Session B2 : Capotages/Encoffrements – Acoustic enclosures

Acoustic enclosures : principle and modelling

Encoffrements acoustiques : principe et modélisation

Jean-Claude Pascal,
Ecole Nationale Supérieure d'Ingénieurs,
du Mans (ENSIM),
Laboratoire d'Acoustique de l'Université,
du Maine (UMR 6 613),
72 000 Le Mans,
e-mail : jean-claude.pascal@univ-lemans.fr

Introduction

The enclosures and cabins are no doubt the tools to be used the most usually for reduction of the harmful effects due to the noise in industry. Enclosures can be found around the machines, turbo-alternator groups, encapsulating engines or in the integral part of manufacturing equipments by constituting its cover. The cabins will, on the other hand, produce a space of relative silence to protect the operators on a test platform. Paradoxically, the designers of enclosure installations have at their disposal little tools to guide them in their work.

Fahy (1 985) explained perfectly in some sentences the difficulty to find simple models to carry out calculations : *"Theoretical predictions of the performance of such enclosures have not been conspicuously successful to date, and designers still rely heavily on empirical data. The reasons are threefold :*

- the enclosure and source surfaces are strongly coupled by the intervening fluid, so that the radiation impedance of the source is affected by the dynamic behaviour of the enclosure,
- the geometries of sources are often such that the cavity wave fields are very complex in form and difficult to model deterministically ; and
- the dimensions of the cavities are not sufficiently large for statistical models of the cavity sound fields to be applied with confidence."

Although one did not stop improving calculation models and methods over the past 15 years, there are not many computation tools for industrials to be easily used. Indeed, the difficulties remain the same as before : sophisticated software requires so many and precise data that the engineer does not in general lay out. For these reasons, even if the large-sized conditions for the cavities are not gathered, the models based on the theory of the diffuse field are often employed. However there are rather simple models that can alternatively be implemented in function of the configurations in order to give results not too different from those in reality. These methods, gathered in the Capot software of CETIM, can be considered as a help in dimensioning enclosures by the engineer who can thus easily understand the influences of the various parameters.

Fig. 1 : The enclosures can be presented in very different configurations.

Fundamental basis

The effectiveness of enclosures is measured by the attenuation of the noise evaluated by the insertion loss which is defined as the difference between the acoustic power of the source without any enclosure and that with the enclosure. It is expressed by :

$$D = L_W - L_{W_T} = 10 \log \frac{W_T}{W} \quad \text{[dB]} \tag{1}$$

where :

L_W level of acoustic power of sources
L_{WT} level of acoustic power transmitted by the enclosure

In the practical guide titled "Handbook of Acoustical Enclosures and Barriers ", Miller and Montone (1 978) give simple rules which are still mainly employed in industry to perform daily calculations of enclosures

- the insertion loss provided by a **partial enclosure** with sound absorption inside will be proportional to the percentage of the machine which is enclosed

$$D \leq \min\left(R-3\,,\ 10\log\frac{100}{100-A\%} \right) \quad \text{[dB]} \qquad (2)$$

R is the transmission loss of the panels [in dB]
A% is the percent of enclosure area [in %]

- the insertion loss of **large enclosures** within "a fairly diffuse reverberant sound field exists within the space between a noise source and an enclosure when the noise source is small compared to the total enclosure volume" (less than 1/3) is

$$D = R + 10\log\alpha \qquad \text{[dB]} \qquad (3)$$

α is the average absorption coefficient inside the enclosure

- the insertion loss of **closely-fitting enclosures** will become negative at some resonance frequency which depends of panel impedance and of air space between the source surface and the panel.

The first rule which is associated with the partial enclosure is very inaccurate but it shows the role of the openings. When the openings become more important the enclosure approaches a simply barrier. The small openings and the leaks which always exist in enclosures will result the insertion losses in being limited.

The second rule also contains great ambiguity : to make the insertion losses to be equivalent to the transmission loss of the panels it is necessary that the average absorption coefficient of the walls should be equal to 1 ! And in the case of a very low coefficient, the insertion losses become negative, which in turn violates the principle of the energy conservation !
To remove this ambiguity it is advisable to consider the diagram of Figure 2 which represents the distribution of energy for a wall composed of a panel and a absorbing material layer.

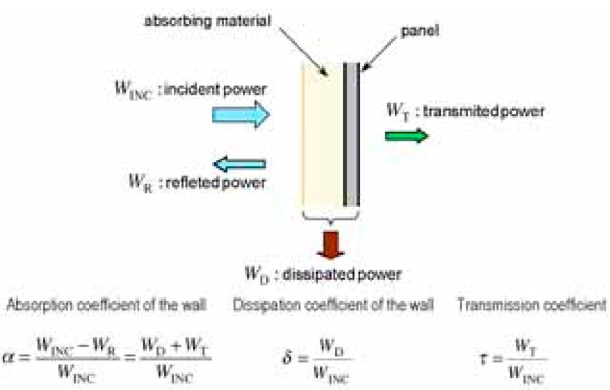

Fig. 2 : Coefficients of the power path through a wall composed of elastic panel with absorbing material

The relationship among the different coefficients indicated in Figure 2 expresses the conservation of energy :

$$\alpha = \delta + \tau \qquad (4)$$

Without absorbing material, the absorption coefficient of the wall corresponds to the sum of the transmission coefficients and dissipation of the panel. The dissipation of the absorbing material will be increased. Substituting Eq. (4) in Eq. (3) yields,

$$D = 10\log\frac{\delta+\tau}{\tau} \qquad (5)$$

Eq. (5) is easier to be understood if considering :

$$R = 10\log(1/\tau) \quad .$$

It is exactly the relation suggested by Pierce (p.281, 1981) for the large enclosures, which concludes : "An implication of Eq. (5) is that no sound power reduction is achieved unless there is some absorption". It is thus of primary importance to estimate well the dissipation of the whole enclosure : intern absorption as well as that of the walls and the dissipation due to the whole porous material and panel.

Although the dissipative effects are determining in the behaviours of the enclosures, the third point (c) shows that they cannot explain all. The reactive effects associated with the interference phenomena with enclosures will produce the modifications on the radiation impedance of the sources and radiated power of enclosures.

The walls of the enclosure constitute a barrier with the airborne noise radiated by the source, but it should not be forgotten that it is the whole transmission path which must be controlled in order to optimise the performances :

the transfer through the walls : the acoustic attenuation and dissipation are the principal characteristics.

the sealing : it must be ensured at the level of the assemblies and the functional openings by the joints, silencers, and baffles, etc.

structure-borne sound : the mounts of the enclosure on the source, the interdependent elements of machine of the enclosure must be isolated on the vibratory plan. It is essential to dissociate the enclosure from all excitation sources, whether it is the machine or the ground.

Modelling of enclosures and their elements

Several more or less sophisticated models were proposed in the literatures. It should be noted that these models, which rely on restrictive assumptions, are relevant according to the type of enclosures. For a given enclosure, these models can be adapted to particular frequency bands.

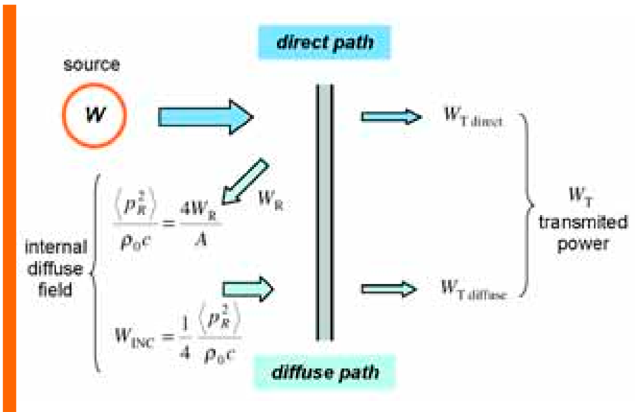

Fig. 3 : Simple energetic model with diffuse field

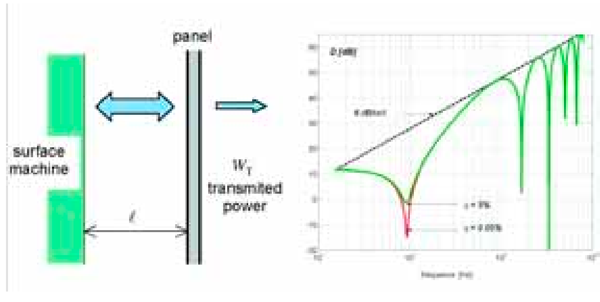

Fig. 4 : Jackson model and its predicted insertion loss for a steel panel of 2 mm at 0,1 m from the vibrating surface machine (m = 15.6 kg/m²)

Simple dissipative model at high frequencies

If enclosure and wall panels show a large number of modes in a frequency band, a statistical method can be used for predicting the insertion loss. High sound transmission loss allows the sound waves to be contained inside and the sound absorbing treatment converts the trapped acoustical energy into head. A high percentage of energy is dissipated in this way at high frequencies. A model of diffuse sound field is used as indicate in Figure 3 (see for example Bies & Hansen, 1996, or Vér, 1992).

We will distinguish the direct way where a part of energy is dissipated and transmitted through the wall. The reflection part will make the diffuse field be increased. represents the total dissipation inside the enclosure caused by wall absorption, transmission through walls, or by other absorbing surfaces. Vér (1992) adds also the sound absorption in air and the dissipation in the walls through viscous damping effects $\delta_W S_W$, with

$$\delta_W = \frac{4\pi\sqrt{12}\,\rho_0 c^3 \sigma}{h\,\rho_S c_L \omega^2}\left(\frac{\rho_S \omega\eta}{2\rho_0 c\sigma + \rho_S\omega\eta}\right), \qquad (6)$$

an expression given by the SEA for the interaction between diffuse sound field and a panel (ρ_S masse per unit area, C_L longitudinal wave speed, h thickness, η loss factor) of radiation efficiency σ.The last term may be of primary importance for an enclosure without sound absorbing treatment. The quarter of mean square pressure gives the sound incident intensity on the panel, according to the diffuse field theory and leads to the transmitted power by the diffuse path.

A complete analysis may include the transmission opening (holes, slits, silencers) for a good prediction.

Reactive model for close fitting enclosures

This model was proposed by Jackson (1962, 1966) to predict the influence of interactions between the source and the enclosure walls, in the case where the volume of the enclosed machine is comparable with the volume of the enclosure. It can be extended to consider the effects of acoustical absorbing material (Byrne et al, 1988). Surface machine and enclosure panel are modelled as two infinite parallel planes (see Figure 4).

In the one-dimensional model, the machine surface vibrates like a piston and is not affected by the presence of the enclosure. In the example shown in Figure 4, the mechanical resonance frequency f_0 of the panel due to the stiffness of the mount is 80 Hz. The stiffness $\rho_0 c^2/l$ added by the cavity increases the resonance frequency f_1 to 92 Hz and the insertion loss is minimum at this frequency :

$$f_1 = f_0 + \frac{2\pi}{\ell}\frac{\rho_0 c^2}{\rho_S} \qquad (7)$$

(ρ_S is mass per unit area of the panel). If the mechanical damping is rather low, more power can be radiated from the enclosure at this resonance frequency than from the unenclosed source because the standing waves inside change the impedance observed by the machine surface. It is of importance to choose f_1 carefully. The damping is the more important factor in controlling D at this frequency. The reactive behaviour of the coupling has also the effect to adding 6 dB to the mass law for the maximum values of the insertion loss at some others frequencies (discontinues black line in Fig.4).

Behaviour of enclosure at high frequencies

Enclosure may be considered as being small for low frequency sound when the largest dimension is less than one-quarter wavelength of sound. For a small sealed acoustical enclosure, Lyon (1963) has given a formulation where the insertion loss depends essentially on the compliance of walls and of the inside air volume V:

$$D = 20\log\left\{1+\frac{V/\rho_0 c^2}{\sum_i^n C_i}\right\} \overset{\text{cubic}}{\cong} 20\log\left(1+41\left(\frac{h}{a}\right)^3 \frac{E}{\rho_0 c^2}\right) \qquad (8)$$

where $C_i = S_i^3\, g(a_i/b_i, cL_i)/B_i$ is the compliance of the i-th wall panel, with S_i the surface of the panel, $g(a_i/b_i, CL_i)$ a function that depends on dimension ratio a_i/b_i and edges conditions and B_i the bending stiffness of the panel. The right hand side of the above equation gives the insertion loss in a special case of a cubical enclosure with clamped wall of edge length *a*, thickness *h* and Young's modulus *E*. The results depend on a large part of the mechanical characteristics of panel material: for a same weight, an enclosure built in aluminium has *D* largely superior to an enclosure built in steel. To increase the insertion loss of a small sealed enclosure, the form stiff plays a role as for a spherical like body of compressor housing.

In general, enclosures are not hermetically sealed so that their owns compliances due to the leakage are added to those of the walls

$$\sum_i C_i \quad \rightarrow \quad 1/j\omega Z_L + \sum_i C_i \qquad (9)$$

where Z_L is the acoustical impedance of leakage (Mechel,1986). When one approaches the first acoustic mode of interior volume, the precedent method is not usable any more. In the intermediate frequency range, the enclosure walls and the cavity exhibit resonance that are not overlapped each other. So that statistical methods shown below are not yet applicable. Because the insertion loss in the intermediate frequency range fluctuates widely with frequency, it is very difficult to make accurate analytical predictions. The reduction of insertion loss is due to the strongly coupled structural modes. Only the numerical methods such as the finite element and the boundary element, can be implemented to solve the problem of coupling between structure dynamics and acoustic resonance (Nefske et al., 1980, Agahi et al., 199). If only the frequency band of the first modes is interesting, the mesh will not have to be very dense and the work for calculation will be reduced enough to adapt to the industrial problems considered.

Modelling enclosure walls

To improve the precision of the results, it is very important to have a good model for the whole enclosure consisting of the panel and absorbing material. The transmission loss of the walls was largely studied (Josse et al.,1964, Nilson, 1972, Galiardini, 1991). Certain formulations adapted for the building (critical frequency at low frequency range, consideration of the coupling of the rooms) are not for the panels of enclosure (critical frequency at high frequency range) for which a lot of data needed for modelling are unknown. Sometimes reduction of panel dimensions is parameters that are taken into account in some models (Sewell, 1970, for example), but generally the influence of the first resonant modes is not at all taken into account. The knowledge of the damping of the panels, which is important to know the dissipation brought to the reverberated field, is also to estimate the coefficient of transmission around and above the critical frequency. However this parameter is in general known in a very approximate way.

This uncertainty on the evaluation of dissipation is also found in the data of the fibrous material manufacturers which coefficients measured in a reverberant room are higher than 1. As it was emphasized at the beginning, it is the wall (panel + absorbing material) which must be considered as a whole in order to satisfy imperatively the relation of conservation (4).
The porous material models of equivalent fluid type (Delany and Bazley or Johnson, Allard, 1993) must be considered as being mounted on a flexible panel (Dowell et al., 1981).

To obtain a more precise estimation, it is necessary to consider the poroelastic behaviours of absorbing material, in agreement with the theory of Biot. Numerical models were developed in order to take into account these effects (Attala et al., 1998) but some simpler models were proposed where the influence of the elastic behaviour of material is considered as an equivalent plate while defining the parameters of a equivalent fluid model (Dauchez, 1999). The composite walls (multi-layer panels) are calculated by using a method of impedance matrix (Brouard et al., 1995). Even if the tools of modelling currently exist, the problem remains the knowledge of the parameters which must have entered. It is thus necessary at the same time to develop the experimental tools which are accessible to the industrial users.

Fig. 5: The enclosures can consist of a number of removable panels, access doors, opening or inspection doors which should be treated

Other transfer paths

Other transmission paths, such that those of the transmission though the walls can considerably reduce the performances of the enclosures. This is the case of flanking transmission of structure-borne sound via the supporting structure or connections between the source and the enclosure walls. It is necessary to separate or isolate the machine foundation from the enclosure walls. Resilient mounts on a heavy foundation and resilient connections are employed. The openings must also be taken into account with attention. The different passages (cable, transmission organs, etc…) must be sealed with the greatest care. In particular, the openings with enclosures for the evacuation of the air must be treated by silencers. The enclosures are generally modular and include many removable panels (see Figure 5) for technical requirements (maintenance, intervention).

The leaks for this type of assembly shown in Figure 5 are not avoidable and should be sealed by joints. A model of an enclosure should thus take them into account for the leaks which inevitably reduce the insertion losses in the middle frequency range. The characteristics of the leaks (thickness of the wall, width of the slit) lead to frequential singularities. This behaviour, as well as their directivity, has been studied (Thomé, 1993). The most complete model has been developed by Mechel (1986). The advantage of this model is that it treats the joint as of a dynamic mass (elastomers) or as an equivalent fluid in the duct of the slit (for materials of sealing).

Figure 6 shows the transmission loss of a steel plate of 2 mm thickness (surface area 0.8 m) with a leak of 2.5 mm of width and 50 mm of thickness.

The simulation results are computed by the use of the transmission loss of the steel plate obtained from measurements and that of the leak using Mechel. The leaks are sealed by dynamic equivalent mass in the model having the half of the joint mass. The model is compared with the measurements carried out in a reverberant room using intensimetry technique (Corlay et al., 2000).

Discussion and conclusion

Due to practical considerations and limitations of space, close fitting enclosures are now widely used. The vibrating source radiates into a small volume and the sound is not diffused inside the enclosure in a large scale of frequency.

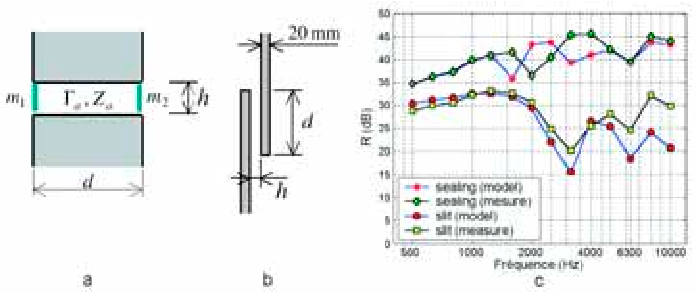

Fig. 6 : Transmission loss of a steel plate with a leak of thickness 2.5 mm. (a) Model parameters, (b) Experimental configuration, (c) Comparisons of measurement and simulation results for the leak and with a sealing of 5 g/m (a dynamic mass of 2.5 g/m² being taken in to account). (Measurements made by CETIM)

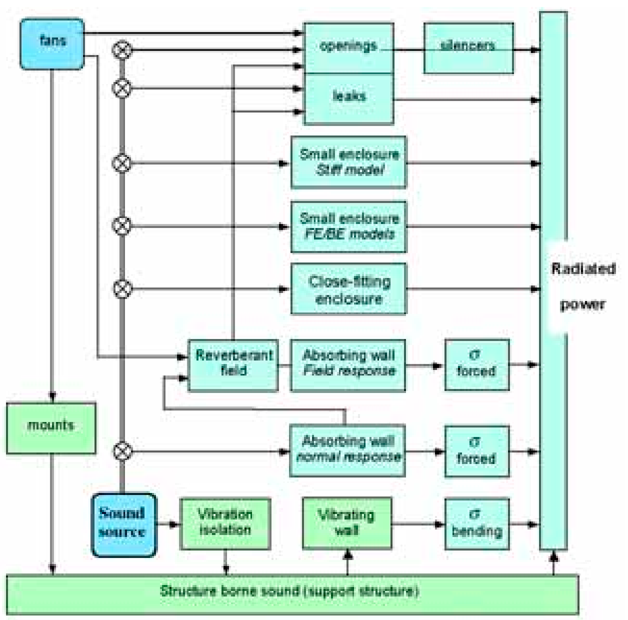

Fig. 7 : Diagram of model arrangement using in software for enclosure design.

At low frequencies, there is close coupling between the source and the enclosure walls, and the stiffness of the air gap and the panels become important parameters. Near the first frequency of the acoustic volume, the major portion of acoustic energy is transmitted by the fundamental vibrations modes of the panels. In this range, numerical methods can predict the behaviour of fluid-structural interaction. In high frequency range, where both the enclosure panels and the interior air volume exhibit a very large number of acoustical resonance, statistical methods can be used.

For example, the SEA may particularly be useful to predict the distribution of energy among the panels. This outstanding behaviour has been a subject of study to improve the theoretical prediction of acoustic performance of the enclosures.

The various models, which were previously described, have generally been proved reliable in the configuration and the band frequency in respecting the prescripted assumptions. However, only one cannot predict the behaviour of enclosures in the whole useful frequency ranges.

An effective way to use these models is to combine them according to the diagram of Figure 10. In terms of numerous parameters, for example, the distance of wall-source, the ratios of wavelength-dimension, it is possible to distribute energy according to models, thus allowing one to use them in the field of validity. It is this principle that is taken in to consideration in the CETIM-Capot software.

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