DESIGN OF PARTIAL ENCLOSURES FOR ACOUSTICAL APPLICATIONS

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ABSTRACT OF THESIS

DESIGN OF PARTIAL ENCLOSURES FOR ACOUSTICAL APPLICATIONS

Enclosures are a very common way to reduce noise emissions from machinery. However, enclosures display complex acoustic behavior that is difficult to predict. The research presented in this thesis uses the boundary element method in order to better understand the acoustic behavior of a partial enclosure. Insertion loss was used as the performance measure and the effect of several design factors on the overall insertion loss was documented. Results indicate that the most important factors affecting enclosure performance are the opening size, amount of absorption, and the source-to-opening distance.

KEYWORDS: Acoustics, Noise Control, Boundary Element Method, Enclosures, Insertion Loss

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March 8, 2006
DESIGN OF PARTIAL ENCLOSURES FOR ACOUSTICAL APPLICATIONS

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Amy Elizabeth Carter

The Graduate School

University of Kentucky

2006
DESIGN OF PARTIAL ENCLOSURES FOR ACOUSTICAL APPLICATIONS

A thesis submitted in partial fulfillment of the requirements for the degree of Master of Science in Mechanical Engineering at the University of Kentucky

By
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Lexington, Kentucky
2006
To Michael
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1.1 Introduction

Noise is one of the most common hazards we face today. It is all around us: in our homes, in our workplaces, in our cars. All too often, its negative effects are ignored. Overexposure to noise can cause immediate symptoms like irritability, stress, and inefficiency in the workplace. The long term effects of noise exposure however, are more daunting. Permanent hearing loss can occur as a result of continued exposure. Once the damage occurs, it is irreversible, but hearing loss is preventable (NIOSH 2005).

In recent years, noise control and hearing loss prevention have become common concerns. Agencies like OSHA began regulating workplace noise in the 1970s and more and more regulations are in place everyday. Many consumer products, including cars, machinery, and office equipment are limited to prescribed noise levels. In fact, even buildings are now subject to code requirements which limit the amount of noise transmitted through their walls (Blanks 1997). Along with this, manufacturers and researchers have become increasingly concerned with better ways of noise reduction and control. The prevention of hearing loss has been named among the 21 priority areas of research in the next century by the National Institute for Occupational Safety and Health (NIOSH 2005).

There are many common ways to reduce noise. Perhaps the best method of reducing noise in a product is to incorporate it into the design process by limiting soured noise through such methods as minimizing input forces, limiting the interaction of moving parts, and using materials with inherent damping. These methods, however, are only useful to an extent and often leave more to be desired. Then designers must rely on other methods to further limit the noise output, particularly treating the noise emitted by the source. Some common methods might include mufflers, barriers, or enclosures. Mufflers are commonly used in exhaust systems, such as in car engines, and they are made up of some combination of absorptive material to dissipate the sound as well as reactive elements that work by reflecting sound waves to create destructive interference of sound waves and effectively “cancel” the noise. Barriers are generally used to block
noise sources that are too large to enclose, such as traffic noise. A barrier is essentially a panel which reflects sound waves and thus reduces the sound that passes to the other side. When a source is completely enclosed by barriers, it becomes an enclosure.

Enclosures are the most commonly used methods of reducing sound transmission from equipment and machines (Beranek 1992). An enclosure is made up simply of barriers surrounding all sides of a noise-emitting object. Usually, the barriers include a rigid outer layer and an inner layer of absorbing material. The outer layer provides stiffness as well as reflecting the impinging sound waves back towards the source while the absorbing material dissipates sound energy. An enclosure may or may not be mechanically connected to the enclosed object. An enclosure that is not connected is designated as *free standing* (Beranek 1992). In addition, an enclosure may be completely sealed or it may have openings, whether intentional, such as for ventilation or equipment function, or unintentional, such as leaks and gaps. Most enclosures contain some openings and, as such, are designated as *partial enclosures*.

This study examines several factors of the geometry and arrangement of enclosures that affect their acoustical performance. The analysis will look at cases with an ideal, theoretical point source and also with a real source. A Cummins B-series diesel engine will be used for the real source. Based on the results derived from these cases, conclusions will be drawn as to how each of the factors studied changes the acoustical behavior of the enclosure, and guidelines will be developed that will help expedite the design process.
1.2 Objectives

This study will use the boundary element method to analyze eight factors that affect enclosure design. The objective will be to understand how each factor affects the performance of partial enclosures.

The factors that will be considered are:

a. Opening Size
b. Opening Location
c. Absorption Coverage
d. Absorption Location
e. Absorption Coefficient
f. Enclosure Size
g. Source Location
h. Velocity Boundary Condition (Input Excitation)

The study will concentrate on the behavior of free-standing, partial enclosures at low frequencies (0 to 1000Hz). Structure-borne noises will not be considered. Although structure-borne noise is important, it is difficult to model and is usually case-dependent. Because of this assumption, the cases studied will not accurately reflect real conditions, but they will better demonstrate the effect of the eight factors under consideration.

1.3 Motivation

Enclosures are common in many industries to help maintain acceptable noise levels. Not only are noise levels strictly regulated, but consumers are looking for increasingly quiet products. Therefore, in order to be competitive, it is necessary for designers to optimize these enclosures. In many cases this is done by building many prototype enclosures and testing each one, using a great deal of resources. The conclusions developed in this study will help designers better understand the acoustics of enclosures to reach a better design with fewer tests.
1.4 **Approach and Justification**

All cases studied will utilize the boundary element method for analysis. The boundary element method, or BEM, is a computer-based numerical analysis program which will divide the surfaces under consideration into discrete boundary elements to obtain a solution. The boundary element method has been widely used in acoustic studies, including those with enclosures, and results compare well with experiments. More information is given on the boundary element method in Chapter 2, and on its previous usage in Chapter 3.

1.5 **Organization**

This thesis is organized into seven chapters, including this introduction. The next chapter, Chapter 2, presents some background information about acoustics, including definitions of common terms and introductory equations. Sound intensity and sound power are discussed, as well as acoustic impedance, decibels, and insertion loss. The boundary element theory is presented as well. Chapter 3 is a review of relevant previous literature pertaining to acoustical enclosures and the boundary element method. The next chapter discusses the setup of the experiments performed in this research. In Chapter 5, the results from all the test cases are presented and discussed. These results are analyzed in Chapter 6. Finally, Chapter 7 concludes this thesis with a summary and some suggestions for future work.
Chapter 2
Background

2.1 Acoustic Waves

Acoustic waves are pressure disturbances that propagate through a compressible fluid, such as air, and are interpreted by the human ear as sound. Normally, these pressure disturbances are very small compared with ambient pressures, but they can be measured using sophisticated microphones. These waves usually propagate uniformly in all directions, unless the wave encounters a difference in impedance. Acoustic waves, just like other mechanical waves, experience reflection, scattering, diffraction, refraction, and interference.

2.1.1 Sound Intensity and Sound Power

When analyzing enclosures, it will be useful to define the sound intensity and sound power of a source. The sound intensity, $I$, at a point is the time average of the instantaneous rate at which work is done by a sound wave as it travels. It is defined as

\[
\vec{I} = \left< \vec{P} \vec{V} \right>_t = \frac{1}{T} \int_0^T \vec{P} \vec{V} \, dt \quad \left( \frac{W}{m^2} \right)
\]  

(2.1)

where $P$ is the complex amplitude of the acoustic pressure, $V$ is complex particle velocity at the point, $T$ is the total time and $t$ is the instantaneous time. The intensity is a vector in the direction of the velocity. For harmonic waves, the intensity can be written as

\[
I = \frac{1}{2} \text{Re}\{P V^*\} \quad \left( \frac{W}{m^2} \right)
\]  

(2.2)
where \( * \) denotes a complex conjugate.

The sound power radiated by a source is defined as the integral of the normal component of the intensity over a surface surrounding the source.

\[
W = \int \tilde{I} \cdot d\tilde{s} \quad (W)
\]  

(2.3)

2.1.2 Acoustic Impedance

The acoustic impedance plays a large role in the propagation of sound waves. By choosing materials of appropriate impedance, engineers can manipulate the sound transmission through a particular path. A wave will tend to continue uninterrupted in its path so long as the acoustical impedance in unchanged. By choosing materials with similar impedance characteristics, engineers can ensure the promotion of wave transmission, such as with ultrasonic testing (Fahy 2001). On the other hand, engineers can also suppress sound by choosing materials with much different impedances. For example, when a wave traveling through air (which has relatively low impedance) encounters a wall (which has very high impedance), the wave will reflect back on itself and much less of the wave will continue on in its previous path. Acoustical impedance is not merely a property of materials. Impedance also changes with changes in cross-sectional area, bends, junctions, and openings.

The acoustic impedance is defined as the ratio of the complex acoustic pressure to the complex particle velocity.

\[
Z = \frac{P}{V} \quad \text{(rayl)}
\]  

(2.4)
When the impedance relates to the interface between different media, the appropriate velocity is the component directed normal to the interface. Then the associated impedance is the boundary impedance, $z_n$ (Fahy 2001). Since the impedance is a complex quantity, it can also be represented in terms of its real and imaginary parts, the resistance and reactance, respectively.

$$z_n = r_n + jx_n$$  \hspace{1cm} (2.5)

It is often convenient to normalize the acoustic impedance by dividing this quantity by the (real) characteristic acoustic impedance of the medium, $\rho_o c$, where $\rho_o$ represents the fluid density and $c$ the speed of sound in the medium through which the incident wave travels.

$$z'_n = \frac{z_n}{\rho_o c} = \frac{r_n}{\rho_o c} + j \frac{x_n}{\rho_o c}$$  \hspace{1cm} (2.6)
2.1.3 Absorption Coefficient

A convenient way to express impedance discontinuities is by the sound power absorption coefficient. The absorption coefficient is the ratio of the sound energy absorbed by a material to the sound energy incident on that material. This quantity varies with the angle at which the wave is incident, \( \phi \), but often only the absorption coefficient for normal incidence (\( \phi = 0 \)), \( \alpha_o \), is reported.

\[
\alpha(\phi) = \frac{4r'_n \cos \phi}{(1 + r'_n \cos \phi)^2 + (x'_n \cos \phi)^2} \quad (2.7)
\]

\[
\alpha_o = \frac{4r'_n}{(1 + r'_n)^2 + (x'_n)^2} \quad (2.8)
\]

**Figure 2.1** Angle of Incidence, \( \phi \)
2.1.4 Decibel Scales

In the field of acoustics, measurements are often reported in terms of decibels. The decibel scale is a logarithmic scale that is defined in terms of a reference point. The decibel system was developed because our sense of hearing responds to sound pressures more or less in a logarithmic, rather than a linear way. The decibel scale also helps us to understand acoustic quantities over a very large range. We refer to decibel quantities as sound levels. For example, the sound intensity level is defined as

\[ L_I = 10 \log_{10} \frac{I}{I_{\text{ref}}}, \]

(2.9)

where the reference intensity, \( I_{\text{ref}} = 10^{-12} \text{ W/m}^2 \).

The sound power level, then, is defined as

\[ L_W = 10 \log_{10} \frac{W}{W_{\text{ref}}}, \]

(2.10)

where the reference power, \( W_{\text{ref}} = 10^{-12} W \).

Since the intensity and power quantities are related to the square of the pressure, the sound pressure level is defined as

\[ L_P = 10 \log_{10} \frac{p_{\text{rms}}^2}{P_{\text{ref}}} = 20 \log_{10} \frac{p_{\text{rms}}}{P_{\text{ref}}}, \]

(2.11)

where the reference pressure, \( P_{\text{ref}} = 20 \mu Pa \) for air.

Once the data has been converted to the decibel scale, it is often given a weighting. The weighting helps to further match the measured quantities to the response
of the human ear. The following chart depicts the A-weighted scale, and some values are given in the table below. The weighted values are then referred to in units of dBA.

![A-Weighted Value](image)

**Figure 2.2** A-weighting values for Decibel Scale

**Table 2.1** A-weighting values for Decibel Scale

<table>
<thead>
<tr>
<th>Frequency (Hz)</th>
<th>A-Weighting (dB)</th>
</tr>
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<tr>
<td>31.5</td>
<td>-39.4</td>
</tr>
<tr>
<td>63</td>
<td>-26.2</td>
</tr>
<tr>
<td>125</td>
<td>-16.1</td>
</tr>
<tr>
<td>250</td>
<td>-8.6</td>
</tr>
<tr>
<td>500</td>
<td>-3.2</td>
</tr>
<tr>
<td>1000</td>
<td>0.0</td>
</tr>
<tr>
<td>2000</td>
<td>1.2</td>
</tr>
</tbody>
</table>
When an overall dBA level for a frequency spectrum is calculated, the results are converted into dBA levels for each frequency as described above. These values must then be converted into a sound power amplitude (in Watts) using the equation below. The sound power amplitude is then summed across all frequencies. This total sound power is then converted back into a sound power level using equation 2.10.

\[
W = W_{ref} \times 10^{\left(\frac{L_W}{W_{ref}}\right)}
\]  

(2.12)

### 2.2 Sound Transmission through Walls and Panels

When a propagating wave encounters an infinite barrier the wave is dispersed into two new waves (Blanks 1997). Some of the wave is reflected back towards the source and some of the wave is transmitted through the panel. In this case, the wave is not diffracted around the barrier because the barrier is considered infinite.

![Diagram of pressure waves](image)

**Figure 2.3** Pressure Waves Normally Incident on an Infinite Barrier
A pressure balance for both sides of the panel shown above requires that

\[ P_i + P_{\text{ref}} + P_{\text{rad}} = P_t \]  \hspace{1cm} (2.12)

Then, if the pressure amplitude of the each wave is represented by \( P \), a transmission coefficient can be defined as

\[ \tau = \frac{|P_i|^2}{|P_t|^2} \]  \hspace{1cm} (2.13)

The transmission loss through the wall is then

\[ TL = 20 \log_{10} \left( \frac{1}{\tau} \right) \]  \hspace{1cm} (2.14)

Defining a transmission coefficient for a panel is convenient because it provides an easy way to calculate the transmission loss for a composite panel. If a panel consists of \( n \) sub panels of differing area, \( S_n \), each with a different transmission loss, the overall transmission loss of the composite panel can be found by calculating the area-weighted transmission coefficient of the composite panel.
Figure 2.4 Composite Panel

\[ TL = 10 \log_{10} \left( \frac{1}{\tau} \right) \]  
(2.15)

\[ \tau = \frac{1}{S} \sum_{n=1}^{N} \tau_n S_n \]  
(2.16)

\[ S = \sum_{n=1}^{N} S_n \]  
(2.17)
2.3 Insertion Loss

The sound power insertion loss, hereafter referred to simply as the insertion loss is the most commonly used measure of the effectiveness of an enclosure. It is defined as the difference between the sound power level radiated by the unenclosed source and the sound power level radiated by the enclosed source. Thus it is the measure of the reduction in sound power due to the enclosure. The definition of insertion loss is also depicted in Figure 2.4 below.

\[
IL_w = 10 \log \left( \frac{W_o}{W_E} \right) = L_{Wo} - L_{WE} \quad \text{(dB)}
\]  (2.18)

Insertion loss is a good way to measure the acoustical performance of an enclosure because the measure is independent of the input sound power. As such, many different cases can be compared even when the input sound power is different. Unlike the transmission loss, insertion loss is installation sensitive, that is, it accounts for any effects produced by adding the enclosure, such as alteration of the source sound power, or changes in the flow (Fahy 2001). It is therefore the most realistic measure of enclosure performance. Throughout this study, insertion loss will be used as the measure of acoustical performance of enclosures.

Figure 2.5 Definition of Insertion Loss (Beranek 1992)
2.4 Point Sources

Point sources are used in this study as a simple, theoretical source in order to gain a more complete understanding of enclosure behavior. The following section describes the theory of the point source.

The sound pressure radiated by a spherical source of amplitude $A$ is

$$p(r) = A \frac{e^{-ikr}}{r}$$

(2.19)

where $k$ is the wave number, $\omega/c$, and $r$ is the distance between the source and the field point. This equation reveals a singularity at the location of the source where the pressure is infinite. The following equations provide a way around this difficulty by using the volume velocity, $Q$.

The volume velocity is found by integrating the radial velocity along a control spherical surface of radius $r_o$ around the point source.

$$Q = 4\pi r_o^2 \cdot v(r_o) = \frac{4\pi A}{i\rho ck} (1 + ikr_o) e^{-ikr_o}$$

(2.20)

where $v(r_o)$ is the radial velocity at a distance $r_o$ from the source, $\rho$ is the density of the acoustic medium, and $c$ is the speed of sound in the acoustic medium. If the radius of the control sphere is small compared with the acoustic wavelength, ($kr_o<<1$), equation 2.20 simplifies to:

$$Q = \frac{4\pi A}{i\rho ck}$$

(2.21)

The acoustic intensity is then given by

$$I(r_o) = \frac{p_{eff}^2(r_o)}{\rho c} = \frac{p^2(r_o)}{2\rho c} = \frac{\rho ck^2 Q^2}{32\pi^2 r_o^2}$$

(2.22)
Finally, the acoustic power is then obtained by integrating the intensity along the spherical surface.

\[ W = 4\pi r_0^2 \cdot I(r_0) = \frac{\rho c k^2 Q^2}{8\pi} \]  

(2.23)

The final result is an equation for the power of a point source which does not depend on the distance \( r_0 \) (Numerical Integration Technologies 1999).

2.5 **Overview of the Boundary Element Method**

The boundary element method, or BEM, is a numerical solution for engineering problems. Mathematical equations are very difficult to develop for complicated geometries. In order to make the process more manageable, the boundary element method divides a surface into discrete elements so that the mathematical equations can be applied to each element individually. Using this method, equations are formulated for each element and combined to obtain a solution for the entire body. Unlike the finite element (FE) method, the BEM requires that only the surface of a body be modeled, rather than the entire object. This gives several advantages over the FE model. First, the model is easier to create. Second, there are far fewer elements, sometimes resulting in a faster solution time. Third, unbounded domains are particularly suited to the BEM. This is especially important for acoustic problems because acoustic domains are often unbounded.

**Direct BEM**

There are two basic categories of the BEM, the direct and indirect formulations. The direct boundary element method (DBEM) is based on the Helmholtz integral equation (Seybert and Wu 1997). The primary variables defined for the DBEM are the acoustic pressure and the particle velocity on the boundary, as shown in Figure 2.5. For the DBEM, the defined boundary must be a closed surface and the domain must be defined as interior or exterior to the boundary, but results are relatively easy to interpret.
The Helmholtz equation used for the direct boundary element method is as follows:

\[
C(P)p(P) = - \int_S \left[ p(Q)G'(Q, P) + jkz_0 v_n(Q)G(Q, P) \right] dS(Q) \tag{2.24}
\]

- \( p(P) \equiv \) sound pressure at any point in the domain
- \( S \equiv \) boundary of domain
- \( p(Q) \equiv \) sound pressure at point \( Q \) on the boundary
- \( v_n(Q) \equiv \) normal velocity at point \( Q \) on the boundary
- \( z_0 \equiv \) characteristic impedance of the medium
- \( C(P) \equiv \) constant that depends on the location of \( P \)
  - \( = 4\pi \) for \( P \) inside domain
  - \( = 0 \) for \( P \) outside domain
  - \( = 4\pi - \int \frac{\partial}{\partial n} \left( \frac{1}{r} \right) dS(Q) \) for \( P \) on the surface

\[
G(Q, P) = \frac{e^{-jkr}}{r} \equiv \text{free space Green function}
\]

\[
r = |Q - P|
\]

\( G' \equiv \) derivative of \( G \) in the normal direction
**Indirect BEM**

The IBEM, on the other hand, is more general. The IBEM considers both sides of the boundary, the interior and exterior simultaneously. This method is derived from potential theory and, rather than using the sound pressure and normal velocity on one side of the boundary, the primary variables are the single- and double-layer potentials. The single layer potential, $\delta dp$, is the difference in normal gradient of the pressure and is related to the normal velocities, $v_{n1}$ and $v_{n2}$ (Hamdi 1982).

![Schematic Showing Variables Defined for an Indirect BEM](image)

**Figure 2.7** Schematic Showing Variables Defined for an Indirect BEM

\[
\delta dp = \frac{\partial p_1}{\partial n_1} - \frac{\partial p_2}{\partial n_2} \tag{2.25}
\]

The double layer potential, $\delta p$, is the difference in acoustic pressure across the boundary, $p_1$ and $p_2$.

\[
\delta p = p_1 - p_2 \tag{2.26}
\]
The equation used for the IBEM is as follows:

\[ p(P) = \int_S \left[ G(r) \delta dp - \frac{\partial G(r)}{\partial n} \delta p \right] dS \]  \hspace{1cm} (2.27)

The IBEM has the advantage of being able to handle open boundaries, such as partial enclosures. Also, the IBEM can model coupled interior-exterior problems and can be efficiently coupled with finite element models.
Chapter 3  
Literature Review

Enclosures are a common and practical way to reduce machinery noise in many cases. The use of enclosures is widespread and diverse, as are attempts to understand and study them. However, enclosures are also extremely complex acoustic devices and their performance is difficult to predict. The acoustic space within an enclosure, although deceivingly similar to an enclosed room, requires an entirely different approach than the well-developed techniques of room acoustics. To date, theoretical predictions of sound fields inside an enclosure have not been entirely successful.

3.1 Review of Theoretical Models for Enclosures

In one of the first attempts to analytically predict the performance of enclosures, Jackson proposed an empirical model in which the source and enclosure were modeled as infinite parallel plates. The first plate vibrates as a constant volume velocity source and the second plate acts as an enclosure wall, and from here, an insertion loss value was calculated (Jackson 1966). There are, of course, inherent drawbacks to this analysis. In practice, sources and enclosures are, of course, not infinite, and the sound power cannot be accurately determined from an r.m.s. velocity. However, this study was useful not only in paving the way for further studies, but it also predicted the negative insertion loss at low frequencies.

Junger later improved on Jackson’s theory by modeling the enclosure as a finite, rectangular panel, with a finite rectangular piston source (Junger 1970). Junger’s model also predicted a significant dip in the insertion loss at low frequencies. This investigation, although more realistic than that of Jackson, still assumes an unrealistic source vibration. Additionally, Tweed and Tree demonstrated that neither the models of Jackson or Junger compare well enough to experimental results to be used for design (Tweed and Tree 1978). Tweed and Tree also examined a study by Ver. Ver divided his models into a low, middle, and high frequency range. The low-frequency range was that below the fundamental panel or acoustic modes and Ver gives an insertion loss that is
independent of frequency in this range. The middle-frequency range was that where both panel and air cavity resonances become important. The high-frequency range is identified as that where the sound field inside the enclosure is reverberant, similar to a diffuse field. This method was lacking as well, because Ver does not present an analytical model for the middle-frequency range, which is of most interest to designers.

More recently, Byrne, Fischer, and Fuchs investigated a procedure for predicting the performance of sealed, close-fitting, machine-mounted enclosures (Byrne et al 1988). Some construction methods are presented which minimize the effect of the structural coupling on the insertion loss. Also, a method is presented for predicting the effect of the structural couplings. However, the subject of this thesis concentrates on enclosures which are not sealed, nor machine-mounted.

In 1991, Oldham and Hillarby published a detailed investigation of close-fitting enclosures (Oldham and Hillarby 1991a). Their analysis was divided into a low-frequency model and a high-frequency model. The low-frequency model, which will be more relevant to this paper, assumes the acoustic pressure incident on the enclosure panels will be uniform up to a cutoff frequency which is based on the dimensions of the enclosure panel. The high-frequency model utilizes the statistical energy analysis (SEA) technique. In an extension of their analysis, Oldham and Hillarby also published some experimental results based on their analysis methods (Oldham, Part 2 1991). They showed that the high frequency model yields good results for its effective range. The low-frequency model, on the other hand, was less successful. It showed good results for simple source configurations. When the source became more complex, i.e., vibrating at mode shapes other than the first mode, agreement between experimental results and theoretical predictions is not as good. Thus, a better method is necessary to account for complex sources, such as real engines and machinery.
3.2 **Boundary Element Method**

The increase in computer power in recent decades has led to the use of numerical solution methods for more and more problems in acoustics. Beginning as early as 1969, researchers began applying the finite element method to acoustic problems (Craggs 1969). More recently, the boundary element method was developed and has become one of the primary methods of acoustical analysis. The boundary element method, or BEM, is believed to be the most valuable approach for the study of close-fitting enclosures (Crocker 1994).

The BEM has been already been successfully applied to several acoustic problems involving enclosures. Bernhard et al. (1987) used both the direct and indirect BEM to model sound fields inside an aircraft cabin, showing good agreement to measured results. Suzuki et al (1988) found accurate results when the BEM was applied to vehicle cabins with complicated boundary conditions. Utsuno et al (1990) reported an investigation using bulk reacting sound absorbing materials, which also yielded good results. Herrin et al (2003) also successfully used the boundary element method to model the sound radiation from an engine cover as well as the interior of a lined enclosure. Quabili (1999) used the boundary element method to model the interior of a vehicle cabin with good results. Martinus (2000) then used the BEM to model the same vehicle cabin as a partial enclosure with louvers. These results also compared well with experimental results.

![Engine Cover tested by Herrin et al (2003)](image)

**Figure 3.1** Engine Cover tested by Herrin et al (2003)
Figure 3.2 Comparison of Sound Power Results from BEM and From Experiment for Engine Cover

Figure 3.3 Comparison of Sound Power BEM results and Measured results for Engine Cover inside Lined Enclosure
3.3 Current Prediction Technique

Despite the difficulty in predicting enclosure performance, extensive efforts have been made to develop equations and analysis techniques to help designers better estimate the effectiveness of an enclosure. These equations, however, are still incomplete and designers are forced to rely mainly on experimental results. This section will discuss some common textbook equations relating to enclosures.

Estimating IL

In the textbook, *Noise and Vibration Control Engineering*, Beranek and Ver include a chapter on acoustic enclosures. In the small section describing partial enclosures, they include a formula for estimating the insertion loss of an enclosure which involves solid angles (Beranek 1992). Solid angles are a three-dimensional counterpart of the common angle with units of radians. The solid angle,Ω, subtended by area \( S \) at the center of a spherical surface of radius \( r \) is defined to be

\[
\Omega = \frac{S}{r^2} \quad (3.1)
\]

The solid angle \( \Omega \) is dimensionless and its unit is called the steradian. Recalling that the surface area of a sphere is \( 4\pi r^2 \), the solid angle of a full sphere is then

\[
\Omega = \frac{4\pi r^2}{r^2} = 4\pi \text{ steradians} \quad (3.2)
\]

(Serway 2000).
Figure 3.4 Illustrations of (a) an Angle and (b) a Solid Angle
(Hull 2005)

Figure 3.5 Measuring (a) an Angle and (b) a Solid Angle
(Hull 2005)

Figure 3.6 An Additional Illustration of Solid Angle
The equation for the insertion loss (in dB) of a partial enclosure is then

\[
IL = 10 \log \left[ 1 + \alpha_{ave} \left( \frac{\Omega_{tot}}{\Omega_{open}} - 1 \right) \right]
\]  

(3.3)

where \( \Omega_{tot} \) = the solid angle of sound radiation of the unenclosed source

\( \Omega_{open} \) = the solid angle the enclosed source “sees”

\( \alpha_{ave} \) = the average absorption coefficient over the surface of the enclosure

(Beranek 1992). The solid angle of radiation of the floating unenclosed source is the solid angle of a full sphere, or \( 4\pi \) steradians. The solid angle the enclosed source sees is given by

\[
\Omega_{open} = \frac{S_{open}}{r^2}
\]  

(3.4)

where \( S_{open} \) is the open area located a distance \( r \) away from the source (Serway 2000).

Also recall that the average absorption over the enclosure is found by using a weighted average.

\[
\alpha_{ave} = \frac{\sum_{i=1}^{n} S_i \alpha_i}{S_{tot}}
\]  

(3.5)

where \( \alpha_i \) is the absorption coefficient on the \( i \)th panel of area \( S_i \), where \( n \) panels make up the enclosure, which has a total surface area of \( S_{tot} \).

The following chart plots the results from the preceding equation along with the results found in the BEM analysis from this investigation. The results shown are for the default enclosure configuration, which has an opening size of 5.85%, or 0.23 m² (See Chapter 4). The absorption coefficient is frequency dependent and is shown in Figure 3.5.

26
Figure 3.7 Comparison of IL Predicted by Textbook Theory and IL found in SYSNOISE

Figure 3.8 Average Absorption Coefficients Over Enclosure
From the preceding chart, it is obvious that, although Beranek’s equation may be useful for obtaining a very rough estimate of the overall insertion loss of an enclosure, it is certainly lacking in a number of areas. First, the equation does not take into consideration the resonance frequencies of the enclosure, which can cause an amplification in sound levels. Second, the equation tends to underestimate the insertion loss (excepting the first mode) by quite a lot. However, the equation is useful for examining trends and effects of some factors. It is also useful for getting a conservative estimate for the overall insertion loss.
Chapter 4
Experiment Setup

In order to analyze all the factors involved in this study, a methodology was developed using the indirect boundary element method to analyze several different cases. The first step was to create a model mesh to be analyzed. Then the mesh was imported into SYSNOISE, boundary conditions were assigned, and then the software was used to analyze the problem. After the analysis was complete, a field point mesh was created and processed, and from there, the sound power radiated by the model was found. This process will be discussed further in the following sections.

This chapter will also discuss the method of analysis and why the classic method of varying one factor at a time was chosen over other experiment designs.

4.1 Analysis Procedure

Because the waves in the field are calculated exactly in the BEM, using a deterministic approach, all wave behavior is taken into account exactly from the boundary conditions given. Remember that the results of the study will not be exact because structureborne noise is not considered boundary conditions may not be exact. The only sources of error will be assumptions made about material properties, geometrical approximations, boundary conditions, and, of course, the discretization error, introduced by the division of the acoustic field into elements. The discretization error is characterized by a maximum frequency for which the results are “reasonably accurate.” For linear boundary elements, it is recommended that the element size be selected such that there are at least six elements per wavelength (Marburg 2002).

4.1.1 Creating the Mesh

All the model meshes in this study were created using I-DEAS. The first mesh that was created was that of the default enclosure. The geometry of the enclosure was chosen to be rectangular, so as to be easiest to model and to analyze. The dimensions
were then chosen to correspond to the dimensions of the engine, leaving but a few (5-10) centimeters between the engine and the enclosure. Ten separate panels were created on the surface of the enclosure – one each on the front and back sides, and two each on the top, bottom, left, and right sides. One of these areas was unmeshed and served as the opening. The mesh was created with thin shell elements with a maximum length of 50 mm. Thus, an analysis of the mesh would be good up to about 1140 Hz. The default enclosure mesh was made up of 1814 nodes and 1793 elements.

Figure 4.1 Enclosure Mesh
Once the enclosure mesh had been created, the engine mesh could be inserted into the enclosure mesh for those cases that used the engine. The engine was modeled from a Cummins B-series diesel engine. The engine mesh was made up of 1918 thin shell elements of 50mm in length and 1854 nodes. Another consideration was avoiding the non-existence problem which is encountered when using the indirect BEM (Coyette and Roissen 1990). When using the IBEM, there is no distinction between the interior and exterior analysis and the primary variables are obtained using information from both sides of the boundary. When resonant frequencies for the interior are encountered, the solution at the exterior points is contaminated by the large differences in pressure between the interior and exterior surfaces. This difficulty is counteracted by adding absorptive panels inside the boundary. Thus, two absorptive planes were added inside the engine to counteract the non-existence problem that is encountered when using the indirect BEM.

Figure 4.2 Engine Mesh
Figure 4.3 Engine Mesh Showing Absorptive Planes

Figure 4.4 Engine with Enclosure

4.1.2 Boundary Conditions
Once the model mesh was imported into SYSNOISE, the next step was to assign boundary conditions to the problem. For a well-posed problem, some information must be known for each node. There were three types of boundary conditions used on the enclosure: acoustic impedances, free edges, and rigid walls. Then, either a point source was assigned or velocity boundary conditions were assigned to the engine mesh.

**Acoustic Impedances**

This study required that absorbing material be placed inside the enclosure as indicated by each test case. A 1-inch thick glass fiber material was used. The impedance of the glass fiber was measured experimentally, using the two-microphone impedance tube method (ASTM 2005). The measured impedance results are shown below in Figure 4.4, with the associated absorption coefficient shown in Figure 4.5. This impedance was then added into the BEM as a boundary condition. Since SYSNOISE does not accept frequency-dependent impedance, the impedance values were converted to admittance values simply by taking the reciprocal. Also, the indirect BEM requires that the direction of the absorption be specified. In this case, the absorbing material is on the inside of the enclosure and the normals are pointed in this direction, so the admittance specified is in the positive direction.

![Impedance Boundary Condition](Figure 4.5)

**Figure 4.5** Impedance Boundary Condition
Absorption Coefficient

Figure 4.6 Absorption Coefficient of Absorbing Material

Free Edges

Since the indirect boundary element method solves simultaneously for the interior and exterior solutions, another type of boundary condition arises. Jump boundary conditions can be assigned which relate the points on the interior of the model to those on the exterior. At the free edges, i.e. the edges of the openings, a zero jump of pressure is specified to insure that there is no pressure difference between the inside and outside of the enclosure at these points. The edges of the absorbing panels interior to the engine were also considered free edges and were assigned a zero jump of pressure.

Rigid Walls

All nodes of the enclosure not assigned an impedance are assumed to be rigid and are left to the default condition in SYSNOISE. This condition is not exactly true in practice. The walls have some inherent absorption. However, this absorption is much lower than the sections where absorbing material is used, so this assumption is valid.
**Source Boundary Conditions**

For the cases using a point source, the source was added using the point source function in SYSNOISE. The source was placed in the center of the enclosure and was chosen to be spherical with a sound power output of 1 W.

For the cases that were analyzed using a real engine as the source, measured velocities from the engine were used as boundary conditions on the engine. (For details on vibration measurements, see reference by Charan 2000).

**4.1.3 SYSNOISE Analysis**

After the model mesh was imported into SYSNOISE and the boundary conditions assigned, SYSNOISE is ready to solve the model. For all the point source cases in this study, the BEM analysis was conducted at every frequency between 0 and 200 Hz, and from 200-1000 Hz in 10 Hz steps. This frequency resolution will allow a very thorough study of the enclosure behavior, particularly at the lower frequencies. The engine cases were limited to frequencies where the engine velocity was known. Thus, the engine cases were solved in 20 Hz steps from 0-1000 Hz. The narrowband results are reported for some representative cases, but most were converted to one-third octave band results.

After the model was solved, a spherical field point mesh was created around the enclosure and analyzed in the same frequency range. The total sound power through the field point mesh can then be found. If the same analysis is performed for the source without the enclosure, this total sound power can also be found. The difference between these two sound power levels will then give the insertion loss for each case.
Figure 4.7 Field Point Mesh Surrounding Enclosure

Figure 4.8 Field Point Mesh Surrounding Engine Alone

(Insertion Loss is calculated by subtracting results from Figure 4.7 from results from Figure 4.8)
4.2 Design of Experiment

The main objective of this study is, of course, to determine how the eight different factors – opening size, opening location, absorption coverage, absorption location, absorption coefficient, enclosure size, source location, and velocity boundary condition - affect the performance of an enclosure. It becomes imperative, then, that an experiment be designed that will test all of the important factors and provide meaningful results that can be interpreted and understood by designers. The design methodologies considered included a factorial method, a Taguchi method, and a single-factor method. The factorial method, in which every possible combination of factors is tested, was rejected because of the sheer number of test cases involved and the enormous amount of resources this would consume. The Taguchi method, where orthogonal factor combinations can be condensed from the factorial method, was also rejected because we cannot assume the factors to be orthogonal. Therefore, the single-factor experiment design was chosen. The experiment was essentially divided into eight different experiments, one for each factor. However, a default enclosure was designed and used as a baseline, allowing consistency and comparability between the experiments. This allowed for an in-depth study of each variable with a reasonable amount of resources. Since all analysis was done within a computer, there were no nuisance factors to consider and randomization was not a concern.

4.2.1 Default Enclosure

The default enclosure used in this study was chosen so as to represent an enclosure that might be used in industry. The enclosure dimensions were chosen to be only slightly larger (5-10 cm) than those of the engine. The enclosure has a single opening in the left rear that accounts for 5.85% of the total surface area. Absorption was added on all other nine panels of the enclosure, representing about 53% of the total surface area. The source was placed in the center of the enclosure and was given boundary conditions as discussed previously.
In some cases, the variation of the factors made it necessary to deviate slightly from the default. These variations were chosen so as to have the least effect on the output and are outlined below.

![Figure 4.9 Default Enclosure Configuration](image)

- **Opening Size**: In analyzing the opening size, open areas of up to 25% were examined. Larger opening sizes required that more than one surface panel be open. Therefore, cases with opening size 6% and smaller were designed so that the opening was in the left rear. For case with opening sizes of 8%, portions of both the left rear panel and right rear panel were open. For cases with 15%-25% open area, portions of all four side panels were left open. Because of this variation, the absorption coverage for all opening size cases was also varied so that it was consistent for all opening sizes. Only the front, back, top, and bottom panels were given absorption, so that the absorption coverage was only 35% of the total surface area.
- **Opening Location**: When the location of the opening was varied, the absorption location was varied accordingly, so that the absorption was found on all nine panels, except the one which was open. In this way, the absorption coverage remained consistent with the default throughout the analysis.

- **Absorption Location**: The location of the opening remained in the left rear, as consistent with the default, in each case except when the absorption was located in the left rear. In this case, the opening was placed on the right rear panel and had the same open area. Also, only one absorbing panel was assigned for each case so the absorption coverage in each case was only 5.85%.

- **Absorption Coverage**: For these cases, absorption was applied to multiple panels to total the amount of coverage under consideration. It was applied first to the side panels, then, for cases with higher coverage, the front, back, top, and bottom panels were added, in that order.

For all other cases, only the factor in question was varied from the default. When looking at the enclosure size the relative position and proportion of the absorption and opening was consistent with the default. The absolute source size and central position also remained the same regardless of the enclosure size. Also, when the location of the source changed inside the enclosure, all other aspects of the enclosure remained consistent with the default.

### 4.2.2 Sources

This experiment utilizes both a point source and a real engine inside the enclosure. This is useful because the point source will allow for more predictable and understandable results. Since the size of the point source is negligible, this is another factor that does not need to be considered in the analysis. Also, its excitation is uniform at all frequencies and in all directions. Again, this should provide more predictable and understandable results. These results may help us to better understand the cases with real sources.
Once the numerical analysis has been performed with the point source, it is also important to study the behavior of the enclosure with an actual source. A diesel engine was used as an example. It is important to study this case as well because the results will be more useful to industry. The diesel engine is a good example for study because it has a large complicated geometry, which could in fact influence the results. It also has complicated boundary conditions that vary by frequency and position, just as a real source is likely to do. All of these test cases should give a good overall representation of enclosure behavior.
Chapter 5
Results

This section discusses the results for each model as taken from SYSNOISE. Each factor is discussed separately and includes both the cases using the point source and the engine. The effects of each factor will be analyzed in detail.

5.1 Default Cases

The following chart shows the results from both the point source case and the engine case for the default enclosure. Recall that the default enclosure has an opening in the left rear, which is 5.85% of the enclosure surface area and that absorption was placed on all other nine panels for a total of 53% coverage. Recall also that the frequency resolution for the point source case was 1 Hz up to 200 Hz and 10 Hz beyond 200 Hz. For the engine case, the frequency resolution was only 20 Hz.

![Comparison of Source Type](image)

Figure 5.1 Comparison of Point Source and Engine with Default Enclosure
Notice that the insertion loss curve for the point source is relatively smooth, while the curve for the engine case has a lot of peaks. This displays the tonal quality of the noise coming from the engine. The point source case, on the other hand, emits a constant input power across the entire frequency range. One can also see that the insertion loss is slightly higher for the point source case. This is true for almost all of the cases tested. Since the input power is constant and evenly distributed in space, it is more likely to strongly excite all the modes of the enclosure. On the other hand, the engine excitation is spatially distributed and some parts of its excitation are likely to be out of phase with the enclosure modes.

5.2 Enclosure Size

The first factor that was examined in this study was the size of the enclosure. The effects of the enclosure size are very useful to know when designing an enclosure. The results will determine whether it is necessary to achieve a certain size for the enclosure. In order to study this factor, the default design was used. For each case, the default enclosure was scaled according to a ratio of volume to the default, close-fitting case. The openings and the absorption were also scaled accordingly so that their relative positions and sizes remained consistent with the default case. The size of the source remained the same in each case.

Figure 5.2 Enclosure with Engine, Volume Ratio = 4
5.2.1 Enclosure with Point Source

The first step in analyzing the effects of the enclosure will be to look at the behavior of the enclosure when a point source is used. The point source is uncomplicated and easily interpreted. Then, once a thorough understanding of enclosure behavior with a point source is achieved, we can apply this knowledge to further study of enclosures with real sources.

The results are shown below in Figure 5.3 for each case. Interestingly, above about 500 Hz, there seems to be little difference in the results. However, below 500 Hz are some interesting trends. First, there is a large dip in the insertion loss at very low frequencies. In fact, this dip becomes significantly negative, signifying that the sound power is actually increased due to the addition of the enclosure at these frequencies. This dip is due to acoustic resonance in the enclosure and is expected (Bai 1992). It is important to realize that an enclosure may not always reduce sound levels.

Although it is important to understand the presence of the negative insertion loss region, its effects may not be critical. The negative insertion loss values all occur below 100 Hz, where the human ear is not especially sensitive (refer to A-weighting curve in Chapter 2, Figure 2.2). The vibrations produced and transmitted by these enclosures would be of greater concern. Care should be taken if an enclosure such as those discussed here were to be placed adjoining an object with a natural frequency in the range of the negative insertion loss, for example, a building. A designer will want to pay close attention to any enclosure located on the roof of or in the basement of a building.

An important effect of the enclosure size is that this dip in the insertion loss shifts to the left as the volume of the enclosure increases. This is because, at low frequencies, the enclosure is dominated by modes in the acoustic space. The first acoustic mode inside the enclosure is a quarter-wavelength mode and its frequency is determined primarily by the dimensions of the enclosure. The differing insertion loss in the region between 100 Hz and 500 Hz is likely also due to acoustic modes. These values also tend to shift to the left as enclosure size increases. To reinforce this concept, Figure 5.4 plots the same insertion loss values versus the dimensionless quantity $\frac{ka}{c}$, where $k$ is the wave number, $\omega$, and $a$ is the longest dimension across the enclosure between diagonal corners which
is different for each case. From this chart it is obvious that once the enclosure dimension is factored out, the insertion loss is nearly identical for each case.

![Comparison of Enclosure Size](image)

**Figure 5.3** Comparison of Enclosure Size with Point Source

![Comparison of Insertion Loss for Different Enclosure Sizes vs. ka](image)

**Figure 5.4** Comparison of Insertion Loss for Different Enclosure Sizes vs. $ka$
The following figures are contour plots of the enclosures at their nadir frequency. They further reinforce the theory of the dominant first mode.

Figure 5.5 Contour Plot Showing Surface Pressure of Enclosure with Volume Ratio = 2 at 42 Hz

Figure 5.6 Contour Plot Showing Surface Pressure of Enclosure with Volume Ratio = 3 at 37 Hz
Figure 5.7 Contour Plot Showing Surface Pressure of Enclosure with Volume Ratio = 4 at 33 Hz

Figure 5.8 Contour Plot Showing Surface Pressure of Enclosure with Volume Ratio = 10 at 24 Hz
5.2.2 Enclosure with Engine

The results for the enclosure with the real engine case are shown in Figure 5.9. In general, insertion loss values are similar to the point source cases. It is apparent that there is still a dip in the insertion loss at low frequencies. The insertion loss settles somewhat at higher frequencies to near 10dB, just as in the point source case. It is more difficult to see, partly because the original frequency resolution was not as good and does not allow for careful analysis at low frequencies, but it also seems that other modes between 100 Hz and 500 Hz are shifted to the left with increasing enclosure size. Thus, the same trends are all confirmed for a practical case. Figure 5.10 also displays the same results with $ka$ on the horizontal axis.

![Comparison of Enclosure Size](image)

**Figure 5.9** Comparison of Enclosure Sizes with Engine, IL vs. Frequency
The insertion loss results for many of the engine cases show a second dip in the insertion loss into the negative region near 100 Hz. This is evidence of a second acoustic mode which comes into play for the larger enclosures. This second mode could, in fact, be of greater importance to the noise control engineer than the first mode. The human ear will have greater sensitivity to these modes which are at higher frequencies.

**Figure 5.10** Comparison of Enclosure Size with Engine, IL vs. ka
5.3 Opening Size

To study the effects of the opening size, the area of the opening was varied. The opening size was measured as a percentage of the total surface area of the enclosure. Opening sizes of 1%, 2%, 3%, 4%, 6%, 8%, 15%, 20% and 25% were studied. A typical engine enclosure will allow for an opening size of 6%-15% for ventilation and cooling. Other sizes were also added for further study and as examples of extreme cases. The size and shape of the enclosure remained exactly the same as the default model. As discussed previously, for consistency, only the top, bottom, front, and rear panels used absorption. The positioning of the opening remained in the left rear for the 1%, 2%, 3%, 4%, and 6% opening size cases. For the case with 8% opening size, the openings were located on both the left and right rear. Then, for the 15%, 20%, and 25% opening size cases, four separate openings were used on the left rear, right rear, left front, and right front of the engine. The absorption coverage was consistent for each case at 35%.

<table>
<thead>
<tr>
<th>Opening Size</th>
<th>Location of Opening</th>
</tr>
</thead>
<tbody>
<tr>
<td>1%</td>
<td>Left Rear</td>
</tr>
<tr>
<td>2%</td>
<td>Left Rear</td>
</tr>
<tr>
<td>3%</td>
<td>Left Rear</td>
</tr>
<tr>
<td>4%</td>
<td>Left Rear</td>
</tr>
<tr>
<td>6%</td>
<td>Left Rear, Right Rear</td>
</tr>
<tr>
<td>8%</td>
<td>Left Rear, Right Rear</td>
</tr>
<tr>
<td>15%</td>
<td>Left Rear, Right Rear, Left Front, Right Front</td>
</tr>
<tr>
<td>20%</td>
<td>Left Rear, Right Rear, Left Front, Right Front</td>
</tr>
<tr>
<td>25%</td>
<td>Left Rear, Right Rear, Left Front, Right Front</td>
</tr>
</tbody>
</table>
5.3.1 Enclosure with Point Source

The results are shown below in Figure 5.12 for all cases. This analysis proves that, as expected, the size of the opening in the enclosure significantly affects insertion loss performance. Even a small difference in opening size can affect the overall performance by several dB. It is also evident that the opening size affects the insertion loss performance at nearly every frequency. This confirms a phenomenon that is intuitive and expected.
At frequencies below 100Hz, the size of the opening in the enclosure has some very important effects. Most notably, the opening size affects the frequency of the nadir. An increase in opening size will shift the nadir further to the right. A change in the size of the opening will change the impedance of the model at the opening location, and will thus change the frequency of the first mode. It is also important to observe that the dip in the insertion loss widens considerably as the opening size is increased. This is again attributed to the change in the impedance of the enclosure. The large opening sizes effectively act as a damper where acoustic modes are present.

The following figures plot the surface pressure of the enclosure at the nadir frequency for each case. The presence of an acoustic mode is evident in each case, despite the number of openings.
Figure 5.13 Contour Plot Showing Surface Pressure on Enclosure with 4% opening at 50Hz

Figure 5.14 Contour Plot Showing Surface Pressure on Enclosure with 8% opening at 75Hz

Figure 5.15 Contour Plot Showing Surface Pressure on Enclosure with 15% opening at 101 Hz
5.3.2 Enclosure with Engine

Now that the results have been studied for the theoretical case of the point source, the next step is to investigate the behavior of the enclosure with a real source, a diesel engine.

The results are shown in Figure 5.16. As the previous analysis of the enclosure with the point source found, here it is evident that the size of the opening greatly affects the overall insertion loss of the enclosure. The damping effects of the opening are also present in the engine cases.

![Comparison of Opening Size](image)

**Figure 5.16** Comparison of Opening Sizes on Enclosure with Engine vs. Frequency
5.4 Opening Location

Another factor that is sometimes considered important in enclosure design is the location of the openings. Often, a minimum opening size is required for ventilation, but this opening can be designed in different locations. The following section will analyze the effects of the opening location on enclosure behavior. The cases in this section all use the default model. The absorption is added on all nine panels which are not open.

5.4.1 Enclosure with Point Source

The results for the Opening Location comparison with a point source are found in Figure 5.17. Because of the symmetry of the source and the enclosure, there are only three independent cases to examine. The results can be divided simply into the cases with the opening on the sides, the top and bottom, and the front and back. There are small differences in the overall insertion loss, caused by the distance from the source to the opening and the greater excitation of some modes. The results below 100 Hz are consistent for each of the three cases because this region is dominated by the first acoustic mode and is dependent only on the enclosure dimensions, which remain unchanged. However, above 100 Hz, different modes may be excited based on the location of the opening.

Figure 5.17 Comparison of Opening Locations in Enclosure with Point Source
5.4.2 Enclosure with Engine

As shown in Figure 5.19, once the engine is placed inside the enclosure, the results are much more varied. Neither the engine geometry nor its boundary conditions are symmetric, so each case produces a different result. The source to opening distance is a factor, but for the engine, one must keep in mind that the excitation is not spatially uniform. There is a greater amount of noise radiating from the front of the engine. Therefore, the enclosures which have an opening near the front show a lower insertion loss and are less effective.

As with the point source case, the results are similar in each case in the low frequency range (below 100 Hz), since this region is dominated by an enclosure mode. The results are not identical because of the spatial nonuniformities presented by the engine. Above 100 Hz, there are different modes excited by changing the opening position.

**Sound Contribution from Engine Components**

![Graph showing sound power contribution from different engine components](image)

**Figure 5.18** Sound Power Radiated from Various Engine Parts
Comparison of Opening Location

Figure 5.19 Comparison of Opening Location on Enclosure with Engine
5.5 Absorption Coverage

Another important factor in the design of enclosures is the amount of surface area that is covered with absorbing material. In this study, the same absorption was used throughout, but the cases vary from 6% of covered surface area to 50%. All other factors remain unaffected and are equivalent with the default case.

5.5.1 Enclosure with Point Source

The results for the cases with varying coverage by absorbing materials are shown in Figure 5.20 below. One would expect that by increasing the amount of absorbing material, more sound would be absorbed, and thus, the insertion loss would increase. Indeed the results of this study confirm this trend. At the higher frequencies, the absorption has a much greater effect. It is worth noticing, however, that the addition of absorbing material tends to shift the nadir frequency to the left, in addition to shortening and broadening the dip. This is due to the change in impedance created by the absorbing material, effectively acting to damp the peak.

![Absorption Coverage Diagram]

Figure 5.20 Comparison of Absorption Coverage on Enclosure with Point Source
5.5.2 Enclosure with Engine

The following figure demonstrates that similar trends are found when the engine is added inside the enclosure. Again, the insertion loss is increased significantly with the addition of absorbing material. The most important effects on the insertion loss again occur at higher frequencies (above 500 Hz), where absorbing material has a very large effect. The damping of the peaks and the shift in the nadir frequency is seen again in these results, although it is more difficult to identify because of the frequency resolution. The results in the 30-40 Hz region appear to show that the larger absorption coverage will lead to a lower insertion loss in this region, but this apparent phenomenon is actually due to the poor frequency resolution in the area of the nadir frequency.

![Comparison of Absorption Coverage](image)

**Figure 5.21** Comparison of Absorption Coverage for Enclosure with Engine
5.6 Absorption Location

This study would not be complete without also considering the location of the absorbing material. In order to determine this effect, the absorbing material was added to each one of the ten different panels individually. Since only one panel was given absorption at a time, the overall absorption coverage for each of the cases in this section was only 5.85%. The opening remained in the left rear for each case, except when the absorption was placed in this location. For that case, the opening was moved to the right rear. All other factors remained consistent with the default case.

5.6.1 Enclosure with Point Source

The results comparing the location of the absorbing material inside the enclosure are shown in Figure 5.22. As with the opening location, the symmetry of this problem means that there are only three distinct cases to consider, those with the absorption on the sides, the top or bottom, or on the front or back. The results show that, for a point source, this is one factor that has no effect at all on the insertion loss. When the source is symmetric and uniform, the insertion loss will be identical no matter where the absorption is placed and designers can ignore this factor in planning a design.

![Absorption Location - Point Source](image)

**Figure 5.22** Comparison of Absorption Location on Enclosure with Point Source
5.6.2 Enclosure with Engine

The results shown in Figure 5.23 following illustrate that changing the location of the absorbing material within the engine can certainly have an effect. However, the overall effect on insertion loss is small. All cases seem to follow the same trend, having peaks in roughly the same places. The peaks and valleys in these cases are much less than those of the point source cases and the overall insertion loss values are higher. It appears that the small amount of absorption has a greater effect when the engine is present. This is probably due to the reduced space inside the enclosure for the acoustic waves to move about. The wave is more likely to be reflected multiple times before leaving the enclosure, thus providing more opportunities for the wave to be dissipated by the absorbing material. Thus, it could prove to be advantageous to have a source that is large relative to the enclosure.

![Absorption Location - Engine](image)

**Figure 5.23** Comparison of Absorption Location in Enclosure with Engine
The following figure demonstrates the much greater effect of damping from the 6% absorption coverage when the engine is present.

**Comparison of 6% Absorption Cases**

- **Engine - 6% Abs 4.8 dBA**
- **Point Source - 6% Abs 0.1 dBA**

**Figure 5.24** Comparison of 6% Absorption Coverage for Point Source and Engine
5.7 Absorption Value

In addition to the amount and location of the absorption, the absorption coefficient or impedance of the absorption used may also be important. Here, cases will be considered with several different absorption coefficients. In each case, the absorption will be placed on nine panels, just as in the default case, for an absorption coverage of 53%. In addition to the default case, a case was created with a constant absorption taken from the value at 100Hz. Also, a case was run using an absorption coefficient of one, representing full absorption.

5.7.1 Enclosure with Point Source

The results are shown in Figure 5.25 below comparing the cases with different absorption values. Of course, a greater absorption coefficient leads to a larger insertion loss and a more effective enclosure. With full absorption, the low frequency dip is

![Absorption Coefficient - Point Source](image)

**Figure 5.25** Comparison of Absorption Values for Enclosure with Point Source
completely eliminated. Also, at higher frequencies, the greater absorption serves to smooth out the insertion loss curve. It effectively lessens the contribution of the modes. In real cases, we can achieve high absorption coefficients at high frequency, and indeed, the default case closely matches the full absorption case at higher frequencies. However, it is difficult to achieve high absorption at low frequencies.

5.7.2 Enclosure with Engine

Just as in the cases with the point source, the results for the cases with the engine show the significant effect of the absorption. The insertion loss does not level out as much at higher frequencies when the engine is used, because it is not spatially uniform. The nonuniformity will cause it to excite some modes, but not all modes will be excited. At low frequencies, the effect of the absorption coefficient is dramatic. With full absorption, the large dip in insertion loss disappears just as in the cases with the point source. In this case, though, even the absorption levels achieved at 100Hz seem to be enough to drastically reduce the large drop in insertion loss. Recall that when the large engine is present inside the enclosure, the effect of the absorption is much greater.

**Absorption Coefficient - Engine**

![Graph showing absorption coefficient for enclosure with engine](image)

**Figure 5.26** Comparison of Absorption Coefficient on Enclosure with Engine
5.8 Source Location

The location of the source relative to the enclosure was also examined in this study. The default condition for the enclosure was used in all cases, but the source inside was moved relative to the enclosure. For the forward, aft, up, and down cases the source was moved 0.2m in that direction. In the left and right cases, the source was moved only 0.15m in that direction, as there was less room. The opening remained in the left rear, with absorption on all other sides.

Figure 5.27 Enclosure with Engine Source in Aft Region

Figure 5.28 Enclosure with Engine Source in Upper Region
5.8.1 Enclosure with Point Source

Figure 5.29 following shows the results for all the cases with the point source moved relative to the enclosure. The source location does indeed affect the overall insertion loss. When the source is moved to a region inside the enclosure, it is more likely to excite modes in that region. Therefore, the results show very high insertion loss peaks for some cases and lesser peaks in other cases. However, the results below 100 Hz remain unaffected in each case because the enclosure dimensions, thus, the first enclosure mode, are unchanged. Also, the position of the source relative to the opening will make a large difference in the insertion loss. With the opening in the left rear, then, it is observed that the forward case shows the best insertion loss, since the source is furthest from the opening. The case with the source moved to the right also shows a high insertion loss. When the source is moved to the left and the rear then, the insertion loss is much lower.

![Comparison of Source Location](image)

**Figure 5.29** Comparison of Point Source Location Relative to Enclosure
5.8.2 Enclosure with Engine

As discovered in the point source cases, the results for the engine within the enclosure also indicate that the insertion loss is greater when the source is moved away to the opening and it is lessened when moved closer. Again, the case where the engine was moved forward, furthest from the opening, shows the greatest overall insertion loss.

**Figure 5.30** Comparison of Engine Location Relative to Enclosure
5.9 **Velocity Boundary Condition**

The final factor that will be investigated is the velocity boundary condition which is input to the engine. The velocity boundary condition represents the vibration of the engine which causes the excitation to the enclosure. The constant boundary condition will provide a more uniform excitation in space and will excite modes more evenly. The enclosure will remain identical to the default in each case, only the boundary condition on the engine will change.

The results are shown in Figure 5.31 below. The overall insertion loss is similar for both cases. The constant boundary condition excites modes more evenly and settles at a slightly lower insertion loss. The real boundary condition case, on the other hand, does not excite all modes, but rather excites certain modes which are aligned with the excitation.

There is a great advantage to the designer in using a constant boundary condition. This would allow a designer to model the engine without having to know the exact behavior of the engine. The designer, then, would not need to perform measurements on the machinery, and the analysis could even be done before a product is ever built. A constant boundary condition is also easier to set up inside a boundary element program. The results for the constant boundary condition case show that, if one is merely interested in the overall insertion loss value, a very close approximation can be obtained by using a simple constant velocity boundary condition. If more detailed information is required about the behavior of the enclosure, actual velocity data is useful.
Figure 5.31 Comparison of Velocity Boundary Conditions on Engine
Chapter 6
Analysis of Results

Most enclosure data used in industry is still found from experiment. This involves a lot of time and resources. Being able to quickly judge the effect of changing a factor in the enclosure design before it is built will greatly reduce the amount of necessary testing. This will significantly reduce the cost and resources involved in design and shorten development time. Thus manufacturers can get the product to the consumer faster and hopefully deliver a better product.

The most important aspect of the enclosure performance to predict will be the overall insertion loss. Knowing the overall insertion loss will help to determine the effectiveness of the enclosure and to compare different potential designs. In this section, the results are analyzed and the precise effect of each factor on the overall insertion loss is examined.

The following figures show the effect of several parameters on the overall insertion loss of the enclosure. In order to obtain the best possible insertion loss, the most important factor to consider is the opening size. The enclosure designer should attempt to minimize the opening size of the enclosure. The designer should also make an effort to use as much absorption as possible and to place the noise source far from the opening. The enclosure size has merely a negligible effect on the insertion loss.
Figure 6.1 Effect of Enclosure Size on Overall Insertion Loss

Figure 6.2 Effect of Opening Size on Overall Insertion Loss
Absorption Coverage vs. IL

$y = 1.9274 \ln(x) + 2.8915$

$y = 2.3089 \ln(x) + 0.7583$

Figure 6.3 Effect of Absorption Coverage on Overall Insertion Loss

Source-to-Opening Distance vs. IL

Figure 6.4 Effect of Source-to-Opening Distance on Overall Insertion Loss
**Opening Size**

The opening size was found to have a logarithmic relationship to the overall insertion loss, as depicted in Figure 6.2 above.

The relationship found here is similar to that shown in the equation from Beranek’s textbook which involves a ratio of solid angles, thus reinforcing the preceding formula.

\[
IL = 10 \log \left[ 1 + \alpha_{ave} \left( \frac{\Omega_{tot}}{\Omega_{open}} - 1 \right) \right]
\]  
(6.1)

**Absorption Coverage**

The amount of absorption covering the inner surface of the enclosure also has a logarithmic effect on the overall insertion loss.

The logarithmic relationship again agrees with Beranek’s equation 6.1 above.

**Distance Between Source and Opening**

The relationship of the source-to-opening distance to the insertion loss is less distinct, so a detailed equation is not presented. The result will also depend on the directivity and the geometry of the source. The designer, however, should remember to place the source as far from the opening as is practical. This relationship is also represented in the equation by Beranek in terms of the solid angle.

\[
\Omega_{open} = \frac{S_{open}}{r^2}
\]  
(6.2)
Figure 6.5 Effect of Open Solid Angle on Insertion Loss
Chapter 7

Conclusions

7.1 **Summary and Conclusions**

The objectives of this research study were to determine the effects of eight factors on the performance of an acoustic enclosure. The eight factors that were examined were:

a. Opening Size
b. Opening Location
c. Absorption Coverage
d. Absorption Location
e. Absorption Coefficient
f. Enclosure Size
g. Source Location
h. Velocity Boundary Condition (Input Excitation)

The study focused on the airborne noise radiated from a free-standing partial enclosure in the frequency range from 0-1000 Hz. The study does not reflect actual cases because structureborne noise was not considered, but does reflect the effects of the factors under consideration.

Although there has been a lot of research on enclosures in the past, their complex nature makes them difficult to understand. There are no analytic models that provide results adequate for design purposes. The development of computer analysis, such as the BEM has greatly improved prediction methods, but sometimes requires large amounts of computer resources. The BEM has been used in this study to develop the results. The results will also help to ensure that fewer analysis cases need to be run in a computer analysis before a final design is achieved.

The research was conducted using SYSNOISE analysis software with a mesh created in I-DEAS. Enclosures were investigated with both a simple point source and also a real Cummins diesel engine as described in chapter 4. The eight factors were examined one at a time to determine their effects.

It was found that the primary factors affecting the overall insertion loss were the size of the openings, the amount of absorption, and the location of the source relative to
the opening. The effect of these factors is intuitive. However, the relative importance of
the opening size in comparison with the other factors is less intuitive. It was seen that the
opening size is of primary concern, since its effect on the overall insertion loss is much
greater than the other factors. Therefore, it is important for the designer of an enclosure to
achieve the smallest possible opening size, and then to attempt to cover the enclosure
with as much absorption as possible and to place the source as far away from the
openings as is feasible.

The results confirm the trends predicted by the handbook equation recorded by
Beranek (1992). The handbook equation, in turn, provides validation of the models
studied in this research.

The results developed from this study will help engineers to develop enclosures
with a great deal less experimental analysis. This will save money and resources and
shorten development time. It could also help to deliver a better product. The study also
revealed some suggestions for modeling enclosures and sources. It was seen that detailed
information about a source is not necessary for a boundary element model to obtain a
reasonable estimate of the overall insertion loss.

7.2 Future Work

This research produces valuable results within the scope of the project. There is,
however, plenty of room for the project to be extended. It would be interesting, for
example, to examine the structure-borne noise of the enclosure and also machine-
mounted enclosures. Other innovative solutions for attenuating the noise in enclosures
could also be considered, including Helmholtz resonators tuned to the nadir frequency,
baffles or partitions to block sound from openings, microperforate absorbers, or active
noise control.
Appendix

Sample SYSNOISE Command File (File to run the case with 1% Open Area with Engine Source)

New Name '1open' Model 1 File 1open.sdb Return
Option  BEM Indirect Variational Uncoupled Unbaffled Frequency  Return
Import Mesh Format Ideas File 1open.unv Return
Check Set Domain Return
Import Set Format Ideas File 1open.unv Return
Set 51 Name "Envelope" Envelope
   Elements All
   Return
Material Fluid
   Name 'air'
   Sound Real 343  Rho Real 1.21
   Return
Boundary Jump Pressure  Real 0 Imag 0
   Nodes Set 51
   Return
Table 1 Name 'imped' File imped.txt Return
Combine
Read Table 1 Return
Invers
Write Table 2 Name 'admit' Return
Return
Boundary Admittance Table 2 Positive
   Elements Set 12
   Elements Set 13
   Elements Set 14
   Elements Set 15
   Elements Set 16
   Elements Set 18
   Return

Boundary Admittance  Real .001 Imag 0
   Elements Set 3
   Elements Set 4
   Singular
   Return
Environment Section SETUP UNIVERSALFILE 'DATASET58' Return
Generate
   Element Set 1
   From Velocities File  baseline.d58 Format Ideas
   Frequency 20 To 1000 LinStep 20
Mesh File  engine_ibem.unv Format Ideas
Algorithm 1 Tolerance 0.050000 Average 8
Return

Parameter Model 1

Physical
Save Potentials Step 1

Save Results none

Store Results none
Return
Near 2
Far 5
Quadrature 3 3 2
Return
Save Return
Solve
Frequency 1 To 200 LinStep 1
Frequency 200 To 1000 LinStep 10
Return
Save Return

Point Sphere Radius 1 Divide 10 Return
PostProcess
Points All
Frequency 1 To 200 LinStep 1
Frequency 200 To 1000 LinStep 10
Near 2
Far 5
Quadrature 3 3 2 Positive
Save Results Step 1
Return
Combine
Read Power  FPPower Return
Write Function Name 'power_1open' File power_1open.txt Return
Return
Save Return

Exit Save Journal journal NoSave Models
BIBLIOGRAPHY


VITA

Amy Elizabeth Carter was born Amy Elizabeth Sandman on June 18, 1981 in Louisville, Kentucky. She received a Bachelor of Science degree in Mechanical Engineering from the University of Kentucky in May 2004. She was awarded the Outstanding Senior in Mechanical Engineering in 2004. She enrolled in the graduate school in the fall of 2003. She was awarded the Wethington Fellowship and the Margaret Ingells Fellowship in 2004 and the Presidential Fellowship in 2005.

Amy Elizabeth Carter