ACOUSTICS 2012

The use of ray-tracing method for the prediction of the insertion loss of enclosures

N. Trompette and J. Chatillon

INRS, Rue du Morvan, CS 60027, 54519 Vandoeuvre Les Nancy, France
jacques.chatillon@inrs.fr

Among the various noise control techniques, enclosures are widely used. Predicting their insertion loss is therefore an important issue. A numerical model was developed which uses a conventional ray-tracing method that consists in determining the paths of acoustic rays and evaluating the energy loss along these paths. In our model, diffraction is not taken into account and the sources inside the enclosure are assumed to be omnidirectional. The model allows calculation of both the insertion loss and the sound pressure outside the enclosure. For validation purposes, a series of tests was run in a semi-anechoic chamber for different enclosures. The predictions of the insertion loss and the pressures outside the enclosure over the entire frequency range [250 Hz-4 k Hz] were satisfactory. However, the model tends to overestimate the pressure levels for the lower frequency band and underestimate them for the upper frequency bands. Ray tracing has proven to be a good tool for predicting the overall acoustic behavior of enclosures, but is inaccurate at low frequencies as it does not take into account the modal behavior of cavities and leaves. The discrepancies at high frequencies can be explained by leaks and the fact that diffraction was not taken into account.

1 Introduction

Enclosure is a noise reduction technique widely used in industry to reduce the radiation of noise sources. It consists of a casing made up of acoustic panels surrounding the source. This casing generally includes openings to enable the entry and exit of materials, glazed sections to monitor the process and doors or sliding panels to gain access to the machine. Enclosure manufacturers know the acoustic performance of the panels. Nevertheless, they run into difficulties in predicting the insertion loss of their enclosures on account of the presence of openings and the inhomogeneous nature of the panels due to glazed sections and doors. In addition, in France, final users check the reduction in sound pressure at the workplace rather than the insertion loss when assessing the efficiency of an enclosure. However, this reduction in sound pressure is impossible to evaluate analytically when the enclosure is complex. INRS has sought to overcome this problem by developing a tool to calculate the acoustic efficiency of enclosures.

Enclosures have already been the subject of research work aimed at developing performance prediction calculation tools. Analytical methods have been proposed ([1] & [2] or [3]) that are capable of accurately predicting the insertion loss of enclosures with simple shapes. Their limitations quickly become apparent as soon as the enclosure exhibits a more complex geometric shape ([2]). Several authors have proposed using the SEA method ([4] to [6]). All have achieved a satisfactory level of prediction. The recent work of Sgard et all [6] has shown that this method allows the influence of the position of the source, and thus the geometric shape, to be accurately taken into account. Moreover, the impact of an opening or the differences between panels is well modelled by SEA. However, as is the case with analytical methods, SEA does not predict what the sound pressure will be at the workplace.

A new numerical model has therefore been developed based on the conventional ray-tracing technique. This technique is widely used for workshops ([7] & [8]). It consists in determining the acoustic path of the ray and in assessing its energy loss along the entire length of this path. When the ray encounters a panel, it is split into a transmitted ray and a reflected ray. The transmitted ray is monitored and its energy assigned at the reception points it encounters outside the enclosure. It is a method that, a priori, is poorly adapted to small spaces as it does not take into account volume or panel modes. On the other hand, it allows the calculation of the radiated acoustic pressure at any given point, and thus at any workplace near the enclosure. The insertion loss is obtained conventionally by the difference between the source energy and the radiated energy.

The model was validated by means of a series of tests conducted in a semi-anechoic chamber on two types of enclosures. Several configurations were tested for the same type of enclosure (addition of an opening and a glazed panel, and changing the characteristics of a panel and the geometry).The insertion loss was measured by intensity measurement applying the scanning technique. The acoustic pressures were also recorded at several points evenly distributed around the enclosure to compare them to the calculated pressures.

The theoretical model is presented in the following paragraphs, and this is followed by a description of the experimental test cases and finally the results of the comparisons between the model and the experiments.

2 Model description

The ray-tracing technique consists in emitting rays from each source and monitoring them over the successive reflections and transmissions through the panels up to a receiver.

2.1 Modeling the source

If we consider an omnidirectional source S, the energy E radiated by the source is evenly distributed throughout the space over a predetermined number of rays N. An energy e = E/N is assigned to each ray whose direction is that of a unit vector U defined in spherical coordinates by the relationship:

$$\mathrm{U} = (\cos\theta\cos\phi, \sin\theta\cos\phi, \sin\phi) \quad (1)$$

where θ and ϕ designate the longitude and latitude of vector U respectively. To retain the omnidirectional characteristics of the source, θ is selected between 0 and 2π and cos(ϕ) between - 1 and + 1.

2.2 Effect of the presence of internal panels and reflection and transmission modeling

Ray tracing takes account of the real geometry of the enclosure. Each panel (either enclosure edges or internal partitions) is described by the equation of the plane and, if the panel is not infinite, by a set of geometric constraints to account for its dimension:

$$ax + by + cz = d$$
$$\text{and } x_{min} < x < x_{max},\ y_{min} < y < y_{max},\ z_{min} < z < z_{max} \quad (2)$$

It is therefore possible to describe the geometry of enclosures with complex shapes. Non-planar panels can be approached by breaking them down into elementary planar elements. In addition, if a panel does not have uniform absorption or transmission, it can be divided into elementary parts, each having given absorption and transmission coefficients.

Reflection and transmission by a panel is modeled as follows: when a ray hits the panel it is subjected to a specular reflection and its energy is reduced by a factor

$(1- \alpha)(1-1/\tau)$, where α is the absorption coefficient of the panel and τ its transparency.

The absorption and transmission coefficients are assumed to be independent of the angle of incidence. A second ray is generated to account for the transmission through the panel. Its energy is that of the incident ray reduced by a factor $1/\tau$, and its direction remains the same as that of the incident ray.

2.3 Sound energy receiver cells

To calculate the sound energy at different points outside the enclosure, the model uses receiver cells. These cells are spherical. They do not all have the same dimensions. For each ray, the model determines which cells are traversed and calculates the trajectory of the ray within the sphere. The energy assigned to the sphere is then calculated as follows:

$$E_{cell} = E_{ray} \times d / V_{cell} \quad (3)$$

where Vcell is the volume of the cell and d the distance traveled within the sphere by the ray. The validity of relationship (3) was demonstrated in [8], and is valid for a relatively large number of rays.

2.4 Limitation of the model

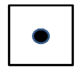
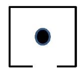
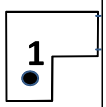
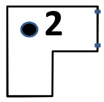
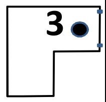
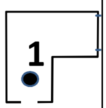
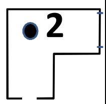
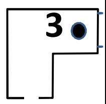
This is a very simple and straightforward model that is easy to compute. It has proven to be reliable for large rooms where fittings do not influence the acoustic field too significantly. However, simplification can affect the results of the calculation: in particular, scattering when rays traverse an opening are ignored and fittings present in the enclosure are not taken into account. In addition, the sources are assumed to be point sources and omnidirectional, although a complex source can be modeled as a distribution of point sources. Besides, this method requires the emission of a large number of rays in order to ensure convergence of the results; this implies long computation durations (up to 1 hour for our test cases) and requires large memory resources.

3 Comparison with tests

3.1 Experimental set-up

The method was tested on three enclosures and two types of panels. The first enclosure was square-shaped and made up of single 1.5 mm-thick steel panels, the inside of which had been absorption treated with 50 mm thick rockwool protected by a 33 % perforated panel. The second was L-shaped and composed of the same panels with the exception of one end of the L including a glass door. At the other end of the L, it was possible to open the panel (400x200 mm rectangular opening). The final enclosure tested was square-shaped. It was made up of double-leaf panels formed by two solid steel facings 0.6 and 1 mm thick respectively, separated by 30 mm of glasswool. The panels were absorption treated with 30 mm thick glasswool protected by a 27 % perforated panel. The configurations tested are summarised in the following table.

Table 1: List of test configurations [O = Opening ; 'N' no opening, 'Y' opening 400x200 mm]

Shape	Panel	Acous tic treatment	Source location	O
Square	Steel 1.5 mm	Rock wool 50mm		N
Square	Steel 1.5 mm	Rock wool 50mm		Y
Rectan gular	Steel 1mm – Glasswool 30mm – Steel .6mm	Glass wool 30mm		N
L shaped	Steel 1.5mm + Glass 4 mm	Rock wool 50mm		N
L shaped	Steel 1.5mm + Glass 4 mm	Rock wool 50mm		N
L shaped	Steel 1.5mm + Glass 4 mm	Rock wool 50mm		N
L shaped	Steel 1.5mm + Glass 4 mm	Rock wool 50mm		Y
L shaped	Steel 1.5mm + Glass 4 mm	Rock wool 50mm		Y
L shaped	Steel 1.5mm + Glass 4 mm	Rock wool 50mm		Y

3.2 Calculation data

Some data were required for the calculations: the absorption coefficient and the transparency of the panels. For the single-leaf steel panels, these data were taken from the literature as this is a very common type of panel. For the door, the manufacturer's data were used. For the double-leaf panels, the data were measured in a dedicated test rig in accordance with ISO 354 for the absorption coefficient and ISO 140-3 for the transparency.

3.3 Results

The comparisons are presented in Figures 1 and 2 (see the end of the paper) in the order of the detailed configurations given in Table I.

For the closed rectangular enclosures, the comparisons show that the method allows the prediction of insertion loss with a high degree of accuracy, whether composed of single-leaf or double-leaf panels. They also allow a very accurate prediction of the impact of an opening. On the enclosure composed of single panels, the differences were nevertheless high at 125 Hz. This result highlights one of the weaknesses of the method: it does not take into account the modal behaviour of the structures. However, this was particularly high at low frequencies, a point accentuated by the small size of the enclosure. Furthermore, for the enclosure made up of double-leaf panels, the differences at the 125 Hz octave were lower due to its being larger in size. While on the subject of this double-panel enclosure, a slight increase in the differences at high frequencies (2 and 4 kHz) and a levelling of the insertion loss measured should be pointed out. This shows that there were leaks despite all the precautions taken during assembly. These may have been introduced into the model, but were detected after the tests and thus were not characterised.

For the L-shaped enclosure, the comparisons were also very good. With the exception of the 125 Hz octave, the differences between the values calculated and measured were low (between 1 and 3 dB on average with a standard deviation of 1 dB). More particularly, the trends linked to the position of the source in relation to both the opening and the glazed panel were very well predicted.

The method therefore allows a very accurate calculation of the insertion loss and takes into account the presence of an opening and the position of the source. Its other interest was that it allowed the calculation of pressures at different points (e.g. at the workstation) outside the enclosure. The acoustic pressures calculated at several points distributed evenly around the enclosure were therefore compared to the pressures measured at these same points. Measurements were taken solely for the rectangular enclosures. The comparisons are presented in Table II (see the end of the paper).

The differences are higher than for the insertion loss comparisons, but nevertheless remain very acceptable for the configurations with closed enclosures, with the exception of the 125Hz octave for the closed enclosure made up of single panels. This point has, however, been explained earlier. On the other hand, when the enclosure includes an opening, the differences are high. The point-by-point analysis of the results shows that the calculation yields high differences between nearby points facing the opening whereas the measurements at the same points are practically equal. It is thus the radiation of the opening that causes the increase in the differences. The model renders it directive when in reality this is hardly the case. This can be put down to not taking into account opening edge diffraction. It goes without saying that the energy balance remains the same, which explains the good results obtained for insertion loss.

4 Conclusion

The model developed allows an accurate calculation of the insertion loss of enclosures and takes into account the influence of the position of the source and the presence of an opening. The lack of taking into account the modal behaviour of the panels and cavities nevertheless produces differences at low frequencies: the smaller the enclosure, the higher the differences. One of the interests of the method was the ability to calculate the radiated pressures. These pressures were accurately calculated for the closed enclosures but the differences increased sharply for the open enclosure. The modelling of the opening was incorrect, notably as regards its directivity. This problem was put down to the lack of taking into account diffraction at the edges of the opening. It is therefore necessary to introduce diffraction into the model to calculate pressures in the case of an enclosure with an opening.

References

[1] J.B. Moreland, “Low frequency noise reduction of acoustic enclosures”, Noise Control Engineering Journal, vol. 23. p. 140-149 (1984).

[2] N. Trompette, T. Loyau and G. Lovat, “Encoffrements de machine – Aide à la conception: règles de base et mise en oeuvre expérimentale”, Cahier de notes documentaires, INRS, Hygiène et sécurité du travail, vol. 182. p. 49-72 (2001).

[3] D.J. Oldham, S.N. Hillarby, « The acoustical performance of small close fitting enclosures, part 1: theoretical models”, Journal of Sound and Vibration, vol. 150(2), p. 261-281 (1991).

[4] Lyon RH., “Noise reduction of rectangular enclosures with one flexible wall”, J. Acoust. Soc. Am., vol. 35(11), p. 1791–7 (1963).

[5] Ming R, Pan J., “Insertion loss of an acoustic enclosure.” J. Acoust. Soc. Am., vol. 116(6), p. 3453–9 (2004).

[6] Sgard F., Nélisse H., Atalla N., Amedin C. K., Oddo R., “Prediction of the acoustical performance of enclosures using a hybrid statistical energy analysis: image source model”, J. Acoust. Soc. Am., 127(2), p. 784-795, 2010.

[7] A. Krokstad, S. Strom, S. Sorsdal, “Calculating the acoustical room response by the use of a ray tracing technique”, Journal of Sound and Vibration, vol. 8, pp. 118–125 (1968).

[8] A.M. Ondet, J.L. Barbry, “Modeling of sound propagation in fitted workshops using ray tracing”, J. Acoust. Soc. Am., vol. 85, pp. 787–796 (1989).

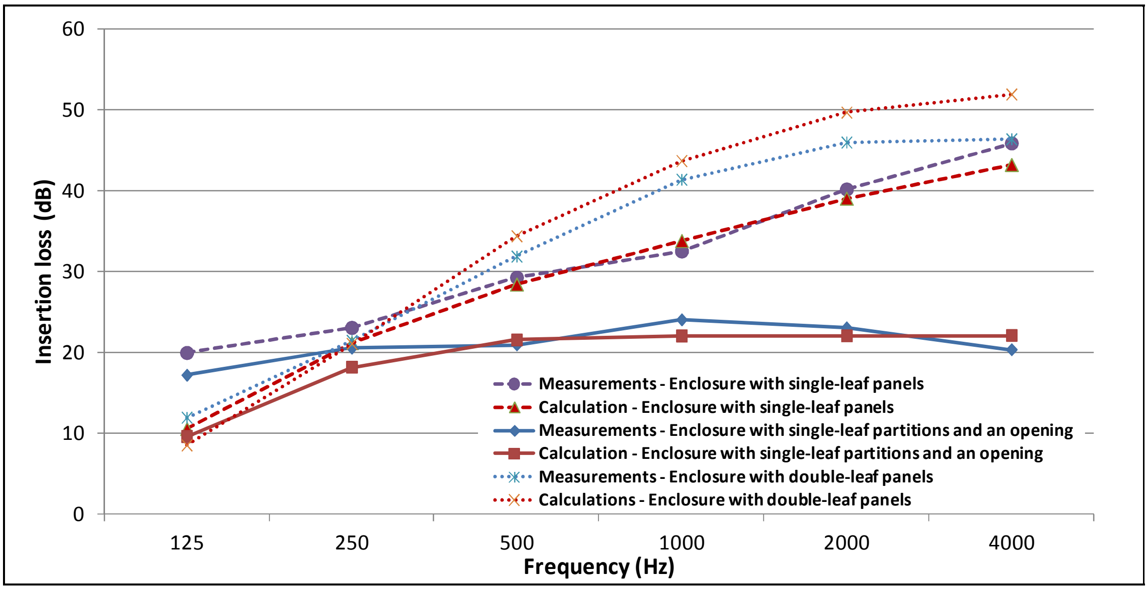


Figure 1: Insertion loss of rectangular enclosures – comparisons between measurements and calculations

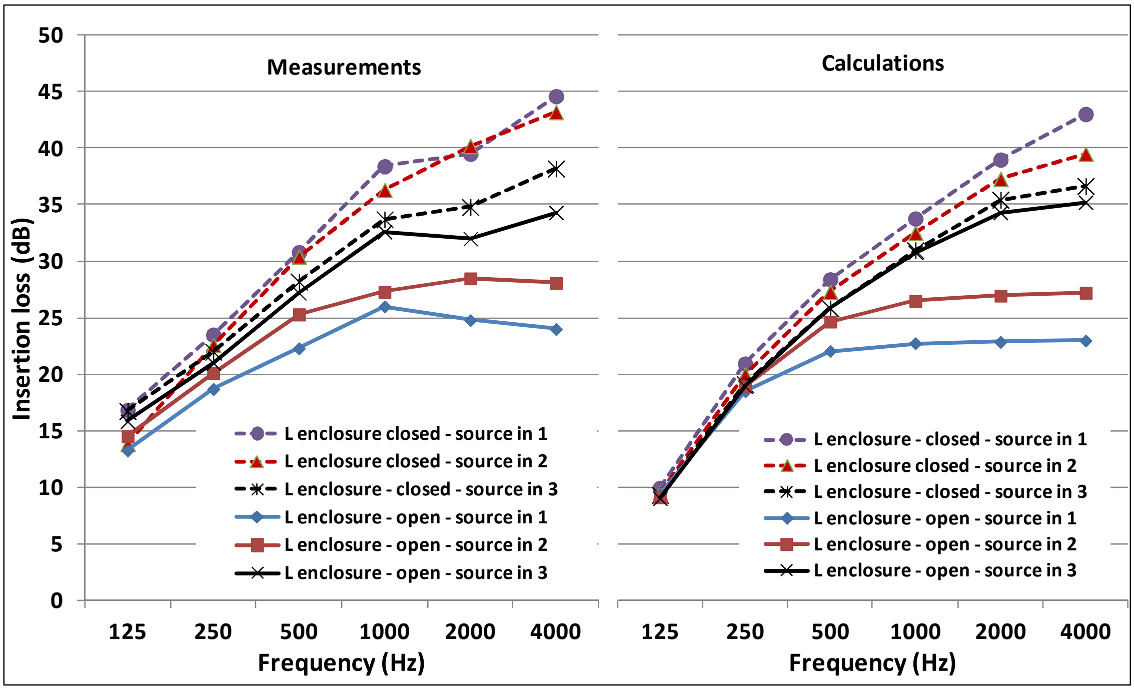


Figure 2: Insertion loss of the L-shaped single-leaf enclosure for various configurations – comparisons between measurements and calculations

Table II: difference between measured and calculated pressure – mean value and standard deviation

	Mean pressure difference over all the measurement points (dB)					
Type of enclosure	Single-leaf panels - closed - 8 measurement points		Single-leaf panels with an opening - 8 measurement points		Double-leaf panels - closed - 9 measurement points	
Octave	Mean difference	Standard deviation	Mean difference	Standard deviation	Mean difference	Standard deviation
125	12.9	4	7.1	4	3.4	2.8
250	4.4	1.2	5	2.7	1.9	1.7
500	1.3	1.5	7	5.2	1.6	2
1k	2	1.2	5.5	4.3	2.7	2.1
2k	2.1	1.8	4.1	4.4	4.8	1.2
4k	3.7	1.4	3.7	3.8	4.6	1.2
SPL dB(A)	5	1.3	3.2	2.9	1.1	1.8