inter·noise 83

THE PERFORMANCE OF ACOUSTIC ENCLOSURES

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INTRODUCTION

Acoustic enclosures are often used to reduce the noise emanating from machinery. Economic considerations usually dictate that the enclosure be as small as possible. It has been found that it is impossible to predict close fitting enclosure performance using techniques developed in the field of architectural acoustics.

Various attempts have been made to develop enclosure prediction techniques [1] [2], but these have not been very successful [3]. In this paper we discuss the shortcomings of the existing models and suggest an alternative approach.

SPECIFICATION OF ENCLOSURE PERFORMANCE

Both Jackson and Junger modelled the source enclosure problem in terms of parallel plates separated by an air gap. Jackson assumed infinite plates and derived an expression for the vibration level of the enclosure plate as a function of the vibration level of the source plate and enclosure plate parameters. Enclosure performance was then specified in terms of the relative vibration level of the plates. Real source and enclosure panels are not infinite and the sound power radiated by either will not just be determined by the rms vibration level.

Junger considered both the source plate and enclosure plate to be of finite area. The former he assumed to vibrate as a uniform piston and the latter to vibrate as a simply supported plate excited by a uniform pressure field. The enclosure performance was specified in terms of the reduction in sound pressure level at a reference point.

Junger's model still has limitations. Modelling the source as a uniformly vibrating piston gives rise to problems. Firstly a real source panel will not vibrate in this manner but will exhibit forced vibrations in a number of modes. As a result the "unenclosed" sound pressure level at a given point and the exciting pressure on the enclosure panels will both differ from those due to a simple piston having the same rms vibration level. Specification of enclosure performance as a reduction in the calculated SPL is liable to result in misleading data since pistons are notoriously directional. Shen and Oldham have shown that small panels do not exhibit marked directivity effects until the critical frequency is exceeded [4]. The Shen and Oldham work further suggests that the directivity of small panels might be very easily characterised and hence a better method of specifying enclosure performance would be in terms of the overall reduction in sound power level with directivity information obtained from charts.

THEORETICAL APPROACH

The differential equation governing the small-amplitude transverse vibration of a rectangular isotropic plate of uniform thickness is:

$$D\nabla^4 w(x,y,t) + \rho h \partial^2 w(x,y,t)/\partial t^2 = f(x,y,t) \qquad (1)$$

where $w(x,y,t)$ is the displacement function of the plate, $f(x,y,t)$ is the pressure difference function across the plate, $\nabla^4 = (\partial^2/\partial x^2+\partial^2/\partial y^2)^2$, x and y being co-ordinates in the plane of the plate, $0 \leqslant x \leqslant a$ and $0 \leqslant y \leqslant b$, a and b being the lengths of the edges of the plate $(a \geqslant b)$, t is time, ρ is the plate material density, h is plate thickness, d is the flexural rigidity of the plate, $D = Eh^3/12(1-\nu^2)$, E is the plate material Young's modulus and ν is Poissons's ratio.

A method of solving this equation for either simply supported or clamped plates has been presented by Shen and Oldham [4]. The radiated sound power level can be calculated using the Rayleigh far field approximation. The solution to equation 1 and the calculation of the radiated sound power level requires considerable computing power.

In order to calculate the reduction in PWL it is necessary to be able to relate the exciting pressure function to the source characteristics. Two source types can be considered - extended sources and small sources.

Extended source

Figure 1 shows a representation of a simple enclosure with one flexible wall parallel to the extended source which is assumed to be a panel vibrating in the 3:3 mode. For low frequencies there will be a tendency for the pressure inside the enclosure to equalise and thus the assumption of uniform pressure excitation will be valid. A similar argument holds for higher order modes. The work of Junger [2] suggests that, for uniform excitation at other than the natural frequencies of a panel, the 1:1 mode is excited to a greater extent than other modes. If this is the case then the calculation of the radiated sound power becomes much easier.

Figure 2 shows the results of measurements made using apparatus simulating the situation shown in Figure 1. Nine loudspeakers were arranged in a 3 by 3 array, each speaker was driven by its own power amplifier and its phase could be reversed independently of the others. By a suitable combination of phases it was possible to simulate vibration in the 1:1, 1:3 or 3:3 modes. Measurements were made of the reduction in sound power level for one third octave random noise resulting from an aluminium panel measuring 0.7m x 0.7m x 3.3mm spaced 370mm from the source for the source operating in different modes.

For very low frequencies the curves are similar but diverge above 160 Hz. Assuming the enclosure panel to be clamped the natural frequencies corresponding to vibration in the 1:1, 1:3 and 3:3 modes are 58, 206 and 295 Hz respectively. The effect of the coupling between source panel modes and enclosure panel modes can be seen clearly at these frequencies.

Small source

The sound pressure field inside an enclosure containing a small source will be determined by the pattern of modes excited. A small enclosure differs from a room in that the fundamental frequency will be relatively high. At frequencies well below the fundamental the pressure field will be uniform. At frequencies above the fundamental there will be resonances and the possibility of matching between enclosure modes and panel modes.

Figure 3 shows the results of measurements made on two 0.5m x 0.5m x 0.5m enclosures, one made of 1.25mm thick aluminium and one made of 1.6mm thick steel, containing a loudspeaker. The reduction in enclosure performance due to a build up of sound pressure at a frequency of 340 Hz corresponding to the 1:0:0, 0:1:0 and 0:0:1 modes can be clearly seen. At higher frequencies, where more modes are excited, and the situation is more room like, the performance of the enclosure can be predicted from knowledge of the random incidence transmission loss.

CONCLUSIONS

The preliminary work described in this paper suggests that enclosure performance is better described in terms of sound power level reduction rather than the sound pressure level reduction at a point. Attempts to derive methods of predicting enclosure performance at low frequencies may be affected by strong coupling between corresponding source modes and enclosure panel modes. Attempts to derive prediction methods for small sources require knowledge of the behaviour of resonant modes in the enclosure.

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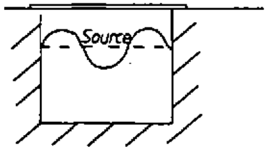

Figure 1 Extended Source

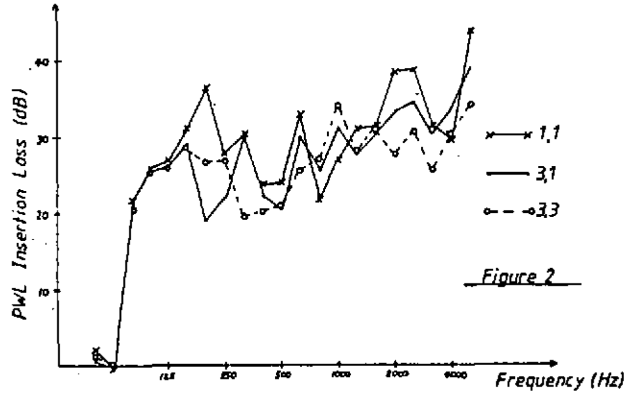

Figure 2 Extended Source Insertion Loss

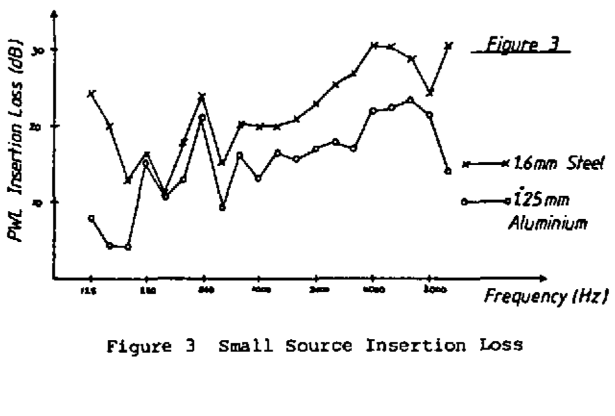

Figure 3 Small Source Insertion Loss