



# Development and validation of spreadsheet tools for noise path characterization

D. W. Herrin<sup>1</sup>

Srinivasa Ippili

Keyu Chen

Xin Hua

University of Kentucky

Lexington, KY

Bryan Beamer<sup>2</sup>

Centers for Disease Control and Prevention

National Institute of Occupational Safety and Health

Cincinnati, OH

David Copley<sup>3</sup>

Caterpillar Corp.

Mossville, IL

## ABSTRACT

**Machinery noise is a major issue in many industries. Some industries, like aerospace and automotive, are well staffed and are equipped to consider noise even in the early product development stages. However, noise control practice is inadequate in many industrial sectors including textile, paper, metal fabrication, and food preparation. The objective of this work is to develop a simple spreadsheet tool that can be used to predict the effectiveness of common noise control measures. Though some basic noise control knowledge is needed, the user need not be a noise control expert. The methods and relevant theory are described, and the model is tested on several examples.**

## 1. INTRODUCTION

Many industries are ill equipped to consider noise control in the early design stages. This is especially the case for manufacturing and food preparation equipment. Workers may be exposed to higher noise levels than if noise control had been considered in the design stages. However, engineers designing the equipment and the in-plant personnel have little noise control experience.

The intent of this work is to acquaint the reader with simple room acoustics models supplemented with empirical data first suggested by Nickolay Ivanov and Gennady Kurtsev over 25 years ago [1].

---

<sup>1</sup> dherrin@engr.uky.edu

<sup>2</sup> zmy4@cdc.gov

<sup>3</sup> copley\_david\_c@cat.com

This approach has been utilized extensively for heavy equipment by Ivanov and Copley [2-11]. The models are appealing because they can be easily implemented into spreadsheet form and are simple enough to be used by consultants and engineers with some noise control experience. Moreover, users do not need expensive measurement equipment to put the approach into practice.

The model employs a source-path-receiver energy accounting approach. General forms for common paths are used as approximations, but it is preferable that each path be validated experimentally and adjusted for a given application. Moreover, the model can be incrementally improved over time, and more sophisticated simulation or experimental models can be integrated into the spreadsheet framework.

The model is statistical in nature and disregards the wave nature of sound except for some correction terms. The assumptions that the model is based on are as follows.

1. Sound sources are incoherent, and the acoustic signals are wide band. Sound sources distributed at distances greater than  $\lambda/6$  may be considered incoherent where  $\lambda$  is the acoustic wavelength [12].
2. Sound fields in closed spaces are quasi-diffuse. As an approximation, the sound field in a closed space may be considered diffuse at frequencies above that of the 10th acoustic mode [12].
3. Sound pressure at any specific point is determined by the energy summation principle.
4. Resonance phenomena are ignored as a rule.
5. Sources generate sound fields which are idealized as spherical, cylindrical, or plane wave.
6. Sources in closed spaces are assumed to be omni-directional.
7. Closed spaces are characterized by an average coefficient of sound absorption.
8. The ratio of the maximum to the minimum linear dimensions of acoustic spaces generally should not exceed 5. Other simplified models should be considered in that case.

In the sections that follow the model will be described and then tested using several examples.

## 2. DESCRIPTION OF THE MODEL

The models suggested by Ivanov and Kurtsev [1] are simple expansions of room acoustics theory which is reviewed in brief. The sound pressure level in a room is a combination of the direct and reverberant fields. The direct field is that produced when the source is placed in an anechoic environment (no reflections) whereas the reverberant field is due to the reflected sound from walls, floor, and ceiling. According to room acoustics theory [13], the sound pressure level ( $L_p$ ) at a given position can be expressed as

$$L_p = L_W + 10 \log_{10} \left( \frac{\Gamma}{4\pi r^2} + \frac{4}{R_r} \right) \quad (1)$$

where  $L_W$  is the sound power level of the source,  $r$  is the distance from the source, and  $R_r$  is the room constant.  $\Gamma$  considers if the source is close to a single plane or multiple planes.  $\Gamma = 1$  if the source is in a free field.  $\Gamma=2$  if the source is located close to a single plane (i.e. sitting on the ground).  $\Gamma = 4$  if the source is adjacent to two planes (the edge of a room).  $\Gamma=8$  if the source is close to three planes (a corner of a room). The model can be tuned to match experimental data by adjusting  $\Gamma$ .

The room constant is expressed as

$$R_r = \frac{\langle \alpha_d \rangle S}{1 - \langle \alpha_d \rangle} \quad (2)$$

where  $S$  is the surface area of the space and  $\langle \alpha_d \rangle$  is the spatially averaged sound absorption coefficient in the room. The spatially averaged sound absorption can be determined using

$$S\langle\alpha_d\rangle = S_1\alpha_1 + S_2\alpha_2 + \cdots + S_n\alpha_n \quad (3)$$

where  $S_i$  and  $\alpha_i$  are subareas and respective sound absorption coefficients, and  $n$  is the total number of subareas. Sound absorption coefficients can be found in textbooks for some standard materials.

The sound transmission coefficient through a wall or panel can be expressed as

$$\tau = \left(\frac{2\rho c S_p}{m\omega}\right)^2 \quad (4)$$

where  $\rho$  and  $c$  are the density and speed of sound or air,  $\omega$  is the angular frequency in rad/s, and  $m$  is the mass of the panel. For composite panels or walls manufactured from different materials, the composite transfer function  $\tau_{eq}$  is expressed as

$$S_p\tau_{eq} = S_1\tau_1 + S_2\tau_2 + \cdots + S_n\tau_n \quad (5)$$

where  $S_i$  and  $\tau_i$  are subpanel areas and transmission coefficients respectively.  $n$  is the total number of subpanels. The sound isolation for normal incidence can be expressed in terms of the transmission coefficient as

$$SI_{\perp} = 10 \log_{10} \left( \frac{1}{\tau_{eq}} \right) \quad (6)$$

and the field incident transmission loss can be written as

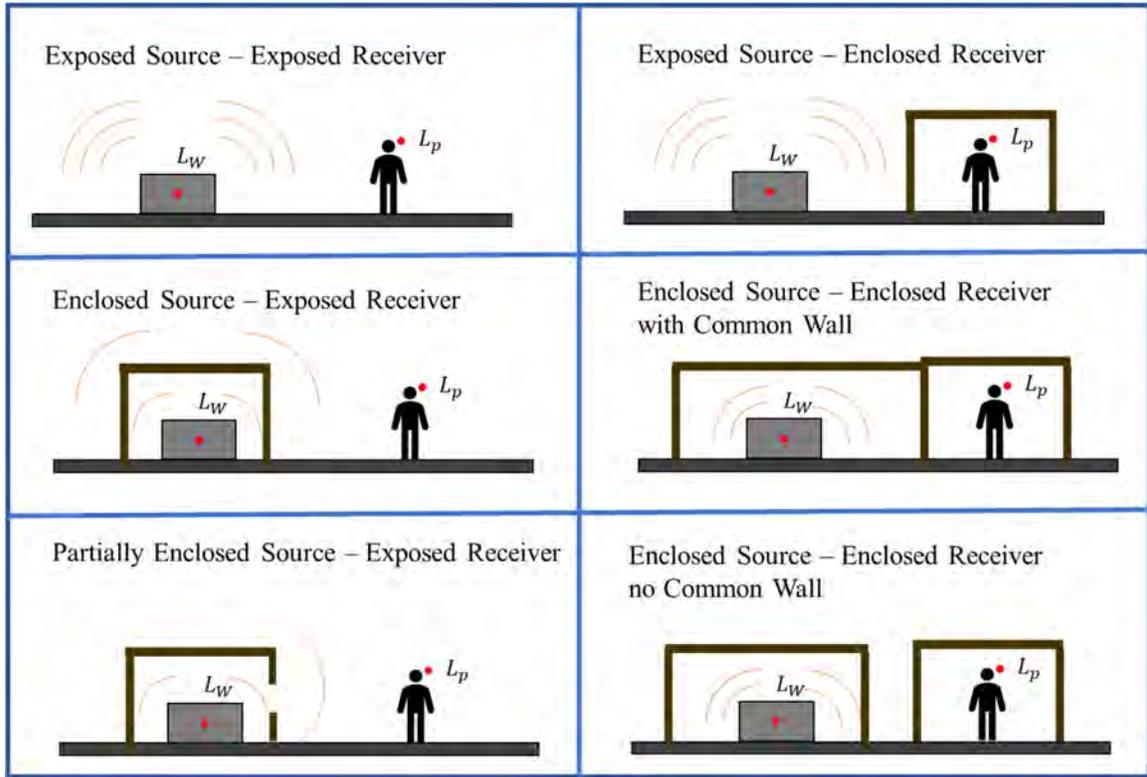
$$SI_{diff} = SI_{\perp} - 5 \quad (7)$$

where either Equation (6) or Equation (7) can be used in the equations which follow depending on which is more appropriate for the case being considered.

### 3. SOUND PROPAGATION MODELS

Sound propagation models are detailed for each of the 6 cases shown in Table 1. Additional models have been developed by Ivanov and Kurtsev [1]. These are the most common and deal with many common cases that encountered in industries lacking NVH expertise.

**Table 1** Sound propagation cases.



### 3.1. Exposed Source – Exposed Receiver

The case of an exposed source and receiver is considered first. The sound pressure level at the receiver position can be expressed as

$$L_p = L_W + 10 \log_{10} \left( \frac{\Gamma \chi}{4\pi r^2} \right) + DI - \beta \quad (8)$$

where  $\chi$  is a near field correction. The near field correction can be expressed as

$$\chi = 2.5 + 1.5 \cos \left( \frac{\pi}{2} \frac{r}{l_{max}} \right) \quad (9)$$

where  $l_{max}$  is the maximum dimension of the source if  $r/l_{max} < 2$ . If  $r/l_{max} > 2$ , then  $\chi = 1$ . Equation (8) is based on the correction curve developed by Ivanov [1].  $DI$  is a directivity index. If the sound source is aimed towards or away from the receiver,  $DI = 4$  dB or  $DI = -4$  dB respectively.  $\beta$  is a source shielding term.  $\beta = 5$  dB if the source is partially shielded and  $\beta = 8$  dB if the source is fully shielded. Notice that the second term on the right-hand side is a direct field term, and the third and fourth terms are adjustments.

### 3.2. Enclosed Source – Exposed Receiver

The case of an enclosed source and exposed receiver is considered. The sound pressure level at the receiver can be expressed as

$$L_p = L_W + 10 \log_{10} \left( \frac{\Gamma_s \chi_{se}}{4\pi r_{se}^2} + \frac{4\psi_{se}}{R_r} \right) + SI_{diff} + 10 \log_{10} \left( \frac{\Gamma_e \chi_{er}}{4\pi r_{er}^2} \right) \quad (10)$$

where  $r_{se}$  and  $r_{er}$  are respective distances from source to enclosure and from enclosure to receiver. The near field correction terms  $\chi_{se}$  and  $\chi_{er}$  are based on the source to enclosure and enclosure to receiver distances respectively.  $R_r$  is the room constant for the enclosure.  $\Gamma_s$  and  $\Gamma_e$  are symmetry terms for the source and enclosure respectively.

$\psi$  is a correction term that accounts for differences from room acoustics theory in the enclosure. If the enclosure has high sound absorption, room acoustics theory will no longer be entirely valid since the sound field inside the room will not be diffuse. This correction term is developed from reference [1] and can be expressed as

$$\psi = e^{\frac{\sqrt{2}}{2}x_\psi} \quad (11)$$

where

$$x_\psi = \frac{\langle \alpha_d \rangle}{1 - \langle \alpha_d \rangle} \quad (12)$$

with  $\langle \alpha_d \rangle$  equal to the spatially averaged sound absorption in the enclosure.

Examining Equation (10), it can be observed that the equation includes the source to enclosure path (the second term on the right hand side), the sound isolation ( $SI_{diff}$ ) of the enclosure, and the enclosure to receiver path (the last term on the right hand side). Also, the enclosure correction term in Equation (11) will usually only change results by a few dB and will be unimportant in many cases.

### 3.3. Partially Enclosed Source – Exposed Receiver

The expression for a partially enclosed source and exposed receiver is similar to that for the fully enclosed source given in Equation (10) except that the panel transmission loss term is replaced by a term to account for the size of the opening. The sound pressure level is expressed as

$$L_p = L_W + 10 \log_{10} \left( \frac{\Gamma_s \chi_{se}}{4\pi r_{se}^2} + \frac{4\psi}{R_r} \right) + 10 \log_{10} \left( \frac{S_o}{S_e} \right) + 10 \log_{10} \left( \frac{\Gamma_e \chi_{er}}{4\pi r_{er}^2} \right) \quad (13)$$

where  $S_o$  is the opening cross-sectional area and  $S_e$  is the surface area of the enclosure itself,  $r_{er}$  is the distance from the opening to the receiver, and  $\chi_{er}$  is the diffuse field violation correction term.

### 3.4. Exposed Source – Enclosed Receiver

The sound pressure level for an exposed source and enclosed receiver can be expressed as

$$L_p = L_W + 10 \log_{10} \left( \frac{\Gamma_s \chi_{se}}{4\pi r_{se}^2} \right) + DI - \beta - SI_\perp - \Lambda + 10 \log_{10} \left( \frac{S_p}{R_r} \right) \quad (14)$$

where the first four terms on the right hand side are identical to Equation (8),  $SI_\perp$  is the sound isolation of the panel for normal incidence,  $S_p$  is the surface area of a panel, and  $R_r$  is the room constant of the enclosure.  $r_{se}$  is the distance from the source to the receiving enclosure and  $\chi_{se}$  is calculated using this distance.  $\Lambda$  is an addition to the panel sound isolation depending on where the receiver panel is located. Values for  $\Lambda$  are defined in Table 1. Equation (14) may be applied panel by panel. However, it can be observed from Table 2 that  $\Lambda$  values are sufficiently large that sound transmission through the panel nearest the receiver should be the most important path unless that panel is significantly more massive than neighbouring panels.

**Table 2** Values for  $\Lambda$  reproduced from [1].

Distance from Source (m)	Receiver Enclosure Panels	Frequency (Hz)							
		63	125	250	500	1000	2000	4000	8000
0.1-2.0	Side	9	9	9	9	9	9	13	17
	Top	5	9	9	12	12	12	15	18
	Rear	11	14	14	14	14	14	17	20
> 2.0	Side	5	7	7	7	7	7	9	9
	Top	6	8	8	8	8	8	10	10
	Rear	5	11	11	13	13	13	18	18

### 3.5. Enclosed Source – Enclosed Receiver with Common Wall

The sound pressure level for an enclosed source and enclosed receiver with a common wall can be expressed as

$$L_p = L_W + 10 \log_{10} \left( \frac{\Gamma_s \chi_{se}}{4\pi r_{se}^2} + \frac{4\psi_{se}}{R_{sou}} \right) - SI_p + 10 \log_{10} \left( \frac{S_p}{R_{rec}} \right) \quad (15)$$

where  $S_p$  is the panel area and  $R_{sou}$  and  $R_{rec}$  are the room constants for the source and receiving enclosures respectively.  $\psi_{sou}$  is the diffuse field violation for the source enclosure. All other terms have been described earlier.

### 3.6. Enclosed Source – Enclosed Receiver without Common Wall

The sound pressure level for an enclosed source and enclosed receiver with no common wall can be expressed as

$$L_p = L_W + 10 \log_{10} \left( \frac{\Gamma \chi_{se}}{4\pi r_{se}^2} + \frac{4\psi_{se}}{R_{sou}} \right) - SI_{se} + 10 \log_{10} \left( \frac{\Gamma \chi_{er}}{4\pi r_{er}^2} \right) - SI_{rec} + 10 \log_{10} \left( \frac{S_p}{R_{rec}} \right) \quad (16)$$

where the subscript *er* refers to the path from source enclosure to receiver enclosure.  $SI_{se}$  and  $SI_{rec}$  refer to the sound isolation of the source and receive enclosures respectively.

### 3.7. Combining Sources

The spreadsheet model developed should be applied path by path. The contributions from each path can be added together to determine the total contribution. Accordingly, the sound pressure level at a given position can be expressed as

$$L_p = 10 \log_{10} \left( \sum_{i=1, N} 10^{\frac{L_{pi}}{10}} \right) \quad (17)$$

where  $L_{pi}$  is the contribution from path *i* assuming there are *N* total paths. Often, the dominant path will be relatively obvious to identify and then it is only a matter of examining the equation for one particular path.

## 4. ENCLOSURE TEST CASES

Several rudimentary measurements were performed using an acoustic source placed inside a 66 cm × 48 cm × 48 cm steel enclosure as the input source. All measurements were performed inside the hemi-anechoic chamber at the University of Kentucky. White noise input was used.

### 4.1. Exposed Source – Exposed Receiver

The case of an exposed source and receiver is considered first. A loudspeaker is placed inside the enclosure with an impedance tube section attached to it. The goal of this test case is to approximate an exhaust source. A photo of the test setup is shown in Figure 1 along with a schematic on the left. Receiver locations are 1.3 m to the right of the source and 2 m behind the source. Equation (8) is used. The sound power at the exit of the tube is measured first and is used as the sound power input to the model.



Figure 1: Schematic showing receiver positions (left) and photo showing test setup (right).

Figures 2 and 3 compared the predicted sound pressure level to that measured directly at positions 1 and 2, respectively. The curves are shown in dB and overall numbers are indicated in dBA. Results are plotted in 1/12<sup>th</sup> octave bands though 1/3 octave, octave, or overall predictions are more likely to be used in practice. Overall results are included in the legends. It can be observed that overall A-weighted sound level predictions are within 1 dBA. There are significant discrepancies at the lower frequencies at Position 2. It is intended that the model be tuned using measurement data. The most straightforward method for doing so is by adjusting the assumption for  $\Gamma$ .

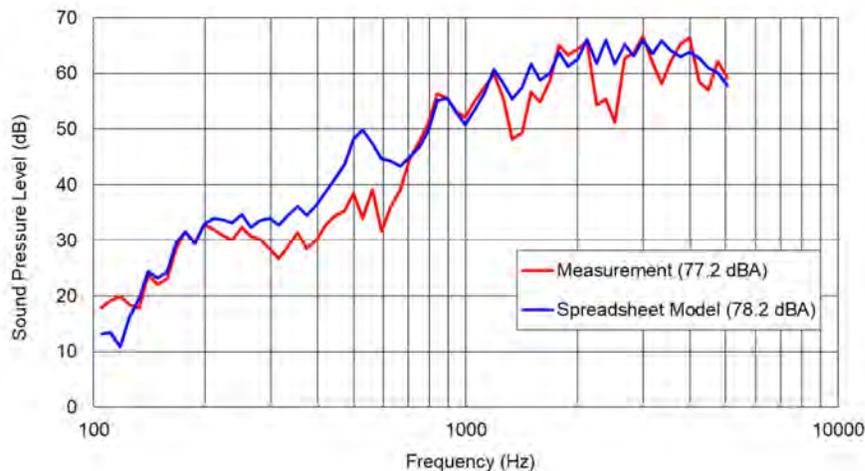


Figure 2: Comparison of measured and predicted sound pressure level at Point 1.

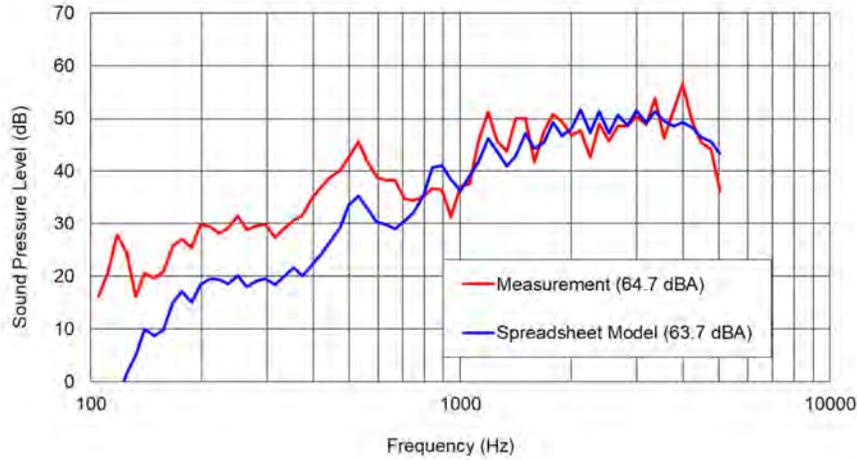


Figure 3: Comparison of measured and predicted sound pressure level at Point 2.

#### 4.2. Enclosed Source – Exposed Receiver

The enclosed source and exposed receiver case was tested by positioning a Brüel & Kjaer omnidirectional source inside the enclosure. The sound power of the source was first measured by sound intensity scanning and then the sound power of the enclosed source was measured. Figure 4 shows a photograph of the test. The enclosure consisted of a single 1.4 mm aluminum panel., and all other panels were 3.2 mm thick steel.

The directly measured sound pressure level is compared to that predicted using the spreadsheet model. Equation (10) is used. Figures 5 and 6 show the sound pressure level results for positions 0.3 m from the 3.2 mm steel and 1.4 mm aluminum panels, respectively. Observe that the A-weighted overall sound pressure levels are both within 3 dBA. Results are acceptable except at the lower frequencies where the enclosure and panel modes significantly affect the radiated sound. In addition, the differences between the two positions are predicted accurately.

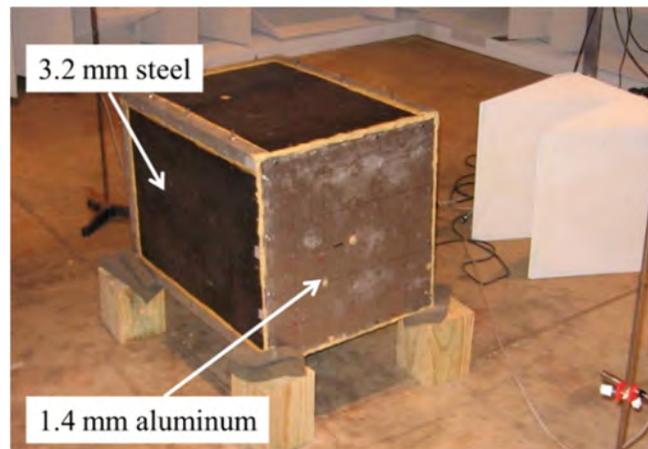


Figure 4: Photograph showing enclosed source and exposed receiver test setup.

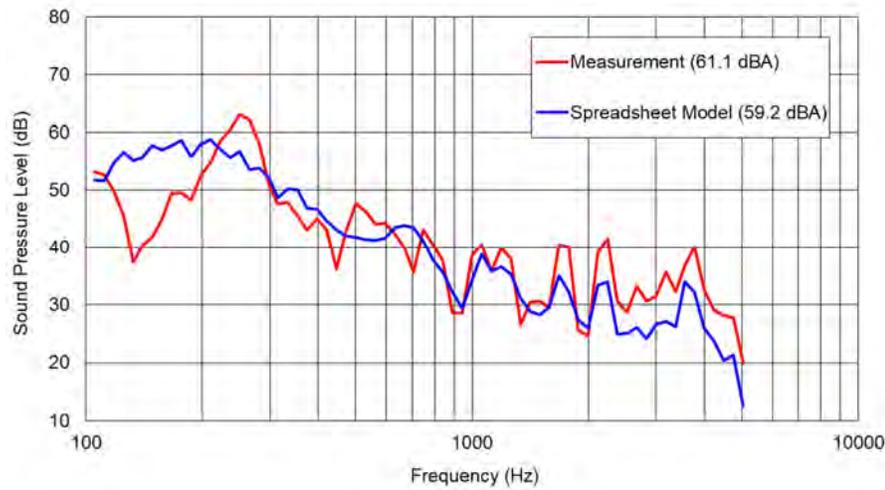


Figure 5: Sound pressure level comparison 0.3 m from 3.2 mm steel panel.

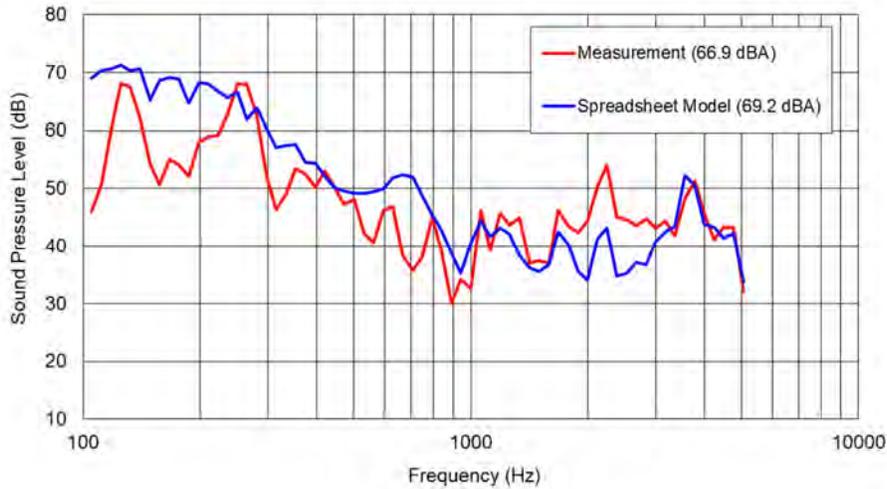


Figure 6: Sound pressure level comparison 0.3 m from 1.4 mm aluminum panel.

#### 4.3. Partially Enclosed Source – Exposed Receiver

The Brüel & Kjaer omnidirectional source was positioned inside the enclosure, and the bottom of the enclosure was removed. The other 5 sides of the enclosure were 3.2 mm thick steel. Accordingly, the transmission through the sides of the enclosure should be negligible compared to that coming from the opening. The inside of the partial enclosure was lined with acoustic foam. Figure 7 shows a photo of the test setup.

The sound pressure level was predicted at 1.5 m from the enclosure using Equation (13) and compared to direct measurement. Results are shown in Figure 8. The distance from the opening to the receiver is approximated by combining the distance from the opening to the floor and the distance from a floor reflection position to the receiver location. The results do not consider the enclosure shielding which would trend the predictions lower. In this case, the model was tuned to correlate well at higher frequencies. The overall dBA sound pressure level predictions are within 5 dBA.

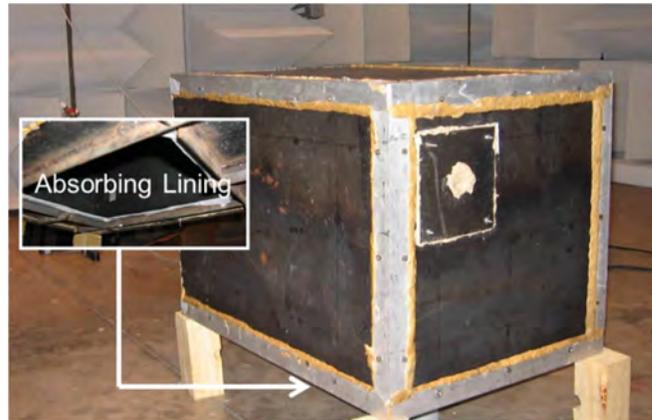


Figure 7: Photograph showing partial enclosure test article.

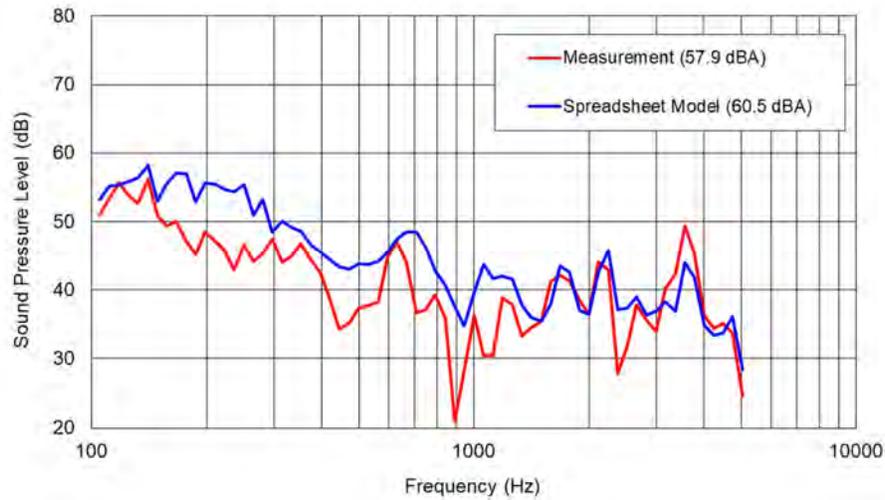


Figure 8: Sound pressure level prediction compared to direct measurement at 1.5 m.

## 5. CONSTRUCTION EQUIPMENT APPLICATIONS

A variation of the spreadsheet model presented in this work has been applied extensively to construction equipment by Ivanov and Copley [3-11]. That body of work is useful for demonstrating how the model can be customized to particular equipment types.

For example, the main noise contributions to the interior cabin noise are illustrated in Figure 9. These include the:

1. Exhaust noise.
2. Intake noise.
3. Diesel noise propagating through the partition between the engine compartment and the cab.
4. Diesel noise propagating through enclosure underneath the opening.
5. Diesel compartment noise propagating through the enclosure panels.
6. Undercarriage noise propagating through the cab panels.
7. Undercarriage noise propagating through the cab floor.
8. Total airborne interior sound field of a tracked dozer.

The sound pressure level from the combined sources can be determined using Equation (17). It can be implied that the main contributions are through the partition between the cab and diesel engine compartment and from the engine noise propagating through the underneath opening. Hence, the noise could be reduced by increasing the sound isolation of the partition and by redirecting the noise from the opening.

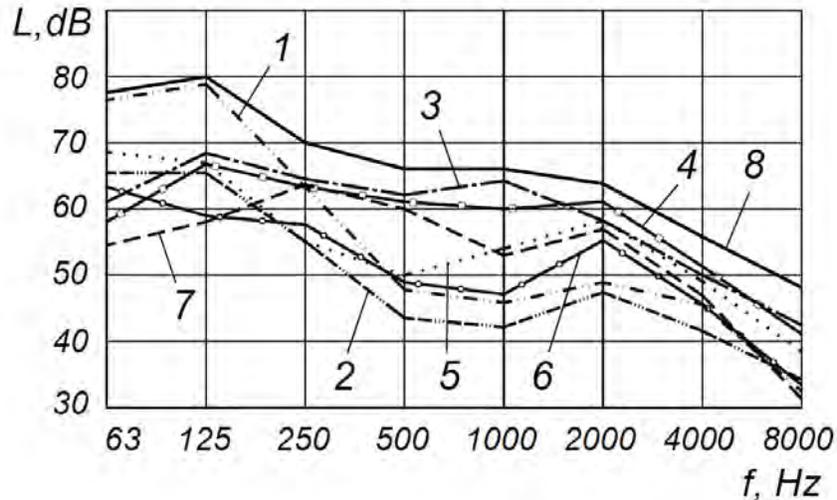


Figure 9: Example of theoretically predicted interior noise spectrum of a tracked dozer [7].

Figure 10 shows a similar prediction for the contributions from the individual components to the sound pressure levels 7.5 m from a dozer tractor. The prediction correlates well with direct measurement, and it can be concluded that the undercarriage noise is the most prominent noise source and should be addressed first. Simple models like this in the early design stages can be used to identify and communicate dominant sources and paths in order to establish component targets and mitigation plans.

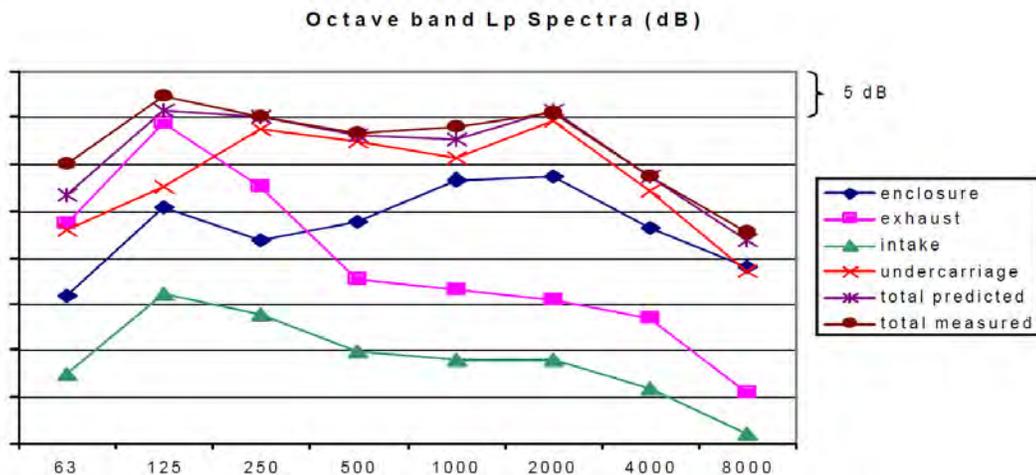


Figure 10: Predicted sound pressure level contributions for dozer tractor at 7.5 m away [3].

## 6. CONCLUSIONS

In summary, equations for simple spreadsheet models to identify the contributions to the sound pressure level have been detailed. Further details are available in Reference [1] and other references in the paper. The spreadsheet model can be easily customized to the equipment of interest, and the model seems ideal for evaluating noise contributions from manufacturing and food processing equipment especially if the noise sources are included. The models can be enhanced to include barriers and structureborne paths. The authors have a spreadsheet that can be shared upon request.

## 7. ACKNOWLEDGEMENTS

We gratefully acknowledge the support of CDC NIOSH for portions of this work. The findings and conclusions in this report are those of the authors and do not necessarily represent the official position of the National Institute for Occupational Safety and Health, Centers for Disease Control and Prevention.

## 8. REFERENCES

- [1] *Noise and Vibration Control in Vehicles*, ed. M. J. Crocker and N. Ivanov, Politekhnik, St. Petersburg, 1993.
- [2] N. Ivanov, Theoretical and practical noise control approaches to vehicle design. *Proc. International Congress of Sound and Vibration*, Lyngby, Denmark, 1999.
- [3] N. Ivanov, D. Kuklin, and D. Copley, Formulation of mathematical models for exterior noise prediction of construction machines. *Proc. Int. Congress of Sound and Vibration*, USA, 2002.
- [4] N. Ivanov and D. Copley, Practical use of noise source contribution prediction methods for effective noise control in construction machines. *Proc. Int. Congress of Sound and Vibration*, Sweden, 2003.
- [5] N. Ivanov and D. Copley, Separation of noise source contributions into exterior and interior sound fields of a vibratory compactor using empirical and experimental data. *Proc. Int. Congress of Sound and Vibration*, Russia, 2004.
- [6] N. Ivanov and M. Drobaha, Requirements for efficient exhaust noise muffler calculation. *Proc. Int. Congress of Sound and Vibration*, Russia, 2004.
- [7] N. Ivanov and D. Copley, Cab noise generation and noise control in construction machinery. *Proc. Int. Congress of Sound and Vibration*, Lisbon, 2005.
- [8] N. Ivanov and D. Copley, Effective exterior noise prediction in earth-moving machines. *Proc. Int. Congress of Sound and Vibration*, Lisbon, 2005.
- [9] N. Ivanov and D. Copley, Investigation of construction machine cab noise generation. *Proc. Int. Congress of Sound and Vibration*, Vienna, 2006.
- [10] N. Ivanov and D. Copley, Structure-borne sound contribution into earth-moving machine cabs, *Proc. Int. Congress of Sound and Vibration*, Australia, 2007.
- [11] N. Ivanov and D. Copley, "Noise and Vibration in Off-Road Vehicle Interiors – Prediction and Control", Chapter 98, *Handbook of Noise and Vibration Control*, ed. M. J. Crocker, Wiley, Hoboken, New Jersey, pp. 1186-1196, 2007.
- [12] U. J. Kurze, Scattering of Sound in Industrial Spaces, *Journal of Sound and Vibration*, Vol. 98, No. 3, pp. 349-364, 1985.
- [13] M. Long, *Architectural Acoustics*, 2nd Edition, Academic Press, 2014.