Sound Insulation of a Box

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Abstract

It is not always possible to modify the acoustic characteristics of a noise source in a surrounding environment and sound insulation solutions have to be thought. Beside wrappings, acoustic enclosures represent an efficient way to reduce the radiated noise from machinery.

Since the usual simplified models of the radiated sound power from a box which covers a noise source are not feasible to implement over an extended frequency range, designing such enclosures is not an easy step.

The goal of the present investigation is to assess simplified models and compare these theoretical results with experimental data. Measurements are made to gather the necessary data for evaluations and they employ different noise sources and porous sound-absorbing layers for a given acoustic enclosure.

The present work regards just the airborne sound, the structure-borne sound being discarded, whereas the efficiency of the enclosure is seen by the insertion loss it provides.
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CHAPTER 1

Introduction

An acoustic enclosure is a noise control mechanism which reduces the radiated sound from a noise source by modifying the transmission path between the source of the noise and the receiver.

Designed to insulate noisy machines or to protect people or noise sensitive equipments from high intensity noise, the acoustical enclosures provide both sound insulation and sound absorption. Masonry, metal, timber, plasterboard, glass and other alike heavy materials stop relatively significant amounts of sound and can be used in acoustic enclosures. The enclosure walls are usually multilayer composite systems, being composed of an interior sound-absorbing material and an exterior sound-impervious layer.

Furthermore, advanced acoustic enclosures frequently integrate vibration insulation and vibration absorption technologies. Typically, vibration attenuation requires resilient components such as springs, rubber mounts, air cushions, pads or fibreglass inserts – among others, in order to absorb energy.

There are many applications for acoustic enclosures, they being close related to engines, power generation devices, compressors and fans among many others.

Although there are some models describing the behaviour of acoustic enclosures, none of them extend accurately to the whole frequency range, discrimination being made between lower, intermediate and higher frequencies. According to the theory, the performance of an acoustic enclosure is measured by its insertion loss. Unluckily, most of the models of the radiated sound power from an enclosure suffer from shortcomings that undermine their usefulness.
Some of the limitations in matching the theoretical models to the actual data could come from the assumptions made. Thus, Goesele’s model [2] assumes that an enclosed source radiates a different amount of power with respect to the free space case. Henning [2] goes further considering that the sound power loss inside the enclosure equals the sound power in free space.

The aim of the present work is to find and examine models of the insertion loss to be used when designing covering boxes for machine installations. Therefore, measurements of real situations will be performed in order to examine the different models. In the following study cases, structure-born sound is ruled out, the attention being focused on air-borne sound control.
CHAPTER 2

Method and theory

2.1 Fundamentals of sound insulation

Before going into a deep analysis of sound insulation provided by an acoustical enclosure, the basic concepts and underlying theory pertaining to them have to be clearly sorted out.

Beside fundamental definitions, this section is intended to define different types of acoustic enclosures and present the corresponding approximations. The approximations are made according to the relationship between the frequency content of the signal under analysis and the designing aspects such as dimensions and significant openings, but taking into account assumptions regarding the sound source, absorption, environment (which has an impact on the sound field characteristic).

2.1.1 Insertion loss and the performance rating of acoustical enclosures

The efficiency of an acoustic enclosure is best described by the insertion loss \( D_i \) it provides which represents a decrease in the radiated sound power due to the insertion of an acoustic enclosure on the sound source under consideration.

The insertion loss measured in decibels is a function of the characteristics and geometry of the material of the enclosure panel and the existing openings and it is defined as the logarithmic ratio of the effective sound power radiated by the noise source in free space \( W \) to the sound power radiated by the enclosed source \( W_i \).
\[ D_e = 10 \log \left( \frac{W}{W_e} \right) = -10 \log \tau_e \] (1)

where \( \tau_e = \frac{W_e}{W} \) is the insertion power ratio.

For convenience, expressing the sound power as sound power level yields to an insertion loss of the form

\[ D_e = L_W - L_{W_e} \] (2)

where

\[ L_W = 10 \log \frac{W}{W_{\text{ref}}} \] (3)

\[ L_{W_e} = 10 \log \frac{W_e}{W_{\text{ref}}} \] (4)

Above, \( L_W \) and \( L_{W_e} \) are the corresponding sound power levels of \( W \) and \( W_e \) respectively, the reference sound power being \( W_{\text{ref}} = 1 \text{ pW} \).

A rule of thumb says that 10 dB attenuation in many daily life sounds yields to a halving in the perceived loudness and consequently an insertion loss of 10 dB is equivalent to a 50% reduction in the perceived loudness, where by loudness is understood the subjective perception of the strength of a sound. That is, a 10 dB loss in power should make an obvious impact on the amount of noise emitted by a noise source after insertion of an acoustical enclosure.
2.1.2 Classification of acoustic enclosures

From the constructive point of view [1], the acoustic enclosures are classified as free standing or equipment mounted, whereas the presence or absence of substantial acoustical leaks divide them in leaky and sealed; when the volume of the enclosure is comparable to the volume of the machine, the enclosure is said to be close fitting.

In respect to the acoustical and bending wavelength, an enclosure can be called acoustically small if both the acoustical and bending wavelength are large compared with the largest inner dimension of the enclosure volume and largest wall panel dimensions, respectively. Conversely, any enclosure is said to be acoustically large if all of its dimensions are large in respect to the acoustical wavelength and it exhibits a large number of acoustical resonances in the frequency range of interest. This being said, a given enclosure may be regarded as either small or large function to the different frequency ranges. Therefore, for low frequencies where neither the enclosure wall panels nor the interior gas volume exhibit any resonances the small enclosure theory has to be applied, and the insertion loss is controlled by the ratio of the volume compliance of the enclosures walls and the volume of the enclosed gas volume.

Consequently, for high frequency region, where both the enclosure wall panels and the interior gas volume exhibit a large number of acoustical resonances, large enclosure description is appropriate. In this case the statistical methods for predicting the sound field inside the enclosure and the transmission of the sound through the enclosure walls are applicable [5]. The insertion loss is mainly controlled by the sound transmission loss of the enclosure wall panels and by the interior sound absorption.

For the intermediate frequency region where either the enclosure walls or the enclosure gas volume, or both exhibit resonances that do not overlap is difficult to have a precise anticipation of the insertion loss. Usually finite-element analy-
sis is employed to cover this range, but coarse estimations that connect the low-
and high-frequency region are well suited as well.

In the following, small and large sealed free-standing acoustical enclosures are
treated. Also the air-borne excitation of the enclosure walls is taken into consid-
eration, the structure-born paths being discarded from the investigation.

**Small, free standing, sealed acoustical enclosure**

A small free standing enclosure has no mechanical connections to the enclosed
vibrating equipment and no acoustical resonances in the interior volume. Be-
sides, if the largest dimension of the interior of the enclosure is less then one
tenth of the acoustical wavelength, the sound pressure is equally distributed
inside the volume. Thus, in low frequency-modelling the acoustic impedance of
the volume of air enclosed in a sealed (lossless) enclosure is represented by a
compliance. However, a physical enclosure presents an acoustical impedance
that exhibits a resistance, a mass and a compliance, all of them depending on
the enclosure characteristics.

At low frequencies, a monopole represents a good approximation to any source
that produces a net displacement of volume velocity, the sound power radiated
being given by the expression

$$W = \frac{\rho ck^2 |Q|^2}{8\pi}$$

where $W$ denote the sound power radiated by the monopole, $Q$ is the volume
velocity of the source, $\rho$ is the density of the medium (air), $c$ speed of sound and
$k$ the wave number.

Recalling the way insertion loss is defined, and assuming the source and box as
monopoles, one notice immediately that a change in the radiated power corre-
sponds to a change in the apparent volume velocity for the source itself and for
the box containing the source.
Therefore, from equations (1) and (5), the insertion loss at low frequencies can be rewritten as

\[ D_e = 10 \log \frac{W_e}{W} = 20 \log \frac{Q_S}{Q_B} \]  (6)

where \( Q_S \) represents the volume velocity of the sound source and \( Q_B \) the volume velocity of the box.

Such a system representing a sound source boxed in can be simply modelled by impedance analogous circuits. For easiness, we consider just one wall panel of the box. In the general situation when all enclosure wall panels are considered, the impedance \( Z_{W_i} \) and compliance \( C_{W_i} \) describing a single wall panel are replaced by the corresponding equivalent values that take into account the whole box. Since the enclosure wall panels act as compliances connected in parallel, the equivalent compliance of the box has the form

\[ C_B = \sum_{i=1}^{n} C_{W_i} \quad m^5 / N \]  (7)

whereas the equivalent impedance becomes

\[ Z_{eq} = \frac{1}{\sum_{i=1}^{n} Z_{W_i}} \]  (8)

where \( i \) designates the corresponding enclosure wall panel.
Figure 2. Impedance analogous circuit for a box covering a sound source. $C_V$ is the volume compliance of the cavity volume, $C_W$ is the compliance of the box panel and $Q_V$ and $Q_B$ are their corresponding volume velocities. $p_c$ designates the pressure inside the cavity, whereas $Q_S$ represents the volume velocity of the source.

The compliance of the gas within the enclosure cavity $C_V$ is defined as

$$C_V = \frac{V_0}{\rho c^2} \, \text{m}^5 / \text{N}$$  \hspace{1cm} (9)

with $V_0$ representing the volume of the enclosure gas, $\rho$ the density of the gas, and $c$ the speed of sound in that gas.

The volume compliance of the $i$th wall panel is

$$C_{W_i} = \frac{\Delta V_{pi}}{p_c} \, \text{m}^5 / \text{N}$$  \hspace{1cm} (10)

where $\Delta V_{pi}$ represents the volume displacement of the corresponding $i$th enclosure wall panel when it is exposed to the uniform pressure $p_c$ from inside cavity.

Taking into account Figure 2, the impedance of the volume cavity $Z_V$ and that of the wall panel $Z_W$ can be written as

$$Z_V = \frac{1}{i\omega C_V}$$

$$Z_W = \frac{1}{i\omega C_W}$$  \hspace{1cm} (11)

where the following substitution was made $Q_V = Q_S - Q_B$ (this is easily seen using the impedance analogous circuit displayed in Figure 2, the volume velocity...
of the source $Q_s$ being the sum of the other two volume velocities $Q_V$ and $Q_B$ presented in the circuit). Solving the last set of equation for the ratio between the volume velocities one gets

$$\frac{Q_B}{Q_s} = \frac{1}{C_V + 1}$$ (12)

Making use of equation (6), the insertion loss $D_e$ becomes

$$D_e = 20\log \left(1 + \frac{C_V}{C_W}\right)$$ (13)

If the whole box is taken into account, the insertion loss of this sealed acoustical enclosure is immediately revealed by replacing the single panel compliance from equation (13) with that representing the equivalent compliance of the entire box:

$$D_e = 20\log \left(1 + \frac{C_V}{C_B}\right) = 20\log \left(1 + \frac{C_V}{\sum_{i=1}^{n} C_{Wi}}\right)$$ (14)

This is the insertion loss that R.H.Lyon [8] found considering a small rigid rectangular enclosure with a flexible wall. Thus, for such a sealed acoustical enclosure, the insertion loss is controlled just by the volume compliance of the enclosed gas volume $C_V$ and the volume compliance of the enclosure wall panels $C_{Wi}$ and is applicable just to very low frequencies.

However, if one wants a model that stretches longer toward higher frequencies, a mass term $M_W$ should be added to the wall impedance in equation (11). Therefore, the equation (11) can be reformulated as

$$\begin{align*}
\frac{p_e}{Q_s - Q_B} &= Z_V = \frac{1}{i\omega C_V} \\
\frac{p_e}{Q_B} &= Z_W = \frac{1}{i\omega C_W} + i\omega M_W
\end{align*}$$ (15)
The last set of equations yields to a volume velocities ratio and an insertion loss of the form

\[
\frac{Q_B}{Q_S} = \frac{Z_V}{Z_W + Z_V}
\]  \hspace{1cm} (16)

\[
D_E = 20\log \frac{Z_W + Z_V}{Z_V}
\]  \hspace{1cm} (17)

In order to determine the enclosure wall panel impedance, the equation of motion for bending waves for thin and homogenous plates is used. Taking for granted that the applied pressure \(p_c\) is uniformly distributed, the bending wave equation becomes [10]

\[
B\Delta^2 w(x, y) - m' \omega^2 w(x, y) = p_c
\]  \hspace{1cm} (18)

where \(w(x, y)\) is the displacement response, \(B\) represents the wall panel bending stiffness per unit width, \(m'\) is its mass per unit area, \(\omega\) is the angular frequency, and \(\Delta\) is the Laplace operator.

Assuming the plate being simply supported and applying a modal solution, the displacement can be expressed as

\[
w(x, y) = \sum_{n=1}^{N} w_n \varphi_n(x, y)
\]  \hspace{1cm} (19)

where \(w_n\) is the complex amplitude of the displacement, whereas \(\varphi_n(x, y)\) represents the \(n\)th mode shape of the wall panel

\[
\varphi_n(x, y) = \sin(\alpha_n x) \sin(\beta_n y)
\]  \hspace{1cm} (20)

The last equation is satisfied by the following set of wave numbers

\[
\alpha_n = \frac{m \pi}{L_x}, \quad \beta_n = \frac{m \pi}{L_y}
\]  \hspace{1cm} (21)

where \(L_x\) and \(L_y\) are the wall panel dimensions, whereas \(n_x, n_y = 1, 2, 3, \ldots\)

Substituting the equations (20) and (19) in equation (18), the bending wave equation reduces to

\[
\sum_{n=1}^{N} \left( B(\alpha_n^2 + \beta_n^2) - m' \omega^2 \right) \varphi_n(x, y) w_n = p_c
\]  \hspace{1cm} (22)
Next, the equation is multiplied with $\varphi_m(x, y)$ on both sides and then is integrated over the surface of the wall panel.

$$\sum_{n=1}^{N} \left( B(\alpha_n^2 + \beta_n^2) - m' \omega^2 \right) \int_S \varphi_m(x, y) \varphi_n(x, y) \, dS \omega_n = p_c \int_S \varphi_m(x, y) \, dS \quad (23)$$

The last expression is then simplified making use of the orthogonality relations

$$\int_S \varphi_m(x, y) \varphi_n(x, y) \, dS = \Lambda \delta_{mn} \quad (24)$$

where $\Lambda$ is the so-called norm, and $\delta_{mn} = \begin{cases} 0, & \text{if } m \neq n \\ 1, & \text{if } m = n \end{cases}$.

The complex amplitude of the displacement $w_n$ transforms to

$$w_n = \frac{p_c \int_S \varphi_n(x, y) \, dS}{\Lambda} \quad (25)$$

Utilizing equations (25) and (19), the displacement $w(x, y)$ is found to be

$$w(x, y) = \frac{p_c}{\Lambda} \sum_{n=1}^{N} \frac{\int_S \varphi_n(x, y) \, dS}{B(\alpha_n^2 + \beta_n^2) - m' \omega^2} \quad (26)$$

Once the displacement field is obtained, the volume velocity is easy calculated by integrating the displacement over the appropriate surface of the wall panel and multiplying with $i \omega$.

$$Q = i \omega \int_S w(x, y) \, dS = i \omega \frac{p_c}{\Lambda} \sum_{n=1}^{N} \frac{\left( \int_S \varphi_n(x, y) \, dS \right)^2}{B(\alpha_n^2 + \beta_n^2) - m' \omega^2} \quad (27)$$

Consequently, the corresponding wall panel impedance becomes

$$Z_W = \frac{p_c}{Q} = \frac{\Lambda}{i \omega} \sum_{n=1}^{N} \frac{\left( \int_S \varphi_n(x, y) \, dS \right)^2}{B(\alpha_n^2 + \beta_n^2) - m' \omega^2} \quad (28)$$

For $n=1$, that is considering just the first mode, the volume velocity and the impedance of the wall panel transform to:
In order to simplify these expressions, some more calculations are to be made.

Using the equations (20), (21), and (24), the norm factor $\Lambda$ can be obtained:

$$
\Lambda = \left( \int_{S} \varphi_m(x, y) dS \right)^2 = \int_{0}^{L_x} \sin^2(\alpha_n x) dx \int_{0}^{L_y} \sin^2(\beta_n y) dy = \frac{L_x L_y}{4} 
$$

Also, further reductions in expression (29) become applicable:

$$
\int_{0}^{L_x} \sin(\alpha_n x) dx = \frac{1 - \cos(\alpha_n L_x)}{\alpha_n} 
$$

$$
\int_{S} \varphi_m(x, y) dS = \int_{0}^{L_x} \sin(\alpha_n x) dx \int_{0}^{L_y} \sin(\beta_n y) dy = \frac{L_x L_y}{\pi^2 n_x n_y} \left[ 1 - (-1)^{n_x} \right] \left[ 1 - (-1)^{n_y} \right] 
$$

Recalling that only the first mod is considered, that is $n_x = n_y = 1$, the wall panel impedance reduces to

$$
Z_W = \frac{1}{i \omega} \frac{\pi^4}{4^3 L_x L_y} \left( B \pi^4 \left( \frac{1}{L^2} + \frac{1}{L_y^2} \right)^2 - m' \omega^2 \right) 
$$

The last equation conveys the wall panel impedance when its mass is taken into account in order to get a broader applicability of the low frequency model (to extend it more toward the high frequency region). If the contribution of the mass term is discarded, the wall panel impedance reduces to a shorter form which represents the R.H.Lyon case (see equation (11)) when no mass component was taken into account:
If the ratio of the wall panel dimensions is \(a\), equation (34) yields to

\[
Z_w = \frac{1}{i\omega} \frac{B\pi^8}{4^3} \left( \frac{1}{L_x^2} + \frac{1}{L_y^2} \right)^2 = \frac{1}{i\omega} \frac{B\pi^8}{4^3} \left( \frac{L_y}{L_x} + \frac{L_x}{L_y} \right)
\]

(34)

Substituting the wall panel impedance given by the last equation into equation (11), the volume compliance of the wall panel becomes

\[
C_w = \frac{S^3}{B} \cdot 10^{-3} F(a) \quad m^5 / N
\]

(36)

with

\[
F(a) = \frac{4^3}{\pi^8} \left( \frac{a^{-1} + a}{} \right)^2 10^7 \approx \frac{6.75}{\left( a^{-1} + a \right)^2}
\]

(37)

where \(S\) represents the surface area of the panel, \(B\) denotes the bending stiffness per unit width, \(F(a)\) is the so-called plate volume compliance function, and \(a\) is the ratio of the edge dimensions of the enclosure wall panel.

![Figure 3. Plate volume compliance function for homogenous isotropic panels with simply supported edges](image-url)
By going back to the simpler Lyon’s case (low frequency approximation where the impedance of the sound source-box ensemble is given using just compliances as seen from equation (11)) and combining the equations (37), (36), (9), and (14), the insertion loss for an enclosure with orthogonal dimensions is of the form

\[
D_i = 20 \log \left[ 1 + \frac{V_s E h^3}{12 \times 10^{-3} \left( 1 - \mu^2 \right) \rho c^2} \sum_{i=1}^{5} \frac{1}{S_i^2 F(a_i)} \right] \quad dB \quad (38)
\]

where the indices \(i\) designate the number of the wall panels.

The insertion loss can be simplified if the enclosure has a cubic shape of edge length \(a\), last equation yielding to

\[
D_i = 20 \log \left[ 1 + 41 \left( \frac{h}{a} \right)^3 \frac{E}{\rho c^2} \right] \quad dB \quad (39)
\]

with \(h\) denoting the thickness of the enclosure wall plate, \(E\) the Young’s modulus and \(\mu\) the Poisson ratio of the enclosure wall.

Analysing the insertion loss for such a cubic enclosure, it is easier to see that higher thickness-to-edge dimension ratio generates higher insertion loss.

However, in order to see whether the theory is applicable for a certain frequency range, both the resonance frequencies of the box and wall panels have to be checked. The natural frequencies of the box (and wall panels as well) are determined using the expression:

\[
f_n = \frac{c}{2} \sqrt{\left( \frac{n_x}{L_x} \right)^2 + \left( \frac{n_y}{L_y} \right)^2 + \left( \frac{n_z}{L_z} \right)^2} \quad (40)
\]

where \(n_x, n_y,\) and \(n_z\) represent the nodes (they are integers greater or equal to 0, but at least one of them has to differ from 0 ), whereas \(L_x, L_y,\) and \(L_z\) stand for the dimensions of the box. Since the wall panels are 2-dimensional plates, the above equation reduces with one term (representing the third dimension).
The resonance frequency of the ensemble noise source-box can easily be calculated by equalizing the total impedance with 0. Having in mind that the equivalent impedance of all five wall panels composing the box is given by the equation (8), the resonance condition for the sound source-box ensemble becomes:

\[ Z_{eq} + Z_V = 0 \]  

(41)

Regarding the more accurate case when the wall panel impedance contains also a mass term as shown by equation (33), the resonance condition gets a very complicated form:

\[
\frac{1}{i\omega \sum_{i=1}^{5} \pi^4 \left( B\pi^4 \left( a + a^{-1} \right)^2 - m' \omega_0^2 S_i^2 \right)} + \frac{\rho \omega^2}{i\omega V_0} = 0
\]

(42)

Considering that \( f_0 = \omega_0 / (2\pi) \), the resonance frequency is then obtained by solving the last equation.

If we conceive just a single wall panel described by equation (33), the resonance condition \( Z_W + Z_V = 0 \) yields to a resonance frequency of the form

\[
f_0 = \frac{1}{2\pi} \sqrt{\frac{B\pi^4 \left( a + a^{-1} \right)^2 + \rho \omega^2 S^2}{S^2 V_0 \pi^2}}
\]

(43)

Since at low frequencies neither the enclosure wall panel nor the interior air volume must exhibit any resonances, equations (40) and (43) allow us to find the break point frequency for the low frequency approximations.

**Large, free standing, sealed acoustical enclosure**

By definition, a free standing sealed enclosure [1] has no mechanical connections to the enclosed vibrating equipment and no substantial acoustical leaks. The term large is given by the frequency range employed which gives rise to a large number of resonant modes in both the enclosure volume and enclosure wall panels. For such enclosures, the level of the interior sound field and sound
radiation of the enclosure wall panels are predicted using statistical methods of room acoustics [5].

For investigating the acoustical conduct of large, free-standing, sealed acoustical enclosures the space-time average mean-square value of the sound pressure within the enclosure is calculated. Next the sound power radiated is determined. For a large, free standing, sealed acoustical enclosure the insertion loss in high-frequency region [1] is provided by the following expression

\[ D_e \cong 10 \log \left( 1 + \frac{S_w \alpha_w + S_i \alpha_i}{S_w} \cdot 10^{R_w/10} \right) \quad dB \]  

(44)

where \( S_w \) represents the total interior wall surface, \( \alpha_w \) denotes the average absorption coefficient of the enclosure wall panels, \( S_i \) is the interior surface in addition to the walls (for example, the machine body itself) and \( \alpha_i \) is the corresponding average absorption coefficient of this additional surface, \( R_w \) stands for the sound transmission loss of the walls.

If one considers that the value of the second term is much greater than 1, the last equation can be rewritten as

\[ D_e \cong R_w + 10 \log \frac{S_w \alpha_w + S_i \alpha_i}{S_w} \quad dB \]  

(45)

where the numerator of the logarithm represents the total equivalent absorption area in the interior of the enclosure. Calling \( A \) this total interior absorption area, the insertion loss reduces to

\[ D_e \cong R_w + 10 \log \frac{A}{S_w} \quad dB \]  

(46)

Thus, this expression shows that the insertion loss of a large, free-standing, sealed acoustical enclosure can approach the sound reduction index of the enclosure wall panels when the average absorption coefficient approaches 1. In other words, the absorption in the cavity is of importance for high-frequency modelling.
Considering the dimension of the machine body (noise source) negligible, the acoustical performance of the enclosure is then evaluated as

\[ D_e = R_w + 10 \log \alpha \quad (47) \]

Since the sound transmission index is defined as \( R_w = 10 \log \frac{W_e}{W} \), the last equation rewrites as

\[ D_e = 10 \log \left( \frac{W_e}{W} \right) + 10 \log \alpha \quad (48) \]

where \( W_w \) denotes the sound power radiated by the source boxed in (which is supposed to be different from the sound power \( W \) radiated by the source in free space) and \( \alpha \) stands for the sound absorption coefficient of the interior porous absorbing material which is flush with the enclosure wall.

The negative part of this high-frequency model lays in the fact that for very small absorption coefficients (\( \alpha \to 0 \)) the insertion loss tends to minus infinite (\( D_e \to -\infty \)).

Figure 4 Source radiating in free space (left); source surrounded by an enclosure lined with sound absorbing material (right).
2.2 Evaluation of the radiated sound power from boxed in sources

Since the insertion loss of an enclosure, a parameter which indicates its acoustical efficiency, shows the loss of power which results by enclosing the noise source, the different models of the insertion loss have to be discussed.

2.2.1 Sound transmission loss

A fast assessment of the high frequency approximations is achieved by making use of the mass law [5] which represents one of the easiest and most important ways of control for the airborne sound transmission through a plate when just the mass $m'$ per unit area of the plate is known.

If a plane sound wave hit a homogenous, isotropic plate at normal incidence, all points of the plate will move in phase at all frequencies and the bending stiffness can be neglected.

Under such circumstances, the sound transmission loss of the plate $R_w$ is given by

$$ R_0 = 20 \log \left( \frac{om'}{2 \rho c} \right) \text{ dB} $$

which can be approximated to

$$ R_0 \approx 20 \log \left( \frac{m'}{1 \text{ kg/m}^2} \right) + 20 \log \left( \frac{f}{100 \text{ Hz}} + 2 \text{ dB} \right) $$

Beside mass $m'$, the transmission loss is function of the working frequency.

If the forced transmission is considered, the critical frequency $f_c$ and the radiation efficiency $\sigma_{fw}$ have to be added to the resultant transmission loss $R_w$:

$$ R_w = R_0 + 10 \log \left[ \left( 1 - \left( \frac{f}{f_c} \right)^2 \right)^2 + \eta^2 \right] - 10 \log \left( 2 \sigma_{fw} \right) $$

where $\eta$ is the loss factor.

---

1 The critical frequency is the frequency at which the speed of a bending wave in a plate equalizes the speed of a sound wave in air.
The critical frequency $f_c$ is defined as

$$f_c = \frac{c^2}{\pi h} \sqrt{\frac{3 \rho_m (1 - \mu^2)}{E}}$$  \hspace{1cm} (52)$$

with $c$ being the speed of sound in air, $h$ the thickness of the wall panel, $\rho_m$ the density of the constituting wall material, $\mu$ the Poisson ratio, and $E$ the Young’s modulus.

The radiation efficiency of a single wall panel $\sigma_{fw}$ due to force vibrations [5] is given by

$$\sigma_{fw} = \frac{1}{2} (0.2 + \ln(ke))$$  \hspace{1cm} (53)$$

where $k$ is the wave number and $e$ is a parameter that characterizes the size of a plate with $L_x$ and $L_y$ edge dimensions

$$e = \frac{2 \cdot L_x \cdot L_y}{L_x + L_y}$$  \hspace{1cm} (54)$$

If the entire box is considered, an approximation can be made for the radiation efficiency $\sigma_f$ of all constituent wall panels, namely it is assumed to be the sum of the radiation efficiency $\sigma_{fw}$ of each wall panel.

### 2.2.2 Mechel’s insertion loss model

Making use of the theory earlier described, a simple model of the radiated sound from the box is introduced.

Slightly different, Mechel’s model [2] takes for granted that the sound power $W_v$ loss inside the capsule equals the effective sound power $W$ radiated in free space at equilibrium, thus yielding to an absorption coefficient carrying the radiated power

$$\alpha = \frac{W_v}{W}$$  \hspace{1cm} (55)$$

and to an insertion power ratio of the form:
\[ \tau_v = \frac{|t_v|^2}{1 - |r_v|^2} \]  
\[ (56) \]

where \( t_v \) is the transmission factor through the enclosure wall, whereas \( r_v \) represents the interior reflection factor of it.

Using equation (1) and (56), one can determine the efficiency \( D_e \) of a large enclosure with plane walls.

Assuming plane waves at an angle of incidence \( \theta \) (see Figure 5) and considering the enclosure lined with a porous interior layer (flush with the outer rigid plate with partition impedance \( Z_W \)), having the characteristic propagation constant \( \Gamma_a \), the characteristic impedance \( Z_a \) and thickness \( d_a \), one get the insertion power ratio \( \tau_v \) of the form

\[ \tau_v = \left| 4b \cdot e^{\frac{-1}{2}} \right|^2 \cdot \left[ (1+b)^2 - (1-b)^2 \cdot e^{-2a} + b \cdot (1+b + (1-b) \cdot e^{-2a}) \cdot Z_W / Z_0 \right]^2 - \left[ (1-b^2)(1-e^{-2a}) + b \cdot (1-b + (1+b) \cdot e^{-2a}) \cdot Z_W / Z_0 \right]^2 \]  
\[ (57) \]

with the abbreviations:

\[
\begin{align*}
    a &= \Gamma_a d_a \cdot \cos \theta_a = k_0 d_a \sqrt{\Gamma_a^2 + \sin^2 \theta} \\
    b &= \frac{Z_0 \cos \theta_a}{Z_a \cos \theta} = \frac{\sqrt{\Gamma_a^2 + \sin^2 \theta}}{\Gamma_a Z_a \cos \theta} \\
    \Gamma_a &= \frac{\Gamma_a}{k_0} \\
    Z_a &= \frac{Z_a}{Z_0}
\end{align*}
\]  
\[ (58) \]

In the above expressions, \( k_0 \) is the wave number and \( Z_0 \) denotes the characteristic impedance of air obtained from

\[ Z_0 = \rho_0 c_0 \quad (\text{Pa} \cdot \text{s}) / \text{m} \quad \text{or} \quad Z_0 = \frac{7064}{\sqrt{T}} \quad (\text{Pa} \cdot \text{s}) / \text{m} \]  
\[ (59) \]

where \( T \) represents the registered temperature expressed in Kelvin during measurements, \( \rho_0 \) the density of air and \( c_0 \) the sound speed in air.
However, for three-dimensional diffuse sound field the insertion power ratio $\tau_e$ [2] reduces to

$$\tau_{e-3\text{diff}} = \frac{2}{\sin^2 \theta_{h/2}} \int_0^\theta_e (\theta) \cos(\theta) \sin(\theta) d\theta$$

(60)

Substituting equation (60) in (57) and then using equation (1), one immediately determines the insertion loss $D_e$ of the enclosure when the diffuse field is assumed.

![Diagram of porous interior layer](image)

**Figure 5.** Porous interior layer of thickness $d_a$ attached to an enclosure wall panel of thickness $h$. The interior incident plane wave $p_i$ is partly reflected and partly transmitted to the exterior through the absorbing layer and the wall panel.

### 2.2.2 Characteristic values calculation

The previous section has presented some theoretical possibilities of predicting insertion loss. In the attempt to compare the earlier mentioned insertion loss models with experimental data, the characteristic values $\Gamma_a$ and $Z_a$ of the absorbing interior layer have to be derived. Different theoretical models of porous material, e.g. Delany-Bazley’s model [7], return good empirical results. Mechel’s model theory [2] evaluates the characteristic values from:
\[
\Gamma_{an} = j \sqrt{\frac{\rho_{\text{eff}}}{\rho_0} \cdot \frac{C_{\text{eff}}}{C_0}} \tag{61}
\]
\[
Z_{an} = \frac{1}{\sigma} \sqrt{\frac{\rho_{\text{eff}}}{\rho_0} \cdot \frac{C_{\text{eff}}}{C_0}} \tag{62}
\]

with
\[
\begin{align*}
\frac{\rho_{\text{eff}}}{\rho_0} &= \chi - j \frac{\sigma \cdot \xi(G)}{2\pi G} \\
\frac{C_{\text{eff}}}{C_0} &= \frac{k + \alpha_1 \cdot \frac{jG}{G_0}}{1 + \frac{jG}{G_0}} \\
\xi(G) &= \gamma_0 + \gamma_1 G + \gamma_2 G^2 \\
G &= \frac{\rho_0 f}{\Xi}
\end{align*}
\] (63)

where \( \rho_0 \) is the density of air, \( f \) represents the frequency, \( \Xi \) stands for flow resistivity, \( \chi \) is the structure factor, \( \sigma \) denotes the porosity, \( k \) represents the adiabatic exponent of air, and \( \alpha_1, \gamma_0, \gamma_1, \gamma_2, G_0 \) are other given coefficients.

All these coefficients describing the different types of porous absorbing materials employed in the experimental work are specified in the Appendix.

Later on, the characteristic values provided by these relations are used in the insertion loss models mentioned in the previous section.

### 2.2.3 Enclosure wall panel impedance

For an angle of sound incidence \( \theta \), the enclosure wall panel impedance at a frequency \( f \) is given by [5]

\[
Z_W = j \omega m' \left( 1 - \left( \frac{f}{f_c} \sin^2 \theta \right)^2 \right) + \eta \omega m' \tag{64}
\]

where \( \omega \) is the angular frequency, \( m' \) is mass per unit area, \( \eta \) is the loss factor, and \( f_c \) is critical frequency defined as:
\[ f_c = \frac{c^2}{\pi h} \sqrt{\frac{3 \rho_m (1 - \mu^2)}{E}} \]  

with \( c \) being the speed of sound in air, \( h \) the thickness of the wall panel, \( \rho_m \) the density of the constituent wall material, \( \mu \) the Poisson ratio, and \( E \) the Young’s modulus.

If diffuse field is assumed, the expression of the wall impedance simplifies to

\[ Z_W = j \omega m' \left( 1 - \left( \frac{f}{f_c} \right)^2 \right) + \eta \omega m' \]  

This wall panel impedance is then used in the prediction of insertion loss according to Mechel’s model (presented in section 2.2.2).

### 2.3 Measurements

The theoretical models of insertion loss are based on assumptions that can never be completely realized in reality. Measurements have to be conducted in order to check how well the empirical values of the insertion loss fit to the theoretical models, this section being a review of the measurements performed.

The experimental insertion loss is determined mainly by equation (1), but the absorption coefficient of the interior porous layer and the reverberation time of the reverberation room have to be employed in calculations due to the requirements of the different models. Since the characteristic values describing the interior absorbing layer were introduced before by means of equations (61), (62), and (63), beside sound power measurements, sound reverberation time and absorption coefficient measurements have to be completed.

The measurements in question are conducted in a special reverberation room.
2.3.1 Sound power measurements

In order to assess the insertion loss, the band-power levels of a source (both un-enclosed and enclosed source) have to be measured out.

A very convenient way of doing this is by measuring the sound pressure it generates using a moving microphone traversing a path at constant speed in a reverberation room [3]. A scheme of the measurement setup can be seen below.

![Diagram of measurement setup]

The noise source engaged in tests is the Sound Power Source Type 4205 produced by B&K which consists of a generator and a sound source (HP 1001). The B&K Real-Time Frequency Analyzer Type 2133 is chosen to generate a pink noise used to drive the sound source (HP 1001) by means of its generator. A condenser microphone mounted on a rotating Microphone Boom Type 3923 is used for measuring the spatial average octave-band sound pressure levels. Measurements are carried out for 6 different microphone boom positions with an observation time of 64s. The frequency range of interest includes those one octave bands with centre frequencies from 125Hz to 8kHz.
The measuring system is calibrated by means of a Sound Level Calibrator Type 4231 from B&K.

During measurements, the sound source under tests (having a distance of at least 1 m to the nearest wall) is placed on the floor. Once the generation of pink noise is initiated, a measurement can start by a simply keystroke. After a completion of a measurement and before each series of new ones the calibration of the measuring system is verified.

The band-pressure levels for each octave band corresponding to each of the 6 positions (depicted by the indices p1, p2, …, p6) selected are used to determine the mean octave band level and then the band-power level (see Appendix), the results being presented in the Tables 1-8.

2.3.2 Reverberation time measurements

The reverberation time of the room was obtained using the interrupted noise method.

The room was driven with pink noise generated by a frequency analyzer type 2133 and the sound pressure in the room recorded by means of an omni-directional microphone type 4152 connected to the input of the analyzer.

The analyzer turned the noise signal on and off (after the sound reached a steady-state inside the room), recorded the decaying sound pressure level in octave bands as a function of time, averaged the sound pressure on a mean-square basis over a number of 10 decays and showed the resulting decay functions from which the reverberation time in each frequency band was determined from the rate of decay. The reverberation time was measured at six different positions in the room in octave bands from 125Hz to 8kHz by averaging over these decays.
2.3.3 Absorption coefficient measurements

The absorption coefficients for normal sound incidence for two sound absorbing materials were measured [4] by means of a standing wave apparatus type 4002. A diagram of the measurements set-up is shown in Figure 7.

![Diagram of the measurements set-up](image)

**Figure 7. Measuring set-up for measuring the acoustic absorption coefficient for zero degree incident sound**

The sine generator type 1023 feed the loudspeaker placed in one end of the standing wave tube with pure tones in the working frequency range. In the other end of the tube, the sample of the absorbing material which is mounted in the sample holder reflects the incident plane sinusoidal wave so that a standing wave pattern is produced due to the interference between the incident wave and the wave reflected by the sample. The maximum and minimum sound pressure levels are measured with a microphone probe attached to a car which slides along a moveable graduated ruler. The microphone output voltage is picked up by the real time frequency analyzer type 2133.

In order to cover a larger frequency range, the measurements were carried out with two measuring tubes with different diameters, the larger one (100 mm) for octave band measurements at 125Hz, 250Hz, 500Hz and 1kHz, whereas the tube with a smaller diameter (30 mm) was used for 2kHz, 4kHz and 6.5kHz pure tones measurements. Due to the apparatus limitations (it can work in the
frequency range from 90Hz to 6.5kHz), the absorption coefficient at 8kHz could not be determined. However, in the Analysis section and further the measurements results at 6.5kHz were also assigned to 8kHz. Even though in reality (in figures) the absorption at 6.5kHz differs from that at 8kHz, the absorption is very high at these high frequencies (tending to 1), so that the approximation before mentioned can be taken (without risking to obtain unreliable result).

The characteristic impedance and the propagation constant of the absorber materials were not determined using the standing wave apparatus, but they were predicted by means of Mechel’s model presented in section 2.2.2 [2]. The empirical method was discarded because a pressure-release termination could not be realized for the whole frequency range due to a missing sample holder with variable depth for one of the measuring tubes.
CHAPTER 3

Results of measurements

Previously, some theoretical models of insertion loss and appropriate measurement methods used to gather data for comparison with these models were presented. The current section will present the outcome of the experimental work completed for assessing the above mentioned theories. All input data and the corresponding measurements setups are given in the Measurements section and Appendix.

As mentioned in section 2.3.1, the sound power of the sound source was found out by measuring the sound pressure it generates in the reverberation room. Since the B&K Sound Power Source Type 4205 produces a very stable output, an ordinary loudspeaker was used for a second set of measurements. Therefore, the sound power level of each sound source with/without enclosure in octave bands from 125 Hz to 8 kHz was determined by averaging the sound pressure on mean-square basis over six positions in the room [3]. Moreover, the experiments were conducted for two types of interior absorbing material applied on the enclosure wall panels.

Since the sound power determination [3] requires the reverberation time of the reverberation room, it was measured using the method of interrupted noise as described in section 2.3.2.

The measurements were carried out in a reverberation room which implies diffuse sound field conditions. The humidity was about 60% at a temperature of 20°C.
### Table 1. Octave-band levels (L_{p1}…L_{p6}), mean octave band levels (\bar{L}_p) and band-power levels (L_w) for the unenclosed B&K sound source.

<table>
<thead>
<tr>
<th>Measurements</th>
<th>Frequency (Hz)</th>
<th>125</th>
<th>250</th>
<th>500</th>
<th>1000</th>
<th>2000</th>
<th>4000</th>
<th>8000</th>
<th>A weighting</th>
<th>Z weighting</th>
</tr>
</thead>
<tbody>
<tr>
<td>L_{p1} (dB)</td>
<td></td>
<td>82.1</td>
<td>79.2</td>
<td>80.7</td>
<td>85.2</td>
<td>82.1</td>
<td>78.9</td>
<td>76.2</td>
<td>88.7</td>
<td>90.1</td>
</tr>
<tr>
<td>L_{p2} (dB)</td>
<td></td>
<td>82.0</td>
<td>79.1</td>
<td>80.7</td>
<td>85.2</td>
<td>82.1</td>
<td>78.9</td>
<td>76.3</td>
<td>88.7</td>
<td>90.1</td>
</tr>
<tr>
<td>L_{p3} (dB)</td>
<td></td>
<td>81.4</td>
<td>79.5</td>
<td>80.6</td>
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<td>82.2</td>
<td>78.9</td>
<td>76.3</td>
<td>88.7</td>
<td>90.0</td>
</tr>
<tr>
<td>L_{p4} (dB)</td>
<td></td>
<td>81.8</td>
<td>79.5</td>
<td>80.7</td>
<td>85.2</td>
<td>82.2</td>
<td>78.9</td>
<td>76.3</td>
<td>88.7</td>
<td>90.1</td>
</tr>
<tr>
<td>L_{p5} (dB)</td>
<td></td>
<td>82.2</td>
<td>79.3</td>
<td>80.7</td>
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<td>88.8</td>
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<td>L_{p6} (dB)</td>
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<td>76.5</td>
<td>88.8</td>
<td>90.1</td>
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<tr>
<td>\bar{L}_p (dB)</td>
<td></td>
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<td>79.3</td>
<td>80.7</td>
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<td>82.2</td>
<td>78.9</td>
<td>76.4</td>
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</tr>
<tr>
<td>L_w (dB)</td>
<td></td>
<td>83.2</td>
<td>80.6</td>
<td>83.4</td>
<td>88.4</td>
<td>86.5</td>
<td>85.3</td>
<td>85.3</td>
<td>92.1</td>
<td>93.5</td>
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</table>

### Table 2. Octave-band levels (L_{p1}…L_{p6}), mean octave band levels (\bar{L}_p) and band-power levels (L_w) for the B&K sound source surrounded by an enclosure without interior porous absorbing material.

<table>
<thead>
<tr>
<th>Measurements</th>
<th>Frequency (Hz)</th>
<th>125</th>
<th>250</th>
<th>500</th>
<th>1000</th>
<th>2000</th>
<th>4000</th>
<th>8000</th>
<th>A weighting</th>
<th>Z weighting</th>
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<tbody>
<tr>
<td>L_{p1} (dB)</td>
<td></td>
<td>72.5</td>
<td>65.1</td>
<td>68.9</td>
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<td>64.8</td>
<td>60.4</td>
<td>75.2</td>
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<tr>
<td>L_{p2} (dB)</td>
<td></td>
<td>72.6</td>
<td>65.0</td>
<td>69.0</td>
<td>71.3</td>
<td>68.6</td>
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<td>60.4</td>
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<tr>
<td>L_{p3} (dB)</td>
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<td>68.9</td>
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<td>60.1</td>
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<td>78.7</td>
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<tr>
<td>L_{p4} (dB)</td>
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<td>65.2</td>
<td>68.8</td>
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<td>64.8</td>
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<td>75.2</td>
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</tr>
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<td>L_{p5} (dB)</td>
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<td>75.3</td>
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<td>L_{p6} (dB)</td>
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<td>65.0</td>
<td>60.3</td>
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<tr>
<td>\bar{L}_p (dB)</td>
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<td>L_w (dB)</td>
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<td>71.3</td>
<td>69.2</td>
<td>78.6</td>
<td>82.3</td>
</tr>
</tbody>
</table>
### Results of measurements

**Table 3.** Octave-band levels ($L_{p1}$…$L_{p6}$), mean octave band levels ($\bar{L}_p$) and band-power levels ($L_w$) for the B&K sound source surrounded by an enclosure with an interior porous absorbing material of thickness 28mm and density of 77 kg/m³.

<table>
<thead>
<tr>
<th>Measurements</th>
<th>Frequency (Hz)</th>
<th>125</th>
<th>250</th>
<th>500</th>
<th>1000</th>
<th>2000</th>
<th>4000</th>
<th>8000</th>
<th>A weighting</th>
<th>Z weighting</th>
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<tbody>
<tr>
<td>$L_{p1}$ (dB)</td>
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<td>65.8</td>
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<td>39.6</td>
<td>33.8</td>
<td>35.5</td>
<td>55.9</td>
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</tr>
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<td>$L_{p2}$ (dB)</td>
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<td>55.0</td>
<td>52.2</td>
<td>51.2</td>
<td>39.7</td>
<td>33.8</td>
<td>35.4</td>
<td>55.9</td>
<td>68.5</td>
</tr>
<tr>
<td>$L_{p3}$ (dB)</td>
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<td>66.0</td>
<td>55.5</td>
<td>52.2</td>
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<td>$L_{p4}$ (dB)</td>
<td></td>
<td>66.0</td>
<td>55.7</td>
<td>52.1</td>
<td>51.2</td>
<td>39.6</td>
<td>33.8</td>
<td>35.6</td>
<td>56.1</td>
<td>68.7</td>
</tr>
<tr>
<td>$L_{p5}$ (dB)</td>
<td></td>
<td>66.2</td>
<td>55.6</td>
<td>52.4</td>
<td>51.3</td>
<td>39.7</td>
<td>33.5</td>
<td>35.1</td>
<td>56.0</td>
<td>68.7</td>
</tr>
<tr>
<td>$L_{p6}$ (dB)</td>
<td></td>
<td>66.2</td>
<td>55.4</td>
<td>52.4</td>
<td>51.3</td>
<td>39.7</td>
<td>33.4</td>
<td>35.1</td>
<td>56.0</td>
<td>68.6</td>
</tr>
<tr>
<td>$\bar{L}_p$ (dB)</td>
<td></td>
<td>66.0</td>
<td>55.4</td>
<td>52.3</td>
<td>51.2</td>
<td>39.7</td>
<td>33.7</td>
<td>35.4</td>
<td>56.0</td>
<td>68.6</td>
</tr>
<tr>
<td>$L_w$ (dB)</td>
<td></td>
<td>67.3</td>
<td>56.7</td>
<td>54.9</td>
<td>54.4</td>
<td>44.0</td>
<td>40.1</td>
<td>44.3</td>
<td>59.3</td>
<td>71.9</td>
</tr>
</tbody>
</table>

**Table 4.** Octave-band levels ($L_{p1}$…$L_{p6}$), mean octave band levels ($\bar{L}_p$) and band-power levels ($L_w$) for the B&K sound source surrounded by an enclosure with an interior porous absorbing material of thickness 46mm and density of 28 kg/m³.

<table>
<thead>
<tr>
<th>Measurements</th>
<th>Frequency (Hz)</th>
<th>125</th>
<th>250</th>
<th>500</th>
<th>1000</th>
<th>2000</th>
<th>4000</th>
<th>8000</th>
<th>A weighting</th>
<th>Z weighting</th>
</tr>
</thead>
<tbody>
<tr>
<td>$L_{p1}$ (dB)</td>
<td></td>
<td>67.6</td>
<td>60.2</td>
<td>50</td>
<td>50.6</td>
<td>41.1</td>
<td>37.1</td>
<td>34.8</td>
<td>56.3</td>
<td>69.8</td>
</tr>
<tr>
<td>$L_{p2}$ (dB)</td>
<td></td>
<td>65.5</td>
<td>60.3</td>
<td>50</td>
<td>50.7</td>
<td>41.1</td>
<td>37.2</td>
<td>35.1</td>
<td>56.3</td>
<td>69.8</td>
</tr>
<tr>
<td>$L_{p3}$ (dB)</td>
<td></td>
<td>67.2</td>
<td>60.5</td>
<td>50</td>
<td>50.7</td>
<td>41.2</td>
<td>37.5</td>
<td>36.2</td>
<td>56.4</td>
<td>59.6</td>
</tr>
<tr>
<td>$L_{p4}$ (dB)</td>
<td></td>
<td>68</td>
<td>60.5</td>
<td>50.3</td>
<td>50.8</td>
<td>41.5</td>
<td>39.3</td>
<td>41.5</td>
<td>56.7</td>
<td>70.1</td>
</tr>
<tr>
<td>$L_{p5}$ (dB)</td>
<td></td>
<td>67.6</td>
<td>60.7</td>
<td>50.4</td>
<td>50.9</td>
<td>41.6</td>
<td>39.4</td>
<td>41.6</td>
<td>56.7</td>
<td>69.9</td>
</tr>
<tr>
<td>$L_{p6}$ (dB)</td>
<td></td>
<td>68</td>
<td>60.5</td>
<td>50.3</td>
<td>50.8</td>
<td>41.5</td>
<td>39.3</td>
<td>41.5</td>
<td>56.7</td>
<td>70.1</td>
</tr>
<tr>
<td>$\bar{L}_p$ (dB)</td>
<td></td>
<td>67.4</td>
<td>60.5</td>
<td>50.2</td>
<td>50.8</td>
<td>41.3</td>
<td>38.4</td>
<td>39.5</td>
<td>56.5</td>
<td>69.2</td>
</tr>
<tr>
<td>$L_w$ (dB)</td>
<td></td>
<td>68.7</td>
<td>61.7</td>
<td>52.8</td>
<td>53.9</td>
<td>45.7</td>
<td>44.8</td>
<td>48.4</td>
<td>59.9</td>
<td>72.6</td>
</tr>
</tbody>
</table>
### Chapter 3

Results of measurements

**Table 5.** Octave-band levels ($L_{p1}$...$L_{p6}$), mean octave band levels ($\overline{L}_p$) and band-power levels ($L_w$) for an unenclosed loudspeaker as a sound source.

<table>
<thead>
<tr>
<th>Measurements</th>
<th>Frequency (Hz)</th>
<th>125</th>
<th>250</th>
<th>500</th>
<th>1000</th>
<th>2000</th>
<th>4000</th>
<th>8000</th>
<th>A weighting</th>
<th>Z weighting</th>
</tr>
</thead>
<tbody>
<tr>
<td>$L_{p1}$ (dB)</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td>89.6</td>
<td>93.5</td>
</tr>
<tr>
<td>$L_{p2}$ (dB)</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td>89.8</td>
<td>93.7</td>
</tr>
<tr>
<td>$L_{p3}$ (dB)</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td>89.8</td>
<td>93.7</td>
</tr>
<tr>
<td>$L_{p4}$ (dB)</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td>89.5</td>
<td>93.6</td>
</tr>
<tr>
<td>$L_{p5}$ (dB)</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td>89.4</td>
<td>93.6</td>
</tr>
<tr>
<td>$L_{p6}$ (dB)</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td>89.3</td>
<td>93.5</td>
</tr>
<tr>
<td>$\overline{L}_p$ (dB)</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td>89.6</td>
<td>93.6</td>
</tr>
<tr>
<td>$L_w$ (dB)</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td>90.8</td>
<td>97.0</td>
</tr>
</tbody>
</table>

**Table 6.** Octave-band levels ($L_{p1}$...$L_{p6}$), mean octave band levels ($\overline{L}_p$) and band-power levels ($L_w$) having the loudspeaker surrounded by an enclosure without interior porous absorbing material.

<table>
<thead>
<tr>
<th>Measurements</th>
<th>Frequency (Hz)</th>
<th>125</th>
<th>250</th>
<th>500</th>
<th>1000</th>
<th>2000</th>
<th>4000</th>
<th>8000</th>
<th>A weighting</th>
<th>Z weighting</th>
</tr>
</thead>
<tbody>
<tr>
<td>$L_{p1}$ (dB)</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td>77.9</td>
<td>82.7</td>
</tr>
<tr>
<td>$L_{p2}$ (dB)</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td>77.9</td>
<td>82.6</td>
</tr>
<tr>
<td>$L_{p3}$ (dB)</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td>77.9</td>
<td>82.5</td>
</tr>
<tr>
<td>$L_{p4}$ (dB)</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
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<td>77.8</td>
<td>82.5</td>
</tr>
<tr>
<td>$L_{p5}$ (dB)</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td>78.0</td>
<td>82.8</td>
</tr>
<tr>
<td>$L_{p6}$ (dB)</td>
<td></td>
<td></td>
<td></td>
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<td></td>
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<td></td>
<td></td>
<td>77.6</td>
<td>83.0</td>
</tr>
<tr>
<td>$\overline{L}_p$ (dB)</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td>77.9</td>
<td>82.7</td>
</tr>
<tr>
<td>$L_w$ (dB)</td>
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<td></td>
<td></td>
<td></td>
<td></td>
<td>79.1</td>
<td>86.0</td>
</tr>
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</table>
### Results of measurements

<table>
<thead>
<tr>
<th>Measurements</th>
<th>Frequency (Hz)</th>
<th>125</th>
<th>250</th>
<th>500</th>
<th>1000</th>
<th>2000</th>
<th>4000</th>
<th>8000</th>
<th>A weighting</th>
<th>Z weighting</th>
</tr>
</thead>
<tbody>
<tr>
<td>$L_{p1}$ (dB)</td>
<td></td>
<td>73.0</td>
<td>70.1</td>
<td>55.0</td>
<td>52.3</td>
<td>42.1</td>
<td>38.0</td>
<td>38.5</td>
<td>63.3</td>
<td>78.7</td>
</tr>
<tr>
<td>$L_{p2}$ (dB)</td>
<td></td>
<td>72.9</td>
<td>70.3</td>
<td>54.9</td>
<td>52.4</td>
<td>42.1</td>
<td>38.0</td>
<td>38.5</td>
<td>63.4</td>
<td>78.6</td>
</tr>
<tr>
<td>$L_{p3}$ (dB)</td>
<td></td>
<td>72.9</td>
<td>70.4</td>
<td>54.8</td>
<td>52.3</td>
<td>42.1</td>
<td>38.0</td>
<td>38.5</td>
<td>63.4</td>
<td>78.6</td>
</tr>
<tr>
<td>$L_{p4}$ (dB)</td>
<td></td>
<td>72.8</td>
<td>70.1</td>
<td>54.9</td>
<td>52.3</td>
<td>42.1</td>
<td>38.0</td>
<td>38.5</td>
<td>63.4</td>
<td>78.6</td>
</tr>
<tr>
<td>$L_{p5}$ (dB)</td>
<td></td>
<td>73.0</td>
<td>69.8</td>
<td>55.0</td>
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<td>42.0</td>
<td>38.0</td>
<td>38.5</td>
<td>63.2</td>
<td>79.3</td>
</tr>
<tr>
<td>$L_{p6}$ (dB)</td>
<td></td>
<td>72.8</td>
<td>69.8</td>
<td>54.9</td>
<td>52.6</td>
<td>42.0</td>
<td>38.0</td>
<td>38.5</td>
<td>63.2</td>
<td>78.9</td>
</tr>
<tr>
<td>$\overline{L}_p$ (dB)</td>
<td></td>
<td>72.9</td>
<td>70.1</td>
<td>54.9</td>
<td>52.4</td>
<td>42.1</td>
<td>38.0</td>
<td>38.5</td>
<td>63.3</td>
<td>78.8</td>
</tr>
<tr>
<td>$L_w$ (dB)</td>
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<td>74.2</td>
<td>71.4</td>
<td>57.6</td>
<td>55.6</td>
<td>46.4</td>
<td>44.4</td>
<td>47.5</td>
<td>66.7</td>
<td>82.1</td>
</tr>
</tbody>
</table>

Table 7. Octave-band levels ($L_{p1}$...$L_{p6}$), mean octave band levels ($\overline{L}_p$) and band-power levels ($L_w$) having the loudspeaker surrounded by an enclosure with an interior porous absorbing material of thickness 28mm and density of 77 kg/m$^3$.

<table>
<thead>
<tr>
<th>Measurements</th>
<th>Frequency (Hz)</th>
<th>125</th>
<th>250</th>
<th>500</th>
<th>1000</th>
<th>2000</th>
<th>4000</th>
<th>8000</th>
<th>A weighting</th>
<th>Z weighting</th>
</tr>
</thead>
<tbody>
<tr>
<td>$L_{p1}$ (dB)</td>
<td></td>
<td>73.7</td>
<td>71.4</td>
<td>56.3</td>
<td>52.1</td>
<td>43.6</td>
<td>40.5</td>
<td>39.6</td>
<td>63.9</td>
<td>79</td>
</tr>
<tr>
<td>$L_{p2}$ (dB)</td>
<td></td>
<td>74.1</td>
<td>71.6</td>
<td>56.4</td>
<td>52.1</td>
<td>43.6</td>
<td>40.5</td>
<td>39.6</td>
<td>64</td>
<td>79</td>
</tr>
<tr>
<td>$L_{p3}$ (dB)</td>
<td></td>
<td>74.1</td>
<td>71.6</td>
<td>56.5</td>
<td>51.8</td>
<td>43.7</td>
<td>40.5</td>
<td>39.6</td>
<td>64</td>
<td>78.7</td>
</tr>
<tr>
<td>$L_{p4}$ (dB)</td>
<td></td>
<td>74.2</td>
<td>71.4</td>
<td>56.6</td>
<td>51.7</td>
<td>43.7</td>
<td>40.5</td>
<td>39.6</td>
<td>63.9</td>
<td>78.6</td>
</tr>
<tr>
<td>$L_{p5}$ (dB)</td>
<td></td>
<td>74.1</td>
<td>71.1</td>
<td>56.3</td>
<td>51.7</td>
<td>43.6</td>
<td>40.4</td>
<td>39.6</td>
<td>63.6</td>
<td>78.6</td>
</tr>
<tr>
<td>$L_{p6}$ (dB)</td>
<td></td>
<td>74.1</td>
<td>71.0</td>
<td>56.3</td>
<td>51.7</td>
<td>43.6</td>
<td>40.3</td>
<td>39.6</td>
<td>63.5</td>
<td>78.6</td>
</tr>
<tr>
<td>$\overline{L}_p$ (dB)</td>
<td></td>
<td>74.0</td>
<td>71.4</td>
<td>56.4</td>
<td>51.8</td>
<td>43.6</td>
<td>40.5</td>
<td>39.6</td>
<td>63.8</td>
<td>78.8</td>
</tr>
<tr>
<td>$L_w$ (dB)</td>
<td></td>
<td>75.3</td>
<td>72.6</td>
<td>59.1</td>
<td>55.0</td>
<td>48.0</td>
<td>46.9</td>
<td>48.6</td>
<td>67.2</td>
<td>82.1</td>
</tr>
</tbody>
</table>

Table 8. Octave-band levels ($L_{p1}$...$L_{p6}$), mean octave band levels ($\overline{L}_p$) and band-power levels ($L_w$) having the loudspeaker surrounded by an enclosure with an interior porous absorbing material of thickness 46mm and density of 28 kg/m$^3$. 

April 19, 2007
CHAPTER 3

Results of measurements

<table>
<thead>
<tr>
<th>Frequency (Hz)</th>
<th>125</th>
<th>250</th>
<th>500</th>
<th>1000</th>
<th>2000</th>
<th>4000</th>
<th>8000</th>
</tr>
</thead>
<tbody>
<tr>
<td>Reverberation time (s)</td>
<td>9</td>
<td>8.93</td>
<td>6.5</td>
<td>5.78</td>
<td>4.42</td>
<td>2.75</td>
<td>1.53</td>
</tr>
<tr>
<td>Volume of the test room (m³)</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td>240</td>
</tr>
</tbody>
</table>

Table 9. Reverberation time for the reverberation room using the interrupted noise method

The absorption coefficients of the two porous absorbing layers attached to the enclosure wall panels were determined by means of a standing wave tube [4] as stated in section 2.3.3. The insulation materials used had both different thickness and density: 28mm and 77kg/m³ first sample, and 46mm and 28kg/m³ second sample, respectively. These measurements gave rise to the following absorption coefficients:

<table>
<thead>
<tr>
<th>Frequency (Hz)</th>
<th>125</th>
<th>250</th>
<th>500</th>
<th>1000</th>
<th>2000</th>
<th>4000</th>
<th>8000</th>
</tr>
</thead>
<tbody>
<tr>
<td>$\alpha$ Sample 1 $\cdot$ 28mm, 77 kg/m³</td>
<td>0.054367</td>
<td>0.14063</td>
<td>0.31263</td>
<td>0.81426</td>
<td>0.97073</td>
<td>0.93567</td>
<td>0.98595</td>
</tr>
<tr>
<td>$\alpha$ Sample 2 $\cdot$ 46mm, 28 kg/m³</td>
<td>0.065366</td>
<td>0.16596</td>
<td>0.43774</td>
<td>0.90951</td>
<td>0.9872</td>
<td>0.95515</td>
<td>0.98595</td>
</tr>
</tbody>
</table>

Table 10. Absorption coefficients for two porous absorbing materials with different thickness and density: 28mm-77kg/m³ and 46mm-28 kg/m³, respectively.
CHAPTER 4

Analysis and discussions

In this chapter, we will analyze how well the measurements fit the theoretical models emphasised earlier in Chapter 2. Therefore, the experimental work is compared directly with each model simulation. The analysis is performed by means of Matlab software. Things of interest are the effect of the different absorption coefficients and the different sound sources from the measurements on the different insertion loss models on one side, and the relationship between these models and measurements (if there is any), on the other side.

The analysis starts by examining the absorption coefficients in octave bands of the two interior porous layers used to line the enclosure.

![Absorption coefficient for two different absorbing materials](image)

*Figure 8. The absorption coefficient for two porous absorbing layers of different thickness and density expressed as a function of frequency (in octave bands).*
The less dense layer shows a slightly better absorption over all frequencies, the thickness of the layers playing its role in this matter. However, the difference in absorption is a bit more pronounced in the mid range area; at high frequencies both materials prove to be very efficient insulators, the absorption coefficient being very close to its absolute value.

Data from each series of measurements was processed separately according to ISO3743-2 [3], and the insertion loss $D_e$ was determined in octave bands from 125 Hz to 8 kHz. The Mechel’s model and the low and high frequency approximations are simulated by means of Matlab software. Further, the insertion loss determined from the experimental data is plotted together with that computed for each model, and then conclusions are drawn based on these analyses. Figure 9 shows the average insertion loss when the box is lined with two different porous absorbing layers.

Figure 9. Average insertion loss of the enclosure according to measurements and 3 different approximations (Mechel’s model, high frequency and low frequency approximations) when the enclosure wall panels are lined with two different porous absorbing materials: either a layer of thickness 28mm and density 77 kg/m$^3$ (indicated on graph by circles) or a layer 46mm thick and density 28 kg/m$^3$ (indicated on graph by crosses).
The average insertion loss depicted in Figure 9 implies the use of both the B&K sound source and the loudspeaker.

The graph shows concerns about the low frequency approximation used due to the unexpected and uncommon high insertion loss; in-depth investigations into the low frequency modeling by means of equation (43) reveals that the interior air volume exhibits a resonance at 167Hz. This resonance is much lower than any other natural frequency of the system (for example, the lowest natural frequency of the box according to equation (40) is 214 Hz); thus applying this low frequency model to frequencies above its first resonance which occurs at 167Hz will yield to results that might be inconsistent with the theory.

However, since no frequency value is employed in any calculus in the low frequency modeling of the insertion loss, see equation (38), the high value of the prediction must originate in the assumptions made at these low frequencies when the sound source-box ensemble was modeled just as compliances. These approximations prove to be insufficient for an accurate description of the insertion loss the box poses at low frequencies. For a better low frequency modeling of the enclosure, its acoustical impedance should also exhibit a mass and maybe a resistance beside the presented compliances. Due to the poor consistency both with measurements and other models, the low frequency model is going to be discarded from analysis, though it is still displayed on the graphs.

Figure 8 and Figure 9 reveal a very important aspects regarding to the relationship between the insertion loss of the enclosure and absorption coefficient of the porous absorbing layer. The measurements show that the most absorbing mineral wool gives rise to the smallest insertion loss, fact contrary to the expectation that the higher the absorption the higher the insertion loss. Besides, at high frequencies, the increase in the absorption of the mineral wool seems to have no effect on the insertion loss of the box. More precisely, the determined insertion loss increases gradually up to 2kHz, then starts to decrease in spite of the absorption coefficient which continues to increase with frequency (and culminates
with a value close to unity at high frequencies). Therefore, the insertion loss determined from measurements appears to be independent of the absorption coefficient of the lining.

However, this seems not to be the case for the insertion loss predicted by the high frequency and Mechel’s models where the porous layer with the highest absorption exhibits the highest insertion loss. Although for the high frequency modeling this is not that evident, it is very obvious for Mechel’s model where the insertion loss when the most absorbing material is used is at least 8 dB higher then when the less absorbing material is utilized.

Contrary to the measurements, the insertion loss predicted by the high frequency and Mechel’s models increases as the absorption coefficient increases. If the insertion loss determined from measurements proved to be independent of the variation of the absorption coefficient of the enclosure lining (which is very obvious above 2kHz), these two models of the insertion loss seem to be somehow dependent on the increase in absorption.

In the first octave-band, at 125Hz, both models exhibit the same insertion loss regardless of the absorbing layer, but about 10dB higher then that determined from measurements. In other words, comparing to the measurements the models in question predict a false reduction of 50% in the perceived loudness. From 250Hz to 1kHz, the models show an ascendant slope of about 8dB per octave; if the insertion loss predicted by Mechel’s model keeps this slope up to 4kHz, the high frequency model seems to give the same insertion loss both at 2kHz and 4kHz. Nonetheless, both models exhibit a dramatic increase after 4kHz, namely about 44dB per octave for Mechel’s model and 30dB per octave when the high frequency modeling is implied. This sudden increase in insertion loss at high frequencies could mean that these models give to much weight to the material characteristics at these frequencies. Excepting the 125Hz and 8kHz octave-bands where the insertion loss predicted by models differs with at least 10dB and 30dB respectively, the high frequency model fits best to the measurements.
Figure 10. The effect of the different sound sources on the insertion loss of the enclosure. The lining of the enclosure is formed from a porous absorbing layer of 77kg/m³ density and 28mm thickness.

Figure 10 shows how the insertion loss of the enclosure behaves when different sound sources are used in the same setup configuration. Since both sources are driven with pink noise identical amplified and the measurements are done using the same enclosure and lining, the difference in the insertion losses determined from measurements reside only in the use of different sources. The explanation lays in the fact that the internal impedance of the loudspeaker has a lower value than that exhibited by the B&K sound source. In order to compensate for this impedance mismatch (that is getting reliable results regardless the sound source used) the source impedance should be included in estimations.
CHAPTER 5

Conclusion

The different insertion loss models that were the subject of this master project show the behaviour exhibited by a box which is intended to insulate a noise source from the surrounding environment. Therefore, these representations are supposed to provide engineers with efficient tools when designing covering boxes for machine installations.

The aim of this project was to find and examine models of the insertion loss by analysing the performance of an acoustical enclosure. Thus, measurements of real situations were performed in order to analyze the different models. Simulations of the models showed that the prediction of insertion loss from high frequency approximations would be better than that from Mechel’s model or low frequency approximations. However, the predicted insertion loss is less accurate than the measured one both at very low and very high frequencies.

The effects of the absorption of the interior lining have been investigated by using two different porous absorbing layers. The illustrations have revealed that the most absorbing layer produces the smallest insertion loss; on the other side, at high frequencies the insertion loss decreases slightly even though the absorption coefficient of the interior lining increases remarkably. Therefore, the conclusion drawn is that the measured insertion loss is not dependent on the absorption coefficient of the interior lining.

The results also showed variations in the insertion loss of the box when sound sources with different internal impedances were used. Therefore, the models should take into account the internal impedance of the source to compensate for this impedance mismatch and to yield trustworthy predictions.
However, it is necessary to emphasize that more data is needed for a statistical approach to work, and thus more measurements are required.

To prevent the limitation that the low frequency model has shown, more accurate approximations have to be done. This limitation is related to the low frequency modeling when the box and sound source were modeled just as compliances. If the enclosure wall panel impedance takes into consideration not just the spring but also the mass of the wall panel, a more accurate result is expected.

Having these results, some improvements could be made for further research. As a first suggestion, more measurements should be made using both different porous absorbing layers and sound sources; also more low frequencies should be covered by measurements (not just octave-bands measurements from 125Hz). Then, the impedance of the different sources has to be taken into account in the various models. In the low frequency modeling, the wall panel impedance should also exhibit a mass term and maybe a resistance beside the already given compliance. It would also be interesting to see more models of insertion loss in idea of comparing them with measurements.
Appendix

A Sound power determination

In the present project the determination of the sound power levels of a noise source is made by means of sound pressure measurements in a special reverberation room. The sound power determined by these measurements is used then for the evaluation of different models of insertion loss. The applied method complies with ISO 3743-2 [3]. Additional information to that already exposed in the Measurements section is displayed below.

A.1 Sound sources under test

The first set of measurements employ a Sound Power Source Type 4205 produced by B&K which consists of a generator and a sound source (HP 1001). This sound power source features a calibrated sound power output continuously variable from 40dB to 100dB (re 1pW), and a selectable wide band or octave-band noise output; the output can be either white or pink noise in the frequency range from 50Hz to 10kHz with a frequency response of ±1dB.

The second set of measurements is conducted using an ordinary loudspeaker that exhibits lower internal impedance and whose output is supposed to be less stable. Below, a comparison between the pink noise spectra produced by these two sound sources engaged in measurements can be found.
Figure 11. Pink noise spectra in octave-bands inside the reverberation room for two different sound sources: B&K sound source (black line) and an ordinary loudspeaker (green line)

The noise-reduction box whose insertion loss was determined has the dimensions $0.8 \times 0.4 \times 0.4$ m. The enclosure wall panel plates are made of medium density fiberboard (MDF) with a density of 800 kg/m$^3$, thickness of 9 mm, damping factor of $\eta = 1.3 \times 10^{-2}$, Poisson ratio $\mu = 0.1$, and Young’s modulus $E = 3.4 \times 10^9$ N/m$^2$.

The mineral wool used to line the box is produced by Rockwool. Two set of measurements were performed for each configuration due to the use of two different porous absorbing layers. The two different absorbing materials employed in measurements are of different thickness and density. If the first insulation material has a thickness of 28 mm and a density of 77 kg/m$^3$, the second is thicker, 46 mm, but less dense, that is 28 kg/m$^3$. 
The noise sources and the ensemble noise source-box were located according to ISO 3743-2 [3] into the reverberation room, that is no closer then 1 m to the nearest surface of the room (see Figure 6 for measurements setup). The pink noise generated by the B&K Real-Time Frequency Analyzer Type 2133 was sent to the generator of the Sound Power Source type 4205 which amplifies it and drives it to either the sound source HP1001 or the ordinary loudspeaker. During measurements, the output of the generator of the Sound Power Source type 4205 was chosen to be 84dB (re 1pW – the reading was possible due to the in-built display).

A.2 Acoustical environment

The measurements were conducted in a reverberation room with a volume of about 240m³.

The surfaces of the reverberation room are highly reflective and adequate diffusing. The nominal reverberation time is given in Table 9. The sound pressure level due to background noise was more then 10dB below band-pressure levels produced by the sources, and therefore no compensation was added to the measurements. The temperature registered was about 20ºC, whereas the humidity reached 60%.

A.3 Instrumentation

The measurement equipment was composed of a B&K Real-time Frequency Analyzer Type 2133, a B&K Microphone Power Supply Type 5395, a B&K Rotating Microphone Boom Type 3923, a B&K condenser microphone type 4152, a B&K Sound Level Calibrator Type 4231, a B&K Sound Power Source Type 4205 and an ordinary loudspeaker (for measurement setup see Figure 6).
A.4 Acoustical data

During measurements, no microphone position was closer to the surface of the room than 0.69m (it represents a quarter of the wavelength of the lowest octave-band in which measurements are made, that is 125Hz). The distance in between any microphone positions and the surface of the noise source (or the ensemble noise source-box) was 1.9m, whereas the distance between any two microphone positions was 1.37m (which is half the wave length of the lowest octave-band in which measurements are made, that is 125Hz).

The sound pressure levels of any noise source or the noise source-box ensemble were measured at a total of six microphone positions. Here, the microphone position depicts in fact the position of the rotating microphone boom on which the microphone is mounted (the microphone is moving traversing a path in the reverberation room at constant speed).

The mean octave-band level $\overline{L_p}$ is determined from the measured band-pressure levels of each octave band by means of the following formula:

$$\overline{L_p} = 10 \log \left[ \frac{1}{n} \left( 10^{0.1L_{p1}} + 10^{0.1L_{p2}} + 10^{0.1L_{p3}} + 10^{0.1L_{p4}} + 10^{0.1L_{p5}} + 10^{0.1L_{p6}} \right) \right] \ dB$$

where $L_{p1}$ is the octave-band level for the first measurement, $L_{p2}$ is for the second, and so on.

The measurement data is tabulated in Tables 1-8.
Appendix

B Absorption coefficient measurements

The absorption coefficients for the two sound absorbing layers used to line the interior of the box are determined according to ISO 10534-1 [4] by using the B&K Standing Wave Apparatus Type 4002 along with the B&K Real-Time Frequency Analyzer Type 2133 and B&K Sine Generator Type 1023.

The two different absorbing materials employed in measurements are produced by Rockwool and have different thickness and density. If the first insulation material has a thickness of 28mm and a density of 77kg/m³, the second is thicker, 46mm, but less dense, that is 28kg/m³. The flow resistivity determined according to their density [9] is $3.4 \times 10^4$ Pa⋅s/m² for the thinner sample and $0.45 \times 10^4$ Pa⋅s/m² for the thicker one (the higher the sample’s density, the higher its flow resistivity).

Since The Standing Wave Apparatus is supplied with two measuring tubes of different diameters (in function of the frequency range intended for), 4 samples of materials are needed in order to determine the absorption coefficients over the whole working frequency range of the tube. The material was carefully cut in circular samples and fitted in the sample holder without deforming its shape. Then the sample holder is clamped at one end of the tube and the loudspeaker is fed with pure tones using the sine generator; the measurements are completed by reading the first maximum and minimum sound pressure levels (starting from the sample) and the distance at which this first minimum occurs by means of the moving microphone probe attached to a car which slides along a ruler (see Figure 7 for measurements setup and the Measurements Chapter for more complementary details), and then the absorption coefficient is immediately determined [4].
C Characteristic values determination

For the determination of the characteristic values (equations (51), (52), and (53)) corresponding to the two different layers of Rockwool mineral wool employed in measurements, the following coefficients were used [2]: $\chi = 1.3$, $\sigma = 0.08$, $k = 1.4 + 0.15j$, $\alpha_1 = 1.1$, $\gamma_2, G_0 = 0.125$, $\gamma_0 = 1.12140$, $\gamma_1 = 1.49953$, $\gamma_2 = 0.468552$.

The flow resistivity $\Xi$ of the absorbing layers was determined according to their density [9], namely $3.4 \times 10^4 \text{ Pa} \cdot \text{s/m}^2$ for the sample with a higher density and $0.45 \times 10^4 \text{ Pa} \cdot \text{s/m}^2$ for the less dense one.


