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Experience Using a Sound Intensity Technique to Determine In-Cab Sound Paths and Treatments

Charles T. Moritz Blachford Acoustics Laboratory, 1445 Powis Road West Chicago, IL 60185, USA

Abstract

In 2002, the author presented a technique for determining the sound paths into a large vehicle cab through the use of sound intensity [1]. With this technique, the interior cab surface is divided into 20 to 30 surfaces and the sound intensity radiated from each of the surfaces is measured when the vehicle is operating at a steady state condition. Then, knowing the area of each surface, the cab room constant and the distance to the receiver, the sound pressure level can be summed at any point inside the cab and the locations contributing to the overall sound level can be rank ordered. Once this information is acquired and a model of the sound paths developed, treatments to the various sound paths can be developed and their performance at the receiver predicted. This paper provides an update to the measurement technique, describes how large component tests, (front of dash and floor system bucks) are integrated into the model, and provides a case study illustrating the models use in solving vehicle noise control problems.

1. INTRODUCTION

The use of sound intensity measurements to determine the sound paths into a large vehicle cab has been in use for several years [1]. With this procedure, the cab interior is typically divided into finite surfaces, and each of the dominant sound paths (floor, engine cover, dash, etc.) are then treated with thick, heavy, noise control treatments. Foam and barrier composites with a 25 mm foam layer and 9.8 kg/m² acoustical barrier are common for this application. With the treatments in place, the sound intensity level of each of the untreated surfaces is measured with the vehicle operating at a steady state condition. Then, one by one, each of the treatments is removed, the sound intensity is measured, and the treatment replaced. The sound power level radiated from each of the interior surfaces is then calculated according to equation 1.

Sound Power (L_w) = Positive Sound Intensity (L_l) + 10^{*}log₁₀ (Surface Area) (1)

Modeling the cab interior as a small room, the sound power level from each surface is then propagated to each receiver location using equation 2.

$$Lp_{i} = Lw_{i} + 10\log_{10}\left[\frac{Q}{4r^{2}\pi} + \frac{4}{A}\right] + K$$
(2)

where: Lw_i = the sound power level from each surface

Lp_i = the sound pressure level at the receiver

r = the distance from the measurement area to the receiver

A = the total sound absorption inside the cab

Q = the directivity factor (in this case Q = 2)

K = a correction factor

While the negative sound intensity is useful for understanding the sound field inside the cab, we are only interested in the sound that propagates from the measurement surfaces to the receiver(s). Therefore, only the positive sound intensity is accounted for in equations 1 and 2.

To determine the sound absorption inside the cab, the reverberation time (T_{60}) is measured and the total absorption is calculated according to equation 3.

$$A = \frac{0.161 * V}{T_{60}} \tag{3}$$

where: V is the interior cab volume

The correction factor K is measured using a small point source loudspeaker of known sound power ($Lw_{speaker}$). The source is placed near each of the measurement surfaces, the sound pressure level at the receiver ($Lp_{receiver}$) is measured, and the correction factor calculated from equation 4.

$$K = Lp_{receiver} - Lw_{speaker} - 10log_{10} \left[\frac{Q}{4r^2 \pi} + \frac{4}{A} \right]$$
(4)

The sound pressure level from each surface is then logarithmically summed to determine the level at the receiver. Results from a typical large vehicle cab are shown in Figure 1. The solid black line indicates the predicted sound pressure level, and the solid red line indicates the average measured level. Error bars are included to show variation in the measured in-cab level.



Figure 1. Typical One-Third Octave Band Model from Sound Intensity Levels

There is good agreement over most of the frequency range. Even though the cab measurements were performed with both 12 mm and 50 mm microphone spacing, sound intensity levels below 125 Hz are difficult to quantify due to the typically high pressure-intensity index values.

With an engineering model describing the sound level in terms of one-third octave band levels, a rank order of the contributions from each of the surfaces can be created based on metrics such as the A-weighted sound level, Speech Interference Level, or the Articulation Index. An example rank order based on the A-weighted level is shown in Table 1 for this same cab.

Cab Surface	Contribution to the A-weighted Sound Level		
Windshield	26 %		
Driver Footwell	14 %		
Driver Instrument Panel	13 %		
Engine Cover (Doghouse)	11 %		
Driver Top of Dash	8 %		
Driver side of Floor	7 %		
Passenger Top of Dash	4 %		
Shift Boot	4 %		
Passenger Instrument Panel	2 %		
Lower Section of Passenger Door	2 %		
Rear Wall of Sleeper	2 %		
Driver Side Door Window	2 %		

Table 1. Contribution to the A-Weighted Sound Level at the Driver

2. TREATMENT DEVELOPMENT

Having information on both the sound power level radiated by each the cab surfaces and the transfer functions for each of the surfaces to the receiver(s) allows examination of the effects of reducing the sound power level of the surfaces or modification of the transfer path. Practical examples would include trade off studies between noise control materials of various weights or determining the effect of increasing the sound absorption in the cab. The model also allows the use of buck studies to optimize treatments and to quickly determine their effect on the sound level at the receiver.

In the example provided in Figure 1 and Table 1, the highest sound levels at the receiver are due to sound from the windshield and front of dash areas. Although it is likely that modifications to the glass will be required to obtain significant reduction in the sound level, some noise reduction is possible through treatment to the front of dash area (driver footwell, instrument panel, engine cover, and top of dash). Treatments for the front of dash are developed by installing a cab in a test section between a reverberation room and a hemi-anechoic room. The engine side of the cab is installed facing the reverberation room and the cab interior is located inside the hemi-anechoic chamber. The sound transmission loss of the front of dash areas are measured utilizing similar surfaces as were used in the full vehicle test. Then alternate treatments are installed and the sound transmission loss for each of the surfaces measured. The differences between the baseline and the alternate treatments are then input back into the full vehicle model and the change in sound level at the receiver is evaluated. In a similar manner, treatments to other areas of the cab, such as front or side glass, floor systems, roofs or cab walls can be evaluated and the effect at the receiver predicted.

Since it is common for large vehicle manufacturers to only update the cab interior rather than to redesign the entire cab, design changes can easily be evaluated in the buck stage without the need of a full, complete vehicle.

3. CASE STUDY

In this case study, we present the results of our measurements and modeling in a Class 8 truck cab. The purpose of the study was to develop treatments reducing the sound level at the driver by 3 dBA while meeting customer specified cost targets. Baseline testing was performed under several operating conditions on road, and inside a hemi-anechoic chamber on a chassis dynamometer. Predicted and measured results are provided in Figure 2 with the contribution ranking in Table 2 for the truck operating at 80 km/h, 300 kW engine load, on the dynamometer. Note that the rank order for this truck is significantly different than for the Cab presented in Figure 1 and Table 1.



Figure 2. Modeling Results for Class 8 Truck Example

For this cab, the dominant sound paths included the roof (28%), floor (17%), and the front of dash area (15%). Additional measurement indicated the roof path was primarily structure borne so damping treatments were recommended. Component level sound transmission loss tests (buck tests) were performed to develop treatment recommendations for the floor and front of dash.

The floor component (buck) tests were performed in the horizontal test fixture of the Blachford Acoustics laboratory, shown in Figure 3. Test samples, 1.2m x 1.2m, were placed between a reverberation chamber and a hemi-anechoic receiving room and the sound

transmission loss of the floor system (metal substrate simulating the truck floor and floormat) were measured according to SAE J-1400[2]. It is important that the samples are evaluated on a simulated floor since the rank ordering of various floormat constructions can be significantly different depending on the test substrate [3].

Cab Surface	Contribution to the A-weighted Sound Level		
Cab Roof	28 %		
Passenger side of Floor	10 %		
Passenger Windshield	7 %		
Driver side of Floor	6 %		
Driver Windshield	6 %		
Engine Cover	5 %		
Driver Footwell	5 %		
Driver side Top of Dash	4 %		
Shift Boot	2 %		
Top half of Cab back Wall	2 %		
Center Section of Floor	2 %		

Table 2.	Contributors to the	A-Weighted Soun	d Level at the Dri	ver for Example	Class 8 Truck
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Figure 3. Horizontal Test Fixture

Measured sound transmission loss (STL) values were then compared to the baseline configuration and the predicted sound pressure level spectrum at the driver from only the floor path, shown in Figures 4 and 5. The floormat construction providing the greatest increase in STL over the baseline case at frequencies most dominant or considered to be most annoying was selected for further in-truck testing. In a typical study, decoupler type, barrier weight, damping, total thickness, and cost are all evaluated. For this project, the sound from the floor was most dominant in the 400 and 630 Hz one-third octave bands. A floor system utilizing damping and





Figure 4. Floor System Noise Reduction Test Results



Figure 5. Predicted Sound Level Contribution from Only the Floor Path

Front of dash buck testing was performed in the 2.4 m x 2.4 m front of dash buck test fixture at the Blachford Acoustics Laboratory. A truck cab was placed between a reverberation room and hemi-anechoic room with the engine side of the cab facing the reverberation room as shown in Figure 6.



Figure 6. Front of Dash Test Fixture

A filler wall was constructed so that sound can only enter the cab through the area of interest. The sound intensity level is then measured on the interior surfaces of the cab in areas similar to those measured in the full truck. Treatments are then evaluated and the noise reduction input into the model. Baseline and treated sound intensity color contours and surface sound power levels for this cab are provided in Figure 7. Higher intensity levels are indicated by white, yellow and orange with lower levels indicated by green, blue and black.



Figure 7. Front of Dash Buck Results, 100-4000 Hz Frequency Range

Based on the floor and front of dash studies, recommendations were made to damp areas of the cab structure, increase the weight of the floor system, and apply decoupled barrier materials to the firewall, engine cover, and rear walls of the cab. Modeling results predicted the recommended treatments would provide a 3 dBA reduction with a below target cost. Development parts were then created an installed in a test vehicle. Measured and predicted levels are shown in Figure 8. There is excellent agreement between the measured and predicted results with slightly greater noise reduction values measured (3.8 dBA) than originally predicted. The noise control package has been introduced into production with excellent customer response.



Figure 8. Comparison between Predicted and Measured Results, 80 km/h 300 kW on Dynamometer

4. CONCLUSION

The use of sound intensity measurements to determine sound paths and to evaluate noise control treatments has been demonstrated. Although excellent agreement between predicted and measured results has been obtained, the technique works best when airborne sound is dominant. Future efforts are focused on developing methods to separate the contributions from airborne and structure-borne sound and to improve our prediction of the noise reduction due to damping.

REFERENCES

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