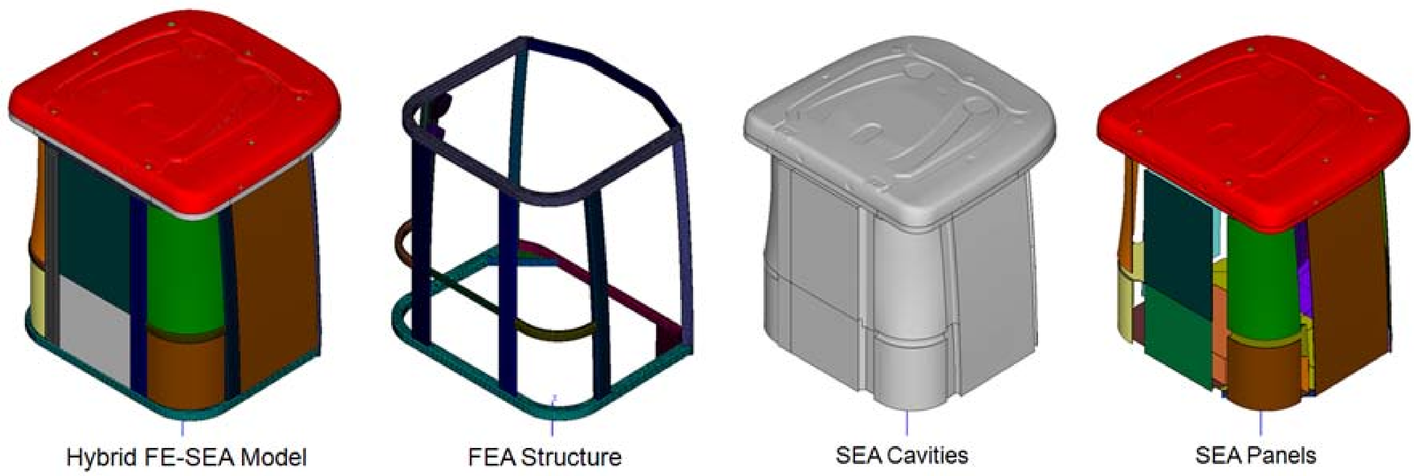

Figure 10. Hybrid FE-SEA model of crop sprayer cab

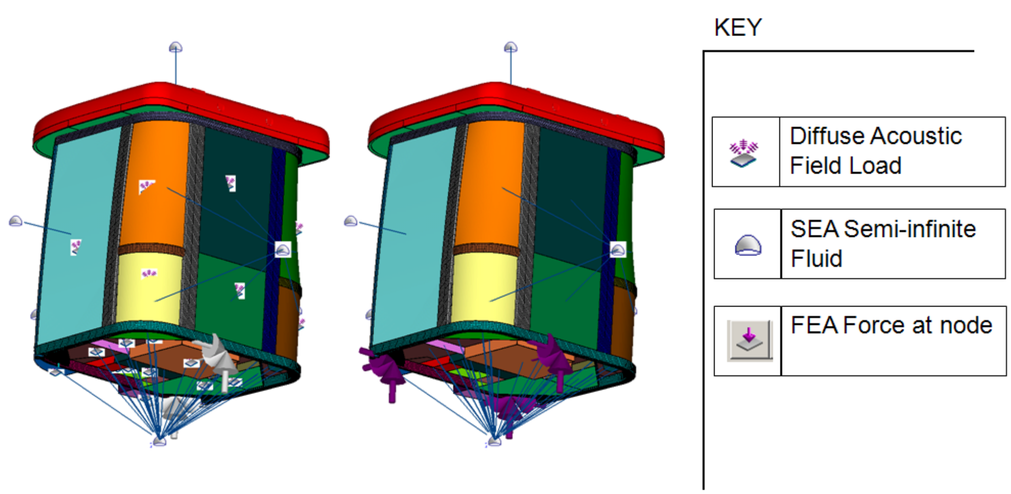

Figure 11: Hybrid SEA/FE model of the cab with the load definitions.

When the FE-SEA model was built from the CAD model created by the design engineering team, material properties for the different structural components were included. For steel structures the material properties were well enough understood, but the modeled glass structures were validated using point accelerance measurements to ensure that the behavior of glass panels was correctly represented. The accelerance measurements were also used to estimate the panel damping using the half-power bandwidth and check the modal density of the panels. After confidence was gained regarding the material properties and geometries, the next step involved validating the entire cab response. This was done by comparing the predicted and measured Frequency Response Functions from the acceleration at the mount locations to the sound pressure at the driver's position (P/F's) and to the glass panels (a/F). This level of validation is intended to confirm that, in addition to the correct material properties and geometries, the connections between the structural components and the interior acoustic treatments are modeled with sufficient accuracy. Once the structural and acoustic FRF's of the cab predicted by the FE/SEA model, were properly validated against the measured FRF's the final step was to correlate the model for different operating loads. This step involves running the model using measured operating loads to predict cabin sound pressure level and panel vibration levels and compare this to the measured interior noise.

Examples showing a comparison of the measured and predicted cab interior noise and glass acceleration levels are shown in Figure 12. Once the FE-SEA model of the current production Crop Sprayer was developed and the model validated, by comparison to operating and response measurements as benchmarks, the baseline model was updated to reflect changes in the cab structure and this new model was used to analyze various iterations of interior treatments to identify best low-noise design options. The acoustic/vibration loads on this validated model were then modified to reflect the predicted loads, from the measured SPC model, that will be present in the new cab.

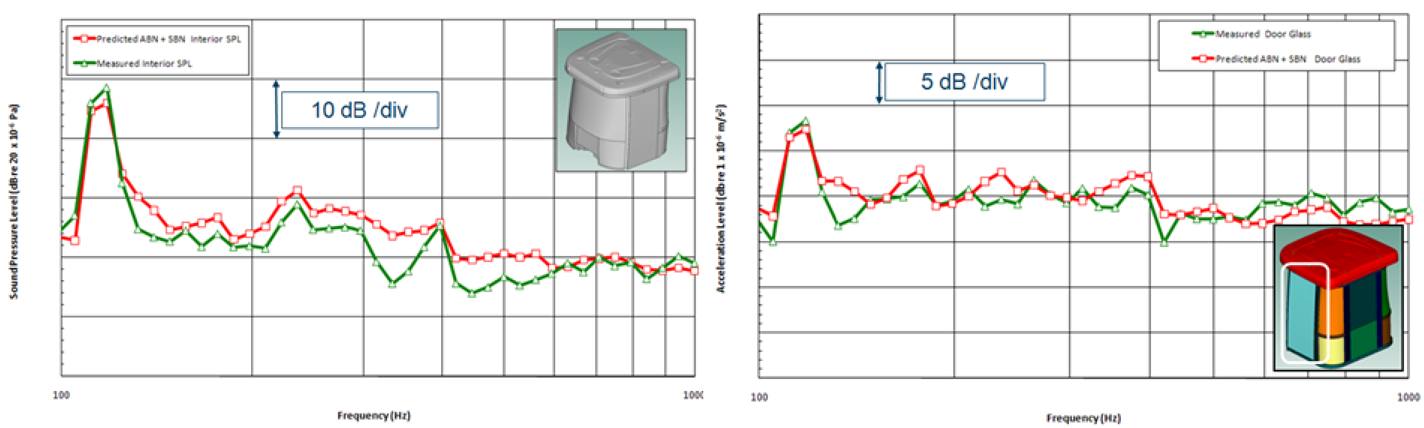

Figure 12: Hybrid FE-SEA – Predicted vs. Measured SPL and Acceleration

CAE MODEL OF NEW CROP SPRAYING MACHINE

Once correlated, the CAE model of the current production machine was updated to reflect a few minor changes to the cab frame and, most importantly, the acoustic and structural loads to the cab due to the different sources (intake, exhaust, engine and hydraulics). The project team first reviewed the new overall design and the technical specifications available for each of the new components and established that the new intake would have a minimum impact on the acoustic load to the cab, due to its design and its relocation farther away from the cab. The engine and fan that were planned for the new model crop sprayer were in use in an applicator type vehicle already in production, so both acoustic and vibration operating measurements were taken under similar operating conditions on the applicator type vehicle to be substituted into the current SPC model. Prototype exhaust hardware that could be tested on an acoustic test bench was available so it was possible to measure the Insertion Loss (IL) of current and new exhaust muffler on the test bench. The difference between the two IL functions was then used to represent the new acoustic exhaust sources in the experimental SPC model. Figure 13 shows the outdoor test setup for the exhaust muffler Insertion Loss.

Figure 13: Insertion Loss test set-up

The new hydraulic pumps that were to be used for the new model were operating at the time in a prototype vehicle so the source levels could be obtained by simply measuring the source levels in the prototype vehicle under similar operating conditions. The last changes expected in the new model involved the hood material and coverage area, as well as a heavy barrier plate at the front of cab-rear engine interface. The changes to the engine hood material were estimated by a simple assumptions based on mass-law TL behavior, while the effect of a change in the coverage area was estimated as difference in sound power proportional to radiating surface. The effect of the change of a barrier-type element in the front of the cab was estimated using the CAE model.

With the new estimated acoustic loads, the model of the new crop sprayer was run to predict the expected baseline SPL. The predicted cab SPL for the current and the new crop sprayer are compared in Figure 14. As shown, the planned changes have a positive effect for f < 500 Hz, where the levels decrease by 5 to 10 dB, with the most notable improvement being a 10 dB reduction of the 3rd engine order. However, the high frequency content was largely unchanged, therefore countermeasures for this frequency range were investigated using the model of the new machine.

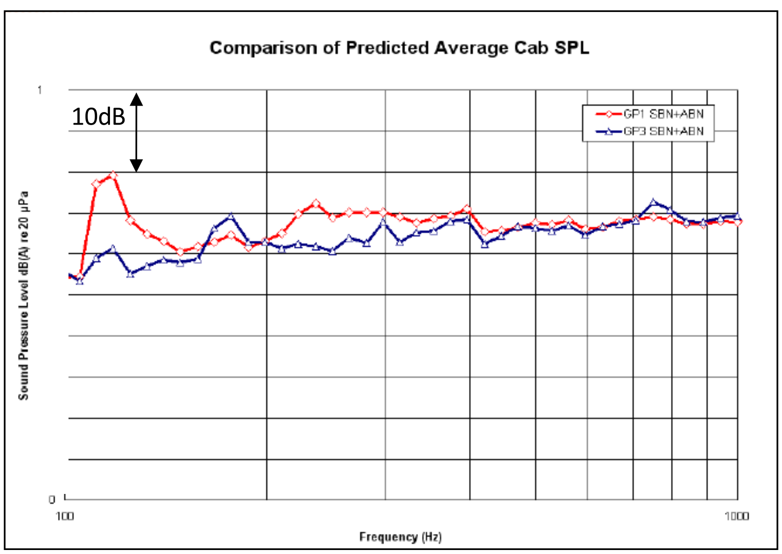

Figure 14. Predicted interior noise spectra for current production (red) and new design machine(blue)

The sound package options for the cab are limited by the available packaging space. There are two main strategies for reducing interior noise levels by improving the noise control treatments. The first is to reduce the noise transmitted into the cab from the body panels by increasing their sound transmission loss and the second is to increase the sound absorption inside the cab. The noise path analysis of the baseline cab FE-SEA model showed that the dominant transmission paths were the glass panels which account for approximately 40% of the cabin surface area. Noise transmission through the floor was predicted to be insignificant due to the heavy floor mat which fully covered the floor. Initial studies showed that for both structure-borne and airborne noise, increasing the thickness of the glass did not reduce the interior noise (<0.2 dB) in a mass or cost effective manner. The model showed the dominant noise transmission paths to the cab interior is through the glass panels, and since the amount of glass cannot be reduced to maintain maximum visibility for the operator, improved sound absorption in the cab was the chosen strategy to reduce A-weighted SPL and improve Sound Quality. This was achieved by improving the sound absorption in the cavity, maximizing the coverage and improving the performance of the selected materials.

Figure 15 shows the cab interior surfaces that could possibly be covered with sound absorption material. A number of design iterations were run, with different combination of surface coverage, material selection and material thicknesses to find the most effective among the production-feasible solutions and determine the sensitivity of the cab noise levels to increased sound absorption.

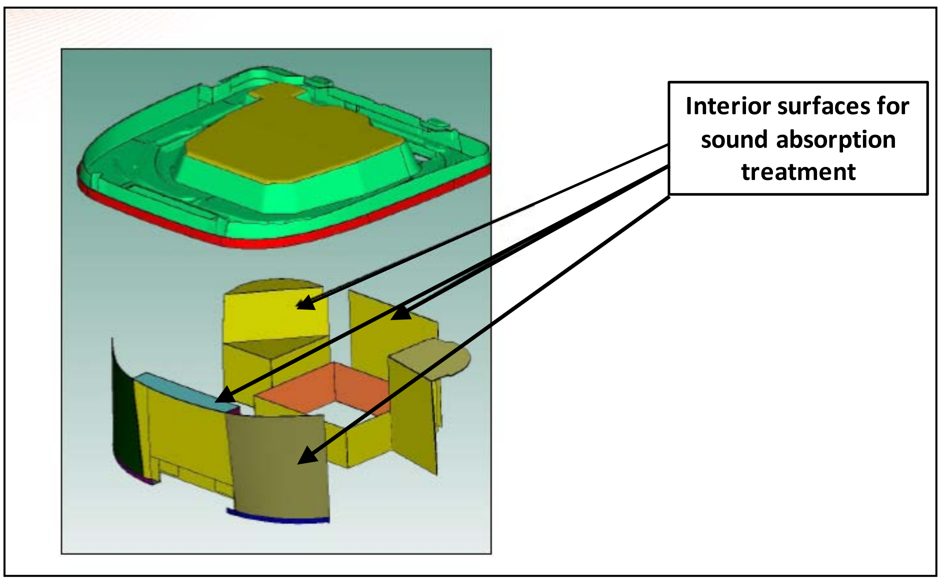

Figure 15: Interior surfaces where sound absorption treatment was simulated.

An example of the effectiveness of four of these iterations is shown in Figure 16, where the total sound absorption in the cab for each treatment is shown in metric Sabines on the left and the corresponding overall predicted SPL is shown on the right. It can be seen that the total sound absorption is improved above 300 Hz and the most effective and production feasible solution yields a reduction > 2dB of the overall A-weighted SPL, however there will be additional benefit in the higher frequency range above 1 KHz. The model did show that further improvements (~4 dB) were possible, however packaging and design constraints limit the additional material that can be added to the cab interior surfaces. Therefore additional improvement could be achieved by reducing the exterior noise level by improving the encapsulation of the engine compartment to reduce the radiated noise.

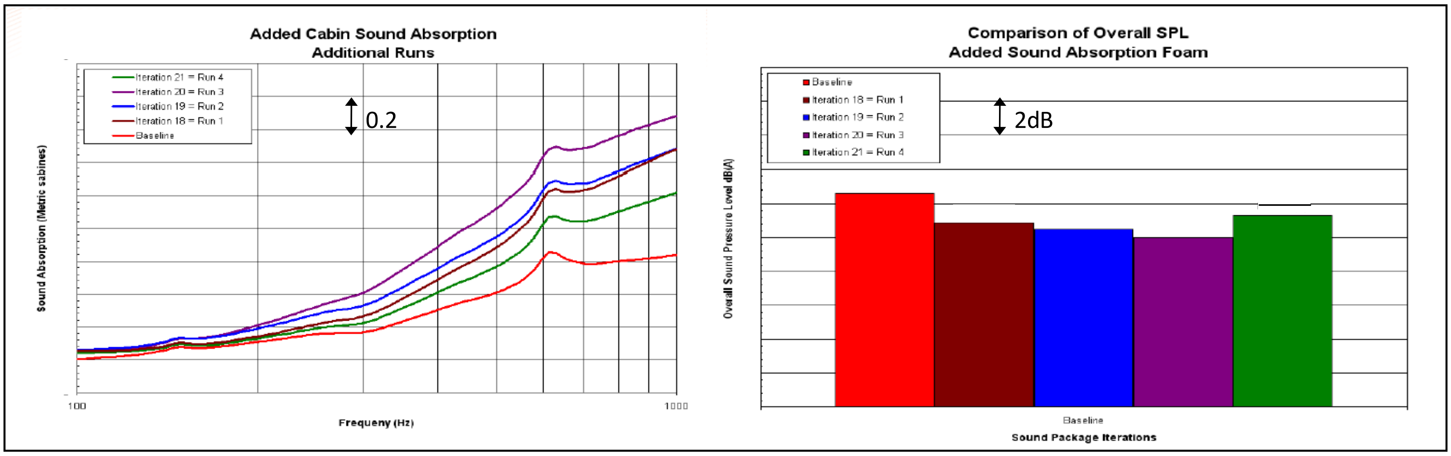

Figure 16. Model predictions for sound absorption in cabin (left) and corresponding overall A-weighted SPL (right) for 4 iterations

CONCLUSIONS

A hybrid experimental-CAE approach was used by a leading agricultural farm equipment manufacturer to ensure the compliance of its new design crop spraying machine to European noise regulations. The tests and evaluations performed on the current production proved very useful to establish realistic and feasible noise level and sound quality targets for the new machine. The combined use of test and analysis data allowed the design and development team to understand early on the impact of design choices on noise and SQ so that better specifications could be given to suppliers and appropriate countermeasures could be designed in.

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