9-12 AUGUST 2015
inter.noise 2015
SAN FRANCISCO, CALIFORNIA USA

# SEA Modeling and Validation of a Truck Cab for Sound Package Optimization

Yong Sok Jang[a)]
Jong Young Kuk[b)]
Jong Chan Park[c)]
Commercial Vehicle CAE Research Lab, Hyundai Motor Company
150, Hyundaiyeonguso-ro, Namyang-eup, Hwaseong-si, Gyeonggi-do, 445-706, South Korea

Rabah Hadjit[d)]
Himanshu Dande[e)]
ESI North America
32605 W12 Mile Rd, Farmington Hills, MI, 48334, USA

Eric Frank[f)]
Sound Answers, Inc.
6855 Commerce Blvd, Canton, MI, 48187, USA

**Consumers and regulations drive the noise requirements to levels constantly decreasing. For vehicles, noise control strategies include the application of noise control treatments on the vibrating panels. The definition of a sound package often requires balancing conflicting attributes like weight, costs and acoustic performance. A Statistical Energy Analysis (SEA) model can be used to support designers and engineers to optimize the sound packages of vehicles. In this paper, we present the results of a project where we studied the performances of the sound package applied on a truck by developing a full vehicle SEA model in the commercial software VA One.**
**The model includes a detailed definition of the main panels and all the noise control treatments. We applied airborne noise excitations for the sources such as the engine, the intake and the exhaust system. The source strength for each excitation has been derived from NVH testing and the model was validated against reference measurements. With this,**

a) email: ysjang76@hyundai.com
b) email: jykuk@hyundai.com
c) email: impactpark@hyundai.com
d) email: rabah.hadjit@esi-group.com
e) email: Himanshu.Dande@esi-group.com
f) email: eric.frank@soundanswers.net

**we demonstrate that SEA modeling is an effective tool to balance the requirements for noise, costs and weight reduction.**

## 1 INTRODUCTION

Product noise requirements are constantly driven to lower levels by consumers but also laws and regulations. For trucks, a reduced interior noise level gives a competitive advantage to the manufacturer as it gives a higher level of comfort for the operator. It also contributes to the brand image since lower noise is always associated with higher quality.

Multiple tools and methods are available to support designers and engineers with the development of the ideal sound package in order to balance the noise requirements with other conflicting attributes such as cost and weight. Statistical Energy Analysis (SEA) is the ideal tool to optimize the sound package of vehicles like car, bus, train and trucks for the airborne noise transfer path. With SEA, engineers build simulation models representing all the noise transfer paths from sources like intake, exhaust, powertrain and tire to the receiver at the driver head space.

In this paper, we present the SEA modeling and simulation for a middle size truck. We first describe the model creation process and we illustrate the model that we built in the commercial software VA One from ESI Group. We then present the Noise and Vibrations measurements that were performed to characterize the noise sources and validate the model. We then present the simulation results and how they compare against the reference measurement data.


Fig. 1 - A middle size Truck

## 2 SEA MODELING

### 2.1 Model Description

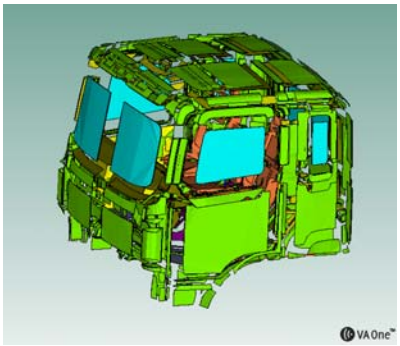

Fig. 2 - SEA model of the Truck (Structural Subsystems).

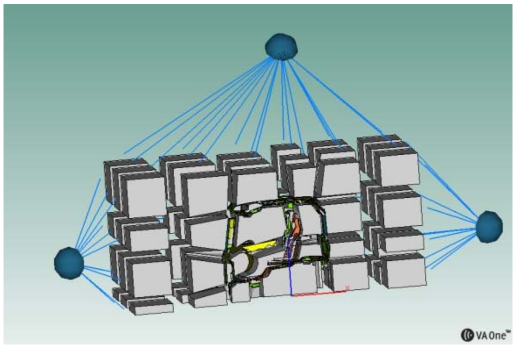

Fig. 3 - SEA half model of the Truck.

The SEA model of the Truck is divided into structural and acoustic subsystems. Structural subsystems consist of flat panels, singly curved shells or doubly curved shells and are capable of storing kinetic energy. The structural panels are represented in Fig. 2. The acoustic subsystems are SEA acoustic cavities modeling air volumes; they store the acoustic energy. The interior volume is divided into several acoustic cavities and such cavities are also used to model the outside air volume touching the vehicle panels.

The SEA subsystems exchange energy through junctions: line and point junctions between structural subsystems and area junction for acoustic subsystems or between structural and acoustic subsystems.

SEA establishes the power balance for each subsystem taking into account the power input coming from the external excitation, the power flowing between subsystems and the power dissipated in each subsystem. The dissipation is characterized by damping loss factors which are defined by the user or calculated from material properties.

The SEA power balance equations are written in frequency bands, typically in 3$^{rd}$ octave as:

$$\Pi_1 = \omega\eta_1 E_1 + \Pi_{12} \tag{1}$$

Where:

- $\Pi_{12}$ is the power flow between subsystems 1 and 2;
- $\omega$ is the center frequency of the band investigated;
- $E_1$ is the energy stored in each subsystem in the frequency band investigated;
- $\Pi_1$ is the power input in the subsystems 1, and,
- $\eta_1$ is the damping loss factor for subsystem 1.

The power flow between 2 subsystems is proportional to the amount of modal energy in the frequency band of interest:

$$\Pi_{12} = \omega\eta_{12}n_1\left(\frac{E_1}{n_1} - \frac{E_2}{n_2}\right) \quad (2)$$

Where:

- $\eta_{12}$ is the coupling loss factor between the subsystems 1 and 2;
- $n_1$ and $n_2$ are the number of modes in the frequency band investigated for the 2 subsystems;
- $E_1$ and $E_2$ are the energy stored in each subsystem in the frequency band investigated.

The SEA has been implemented in the commercial code VA One from ESI Group. The software provides a graphical user interface (GUI) and a solver. The user can use the GUI to create and define the subsystems. The software allows creating junctions automatically when subsystems share common nodes and when required, additional junctions can be defined by the user.

The solver provides the user with tools and methods to calculate all the parameters required to build and solve the SEA equations.

This includes:

- Algorithms to calculate the modal density or number of modes in band for structural and acoustic subsystems based on their geometry and material properties;
- Algorithms to calculate the coupling loss factors, and
- Algorithms to calculate the power input corresponding to the excitations applied in the system.

If available, user defined value can be used to override any of the parameters calculated by the software.

The SEA equations are assembled using all the calculated parameters and solved to compute the energy of each subsystem for all the frequency bands. With these results, we can analyze the energy flow within a system starting from the excitation to the Sound Pressure Level (SPL) at the target locations.

**2.2 Model Creation Process**

The modeling process in VA One starts from a CAD model or a Finite Element (FE) model. Both were available at the start of this project and used as support to create the structural SEA panels. Generally, the structural panels are only created for half the model, either driver or passenger side. The sub-structuring into multiple panels is defined by the requirement for homogeneity for each panel and the discontinuity in the system. As a general guideline, we try to avoid small subsystems which may not verify the requirement of at least 3 modes in band in order for the SEA assumptions to be valid.

Once created, the SEA structural subsystems are assigned a construction (uniform, composite, sandwich or general laminate) and corresponding material properties. Additional dummy panels are created at the symmetry plane and outside the vehicle; they will be used to partition the volume into multiple acoustic cavities. Most of the panels are flat plates and singly curved shell or doubly curve shells are used to account for the curvature of panels like windshield or the wheel well. For this model, we also accounted for the ribbing on the main structural panels by using stiffness multipliers. These stiffness multipliers were calculated from local FE models by fitting the SEA modal densities to the one from the component level FE model.

Once all the structural panels are created, utility scripts are run to generate all the acoustic cavities. Typically, the interior volume is divided into multiple cavities for each seat row into a headspace and a waist space. One or several layers of cavities are added outside the cab to model the sound propagation around the vehicles. The free faces of the external cavities are connected to SEA Semi-Infinite Fluids (SIF). A SIF is an energy sink for the acoustic energy and can then represent the noise propagation in free-field.

Acoustic Cavities are also created inside the pillars to model the noise transfer path.

The SEA half model of the Truck including the SIFs is represented in Fig. 3. The internal and external cavities are represented respectively in Fig. 4 and in Fig. 5

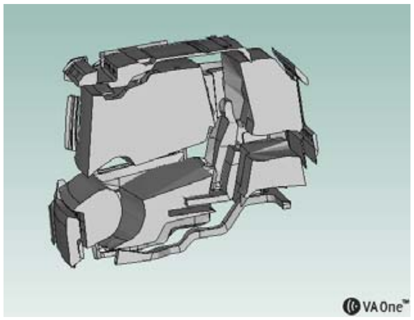

Fig. 4 – Internal Acoustic Cavities

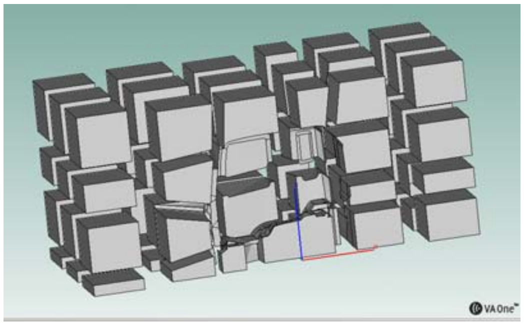

Fig. 5 – External Acoustic Cavities

### 2.3 Noise Control Treatments

The sound package off the truck is modeled using Noise Control Treatments (NCT) in VA One. Each NCT is defined as a stack of materials chosen from foam, fiber, air gap, panel, septum, etc. For each layer, we had to define the corresponding material properties. This consists of the Biot parameters for the foam and fiber, structural properties (Young's modulus, Poison ratio and density) for panel, density for septum and air properties (density and speed of sound) for gaps. The damping loss factor must also be defined for all the materials. Fig. 6 represents a dialog to define a NCT.

The NCTs are applied on the structural panels, the user defines on which side they are applied and what is the percentage of coverage. Multiple NCTs can be applied to a single panel with different percentage of coverage,

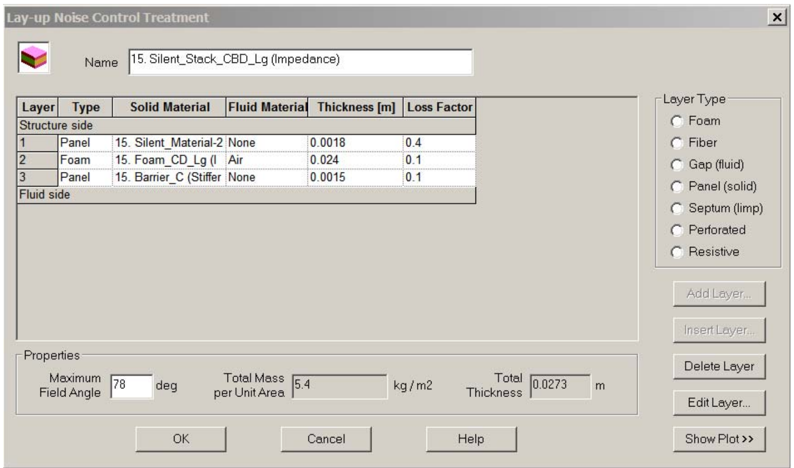


Fig. 6 – NCT Definition

VA One uses a transfer matrix method approach to model the effect of the trim including the added mass, stiffness and damping on the structural panels and the absorption for the acoustic cavities.

## 2.4 Hybrid FE-SEA Updates

The Truck SEA model also accounted for the pass-throughs since they can become a dominant path and they need to be accounted for when evaluating the effect of the sound packaging. The software provides a general formulation for different shape of pass-throughs or leaks that can be added to an area junction between a panel and 2 acoustic cavities. If measurement data are available (i.e. transmission loss and surface area), they can be entered as a user defined leak. In this project, CAD data of grommets were available and the transmission loss of the pass-throughs was estimated analytically. We developed component level models of the grommets and calculated the corresponding transmission loss. We then used this information to update the leaks information for the corresponding junctions in the cab SEA model. Fig. 7 represents one of the grommet component level models developed. The model includes a structural FE model of the grommet coupled to a FE acoustic cavity representing the air around the grommet. The front face of the acoustic cavity and the back face of the grommet are connected to SEA semi-infinite fluids and the excitation is applied using a diffuse acoustic field on the acoustic face. The transmission loss is calculated using the power input from the diffuse acoustic field to the FE acoustic cavity and the power radiated into the semi-infinite fluid by the grommet structural FE model.

Additional updates were applied to the SEA model to account for local geometry of the panels. SEA panels have an idealized geometry, e.g. flat panel or simply curve shell, for which a formulations are available to calculate the modal densities and the number of modes in band. In practice, the actual geometry of the panel is more complex and the panel may have an irregular ribbing pattern or a complex shape that will deviate from the standard SEA geometry. To cope with this, we built local FE models of the main structural panels and we calculated the number of modes in band from the FE model. We then apply stiffness multipliers on the SEA panel to adjust the modal densities to better represent the FE results. Fig. 8 illustrates the results for the roof panel. We represent the number of modes in band as a function of the frequency for the FE panel, the original SEA model and the updated SEA model.

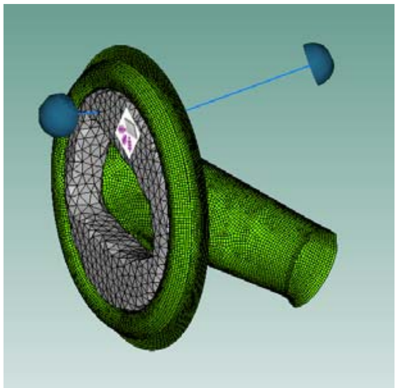

Fig. 7 – Grommet Hybrid FE-SEA Model

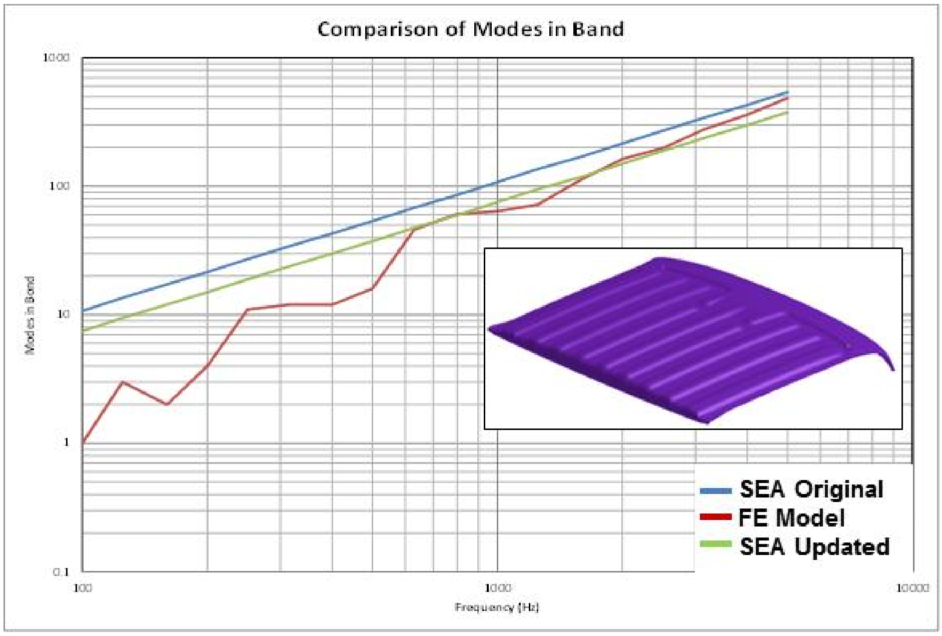


Fig. 8 – Modes in Band for the Roof Panel

## 3 NOISE AND VIBRATION TESTING

The objective in this project is to simulate the noise transfer path from the external sources such as the engine, intake and exhaust in order to optimize the sound package. As part of the project, a comprehensive noise and vibration test and analysis campaign has been performed at the Application Research Center by the team from Sound Answers Inc. The measurements were performed on the road and on the 4-wheel drive chassis dynamometer – See Fig. 9. The objectives of the measurements were to develop a Source-Path-Contribution (SPC) model to identify the relative contributions to the interior noise of each structural and airbone noise transfer path. Additional measurements were also performed to characterize the properties and the performances of the sound package based on noise reduction measurements, tear down tests

(RT60), impedance tube and absorption measurements. The extensive test results provided all the data needed to define the noise sources in the simulation model and to validate the simulation results.

Fig. 9 – Test Setup on the 4 Wheel Drive Chassis Dyno

From the SPC model developed by Sound Answers, we extracted equivalent sound power frequency curve for each of the airbone noise sources and we modeled them in the SEA model as direct power inputs in the corresponding external cavities. The direct measurements results include the contribution of both the airbone and structure-borne sources while the SEA simulation results only account for the airborne sources. Sound Answers extracted from the SPC model the relative contribution of the airbone and structure-borne sources. Fig. 10 represents the sound pressure level measured at the driver hear position in 3$^{rd}$ octave bands. The plot also includes the contribution of the airborne and structure-borne sources extracted from the SPC model. These results show that the airbone sources can be neglected below approximately 200Hz, while the structure-borne noise sources do not contribute significantly to the sound pressure level above 2 kHz. Between these 2 frequencies, both the airbone and structure-borne noise sources are contributing to the total sound pressure level.

In order to develop meaningful comparisons, Sound Answers extracted the total contribution of the airborne noise sources and this served as reference for the validation of the SEA simulation results.

## 4 RESULTS AND VALIDATION

As a first step in the model validation process, we compared the noise reduction predicted by the model against the reference measurement data. Noise sources were placed sequentially around the vehicle and the sound pressure level was measured at the driver or passenger headspace and at the source location. The difference of the dB levels gives the noise reduction.

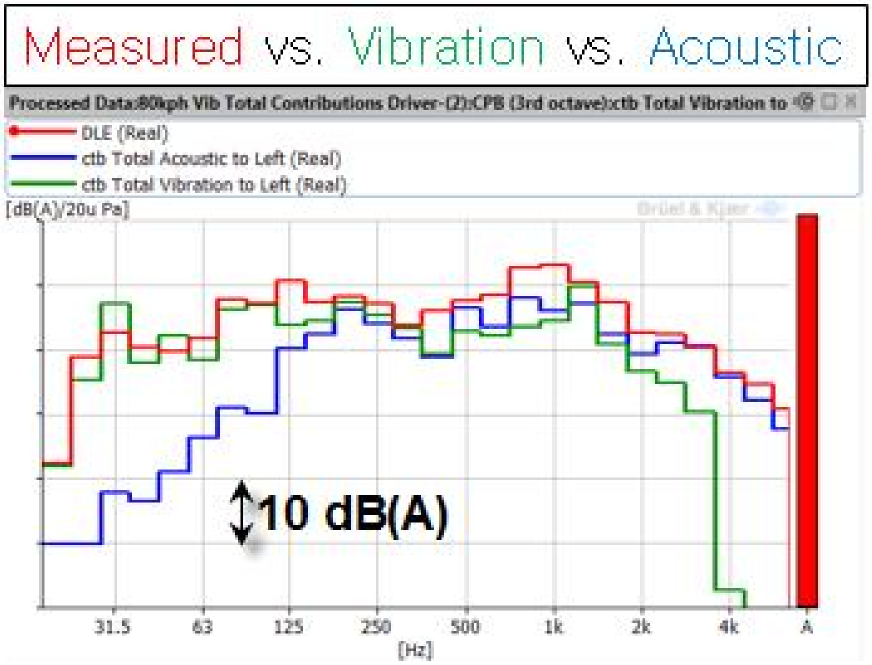


Fig. 10 – Measured SPL, Airborne and Structure-Borne Contributions

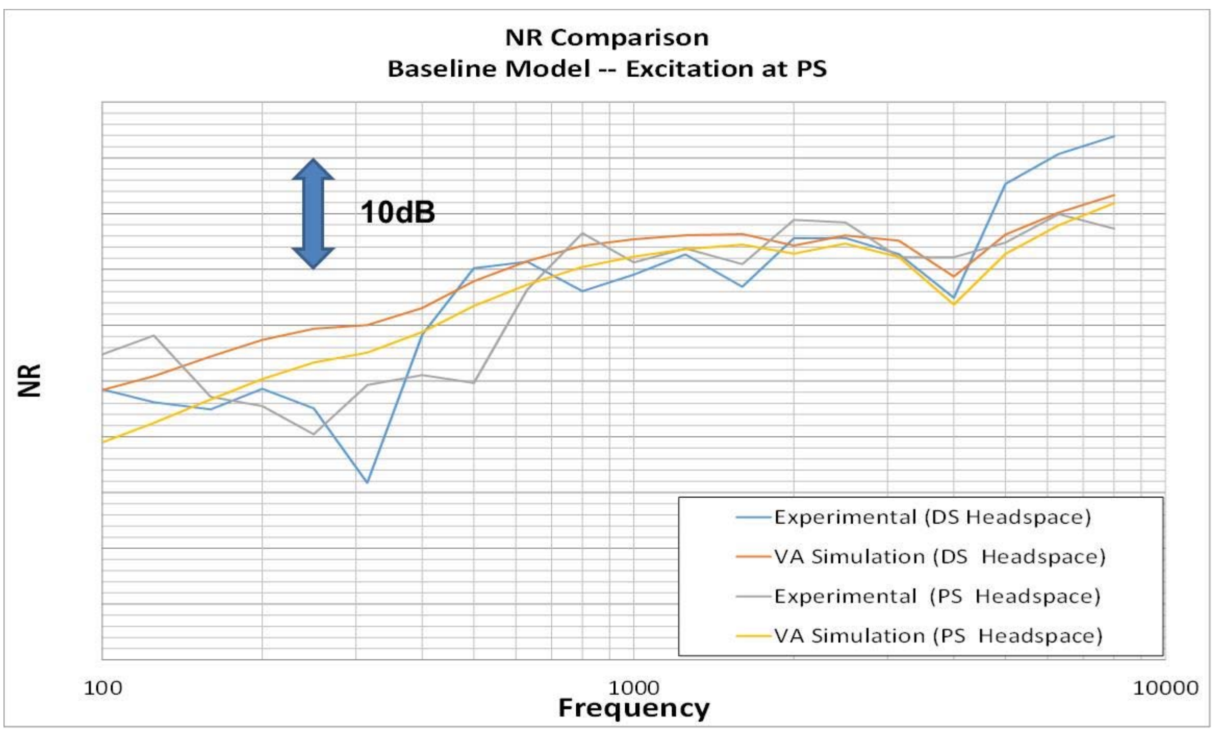


Fig. 11 – Noise Reduction Results: Simulation vs. Test

An example of simulation results and how they compare against the reference measurement data is given in Fig. 11. These results correspond to the test configuration with a source outside the vehicle at the passenger side. These results are representative of the overall correlation; they show that the model reproduces the trend observed in the measurements quite accurately starting from 300Hz. The results below 300Hz were not considered as it is difficult to meet the SEA requirements of at least 3 modes per band but also, the relative contribution of the airbone noise sources may not be relevant.

In the second step in the validation process, we compared the simulation results against operational measurements. During the testing, multiple conditions were measured including idle, wide open throttle for multiple gears, constant speed (60 and 80kph e.g.). Only the steady-state conditions were considered for the simulation model.

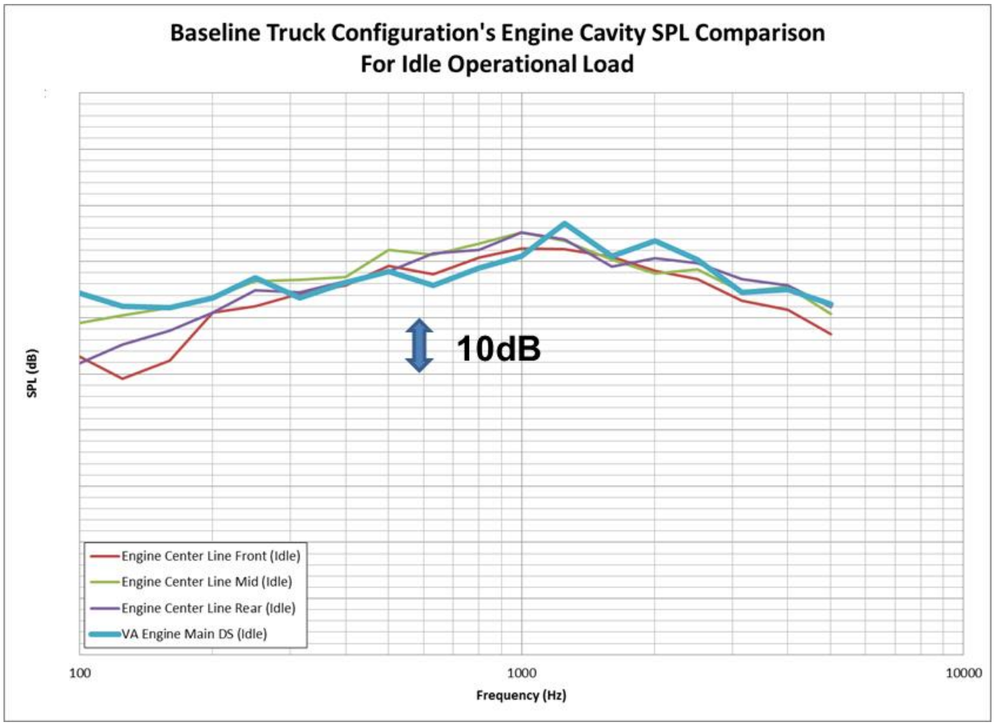


Fig. 12 – Idle Load Case Validation

Fig. 12 shows how the predicted sound pressure level at the drive head space compares against the measurement results for the idle load case. As explained earlier, the measurement results correspond to the contribution of all the airborne noise sources. We observed a very good correlation between the simulation prediction and the reference measurement data. With this, we confirmed that the model is certainly adequate to investigate the effect of design changes in the sound package in order to optimize the noise performance and balance that with the other requirements for weight and costs.

At this point, we started to exercise the model to get additional insight to complement the test and analysis results. We identified for each load case and each frequency the dominant sources and dominant paths. We also quantified the importance of each leaks.

Fig. 13 gives an example of the post-processing of the simulation results. We represent the energy flow for the idle load case in a single frequency band. The color of interior cavities is linked to their level of energy and the arrows represent the power flow through each visible junction. We clearly see that at this frequency, the noise transfer path is through the front floor of the cab. If the energy in this band is significant and need to be reduced, we get then useful information to define a list of potential countermeasures or simply rule out certain design alternatives. For example, we assessed the effect of switching the headliner to a more or less absorptive material and increasing the percentage of coverage on the key panels like the floor. We also quantified where it is the most effective to add mass and where we could take off some weight from the sound package. All these results became very valuable when we started the optimization of sound package. The optimization process and results are presented in Jang[5].

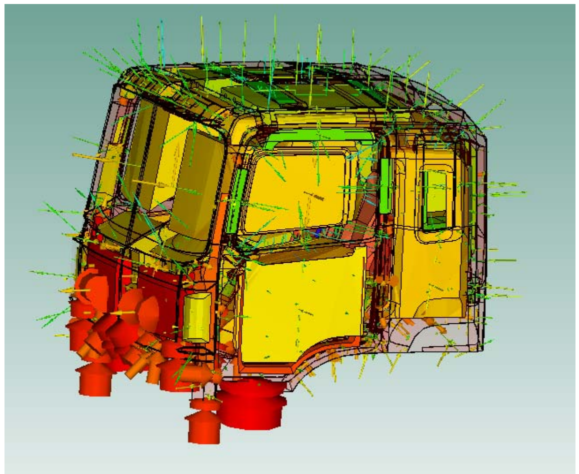

Fig. 13 – Example of Energy Flow

## 5 CONCLUSIONS

We presented in this paper some of the results of a comprehensive project that had as objectives to optimize the sound package of a truck. The project included an extensive noise and vibration test and analysis phase to characterize the performance of the truck and to provide reference data to load the SEA model and validate the simulation results. We briefly presented the model creation process and demonstrated that the simulation results correlate very well with the measurement results. With this, we concluded that the model is suitable to investigate the effect of design modifications of the sound package and to proceed with an optimization of the sound package to balance NVH with cost and weight reduction. The results of the optimization are presented in a separate paper (Jang[5]).

Additional analyses were performed in parallel with the work presented in this paper to build a model for the simulation of the structure-borne path at low and mid-frequency. These models were developed using a Finite-Element approach (Structural and Acoustics) and a coupled Hybrid FE-SEA modeling solution. All these models cover the full frequency range – low, mid and high- for vibro-acoustic simulation and analysis with VA One and account for both the airborne and structure-borne paths.

## 6 ACKNOWLEDGEMENTS

The work presented in this paper is the result of a joint collaboration between Hyundai Motor Company, Sound Answers Inc. and ESI Group and it would not have been possible without the key contribution of all the partners in this project.

## 7 REFERENCES

1. Lyon, R. H, Maidanik G., "Power flow between linearly coupled oscillators", Journal of the Acoustical Society of America, vol.34, no.5, May 1962, pp.623-639.

2. Lyon, R. H., Dejong, R. C., Theory and Application of Statistical Energy Analysis, Second Edition, Boston Butterworth-Heinemann, c1995.

3. Shorter, P. J., Gardner, B., "Statistical Approaches to Structural-Acoustics", 45th AIAA/ASME/ASCE/AHS/ASC Structures, Structural Dynamics & Materials Conference. April 2004.

4. VA One 2014.5 User Guide, ESI-Group.

5. Y. S. Jang, H. Dande, R. Hadjit and E. Frank, "Optimization of the Sound Package of a Truck using Statistical Energy Analysis", Proceedings of InterNOISE 2015.