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THE EFFECTS OF DAMPING TREAMENTS ON THE RADIATED SOUND FROM A CLOSED CYLINDRICAL SHELL

C.J. Wu (1), Y.Y. Ma (2) and R.G. White (1)

(1) Institute of Sound and Vibration Research, University of Southampton, Southampton SO9 5NH, England
(2) Wuhan Ship Design and Development Institute, P.O. Box 64291, 430064, P.R. China

ABSTRACT

The effects of a damping treatment on the sound radiated from closed, cylindrical shells have been studied theoretically and experimentally. An expression relating the insertion loss of sound pressure level ΔSPL to the composite loss factor of the structure due to additive damping materials is presented, which is of use for predicting the performance of treatments to practical structures. The formula is valuable for practical engineering estimation. Data from experiments on a model cylinder carried out by a single impact method show good agreement with predicted results.

1. INTRODUCTION

Closed, cylindrical shells are a basic machinery structure, or component of many machines found in industry. This kind of structure generally has a high radiation efficiency, even at low frequency. Therefore it can create high noise levels with wide frequency range, e.g. the A weighted sound pressure level from a cylinder ball grinder can be as high as 115 ~ 123.5 dB(A). Hence it is of considerable interest to know to what extent additive damping will affect the radiated sound from this type of structure.

However, in existing literature, the use of damping treatments is generally linked with structural vibration control rather than noise control. To seek simplified expressions for engineers to quantitatively estimate the insertion loss of sound pressure level ΔSPL created by use of a damping treatment is a worthwhile objective. A simple formula is presented here, which is particularly useful in practical engineering. The following assumptions are made for the closed cylinders considered here:

(a) The cylinders considered here are of similar dimensions to those found in industrial machinery, and the velocity responses of the ends, i.e. circular plates, are neglected compared with velocities on the cylindrical surface;
(b) Experiments were carried out to excite test cylinders via use of a falling ball inside the shell. It was assumed that the excitation was the same in all tests and was not influenced by the addition of a damping treatment.

Using assumption (a), an expression is developed for estimation of the insertion loss created by application of a damping treatment. Following (b), experimental data obtained from a large scale model structure are shown to be in good agreement with predicted values.

2. THE ENERGY BALANCE EQUATION

A large-scale experimental model of a closed, cylindrical shell structure is shown in Figure 1, which shows how a ball impacted perpendicularly on the inside surface of the shell. For a single impact and energy input E_{in} one has the equation which describes the energy balance between radiated sound energy E_{rad}, dissipated energy E_{struc} and the energy loss to the ground via the supports E_{gr}, which is

$$E_{in} = E_{rad} + E_{struc} + E_{gr} \tag{1}$$

where the dissipated energy E_{struc} is converted into heat by structural damping. If the cylinder is perfectly simply supported, see Figure 1, the energy transmitted to the ground is zero, i.e. $E_{gr} = 0$ in equation (1). By use of radiation efficiency methods, the radiated sound energy is represented as [7]

$$E_{rad} = \rho c\, S\, \sigma_r < \overline{\upsilon^2} > \tag{2}$$

where S is the effective area of the radiating surface, which is merely the cylindrical shell surface rather than the whole surface of the structure because of assumption (a). This assumption has been supported by experimental measurements. Now, $< \overline{\upsilon^2} >$ in equation (2) denotes the measured time and space average of the squared response velocity of the cylindrical surface. The dissipated energy E_{struc} due to damping is related to the structural loss factor η_s by the definition, Ref [7]

$$\eta_s = E_{struc} / \omega E_{vib} = E_{struc} / 2\pi f\, E_{vib} \tag{3}$$

where E_{vib} is the average structural vibrational energy. For a cylindrical shell with radiating surface area S and thickness h, equation (3), together with $E_{vib} = \frac{1}{2} M < \overline{\upsilon^2} >$, may be rearranged to be

$$E_{struc} = 2\pi f\, \eta_s\, E_{vib} = \pi f\, \rho_m \eta_s\, S\, h < \overline{\upsilon^2} > \tag{4}$$

where η_s represents the structural loss factor before application of a damping treatment. In practice, for economy, damping material would only be applied to the cylindrical surface and also because the response velocity there is usually one order higher than that of the end plates. When the cylindrical surface of the structure is damped with additive visco-elastic damping material, either as a free or constrained-layer, expressions for E'_{rad} and E'_{struc} becomes

$$E'_{rad} = \rho c\, S\, \sigma_r < \overline{\upsilon_d^2} > \tag{5}$$

$$E'_{struc} = 2\pi f\, \eta'_s\, E_{vib} = \pi\, f \rho_m \eta'_s\, Sh\, (\, 1 + \rho_d\, h_d / \rho_m h\,) < \overline{\upsilon_d^2} > \tag{6}$$

where ρ_d is the equivalent mass density of the damping material, h_d is the thickness of the applied damping material and η_s' is the composite loss factor of the structure with the damping treatment applied. The energy balance equation is then

$$E_{in}' = E_{rad}' + E_{struc}' \tag{7}$$

3. INSERTION LOSS ΔSPL WITH DAMPING

Under excitation with the same input energy, e.g. the ball drops from the same height before and after application of the damping treatment, $E_{in} = E_{in}'$, hence one has

$$E_{rad} + E_{struc} = E_{rad}' + E_{struc}' \tag{8}$$

From equation (8),

$$\frac{<\overline{\upsilon_d^2}>}{<\overline{\upsilon^2}>} = \frac{\rho c \sigma_r + \pi f \rho_m \eta_s h(1+\rho_d h_d/\rho_m h)}{\rho c \sigma_r + \pi f \rho_m \eta_s' h} \tag{9}$$

Equation (9) gives the mean square velocity with added damping relative to that before application of damping material. The relationship between damping value and radiated energy may then be derived,

$$\frac{E_{rad}'}{E_{rad}} = \frac{\rho c S \sigma_r <\overline{\upsilon^2}>}{\rho c S \sigma_r <\overline{\upsilon_d^2}>} = \frac{<\overline{\upsilon^2}>}{<\overline{\upsilon_d^2}>} \tag{10}$$

Combining equation (9) with (10),

$$\frac{E_{rad}'}{E_{rad}} = \frac{\rho c \sigma_r + \pi f \rho_m \eta_s h}{\rho c \sigma_r + \pi f \rho_m \eta_s' h\ (1+\mu)} \tag{11}$$

where

$$\mu = \rho_d h_d / \rho_m h$$

the area density ratio of total additive damping materials to the base material of the cylinder. Noting the relationship between pressure and sound energy, equation (11) may be expressed in dB terms as the insertion loss of sound pressure level

$$\Delta SPL = 10 \log\left(\frac{E_{rad}'}{E_{rad}}\right) = 10 \log\left(\frac{\rho c \sigma_r + \pi f \rho_m \eta_s h}{\rho c \sigma_r + \pi f \rho_m \eta_s' h\ (1+\mu)}\right) \tag{12}$$

Equation (12) shows that the reduction effect is related to the loss factor of the structure after application of the damping treatment, and the initial damping value. It should be emphasised here that an industrial cylindrical structure such as a ball-grinder is composed of a cylindrical shell combined with a steel liner separated by a damping material, therefore the initial damping may be quite high. Thus very high insertion loss of sound pressure level may not always be expected when modifications are made.

4. DISCUSSION OF RESULTS

Equation (12) yields estimates of insertion loss which, for a given cylinder, depends on the loss factor after the damping treatment has been applied. Among the parameters is σ_r, the radiation efficiency, which is usually difficult to calculate for a finite cylinder but may bé replaced by that of an infinite cylinder for practical estimation. Table 1 gives the radiation efficiency of an infinite cylinder with diameter D=530 mm. These data are used in the following example of a practical, closed cylinder.

In air at temperature (20°C) and standard pressure (1 atm), the characteristic acoustic impedance ρc=414 $kgm^{-2}s^{-1}$. The loss factors of the structure, shown in Figure 1, before and after damping treatment, measured by the half-power band width method, were $\eta_s = 0.005$ and $\eta'_s = 0.155$, respectively. The applied damping treatment was damping material SN-303 in two constrained layers, which are mild steel plates 2 mm thick. The density of the cylinder material $\rho_m = 7900$ kg m^{-3}, and the mass ratio of the additive material is $\mu = 1.026$. Then the insertion loss with the applied damping treatment is a function of frequency, depending on the variation of radiation efficiency with frequency.

Experimental measurements have been carried out on a closed cylinder with diameter D=530 mm and length L=600 mm. Impact excitation was performed by dropping a steel ball of 50 mm diameter from the same height before and after application of the damping treatment. The measured sound pressures in Octave frequency bands are listed in table 2, Damp-1 being light damping with the additive damping material 8 mm thick and Damp-2 heavy damping, 16 mm thick, see Figure 1. The data are also presented in Figure 2 by a solid line (before damping treatment) and a dashed line (after damping treatment by Damp-2). The insertion loss of sound pressure level, by Damp-2, is shown in Figure 3, where the solid line represents measured values; while the dashed line shows the predicted values. It can be seen from the figure that there is reasonable agreement between the two sets of data.

The predicted values are generally lower than the measured values in Figure 3. This is partly because the loss factor used in the evaluation of equation (12) was a constant value, which was measured at the lowest natural frequency of the cylinder. In practice, loss factors at high frequency are significantly larger than those at lower frequency in structures with constrained layer damping treatments.

5. CONCLUSIONS

Equation (12) can be used to estimate the insertion loss in radiated sound pressure level from cylinders after application of a damping treatment. The reduction in level of radiated sound not only depends on the loss factor of the structure after damping treatment, but also involves the initial damping value of the structure. It appears that insertion loss may be predicted by sufficient accuracy by equation (12) for engineering calculations, particularly if a form of damping treatment is to be assessed. It is relatively simple to measure modal loss factor at low frequency and this was done in the work reported here; better estimates of loss factor are required for high frequency predictions and this could be achieved by use of a frequency-dependant term or by use of data measured by decay techniques. Further study of large practical structures would be useful.

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APPENDIX: NOMENCLATURE

E_{in}	input energy
E_{rad}	radiated energy
E_{struc}	dissipated energy
E_{vib}	vibrational energy
h	thickness of the shell
h_d	equivalent thickness of additive damping material
S	radiating surface area
SPL	sound pressure level, in dB(A)
ρc	characteristic impedance
ρ_d	volume density of visco-elastic material
ρ_m	volume density of cylindrical shell material
η_s	loss factor of the structure without damping treatment
η_s'	composite loss factor of the structure after damping treatment
σ_r	radiation efficiency
—	spatial average
< >	time average

Table 1. radiation efficiency of an infinite cylinder with diameter D=530 mm

f (Hz)	31.5	63	125	250	500	1 k	2 k	4 k	8 k
σ_r	0.015	0.41	0.68	0.84	0.95	0.98	0.99	1.0	1.0

Table 2. Sound pressure level without and with damping treatment

f (Hz)	L(dB)	A(dB)	31.5	63	125	250	500	1 k	2 k	4 k	8 k
without	106.5	104.5	68.0	74.0	73.0	93.0	105.5	97.5	99.8	91.0	82.0
Damp-1	101.0	99.2	64.0	68.0	71.0	90.5	99.3	95.2	91.0	82.5	73.5
Damp-2	90.5	87.5	57.0	61.0	63.0	84.5	86.5	85.0	83.0	73.5	62.5

* Damp-1: light damping with the thickness of damping material h_d = 8 mm;
Damp-2: heavy damping with the thickness of damping material h_d = 16 mm; see Fig 1.

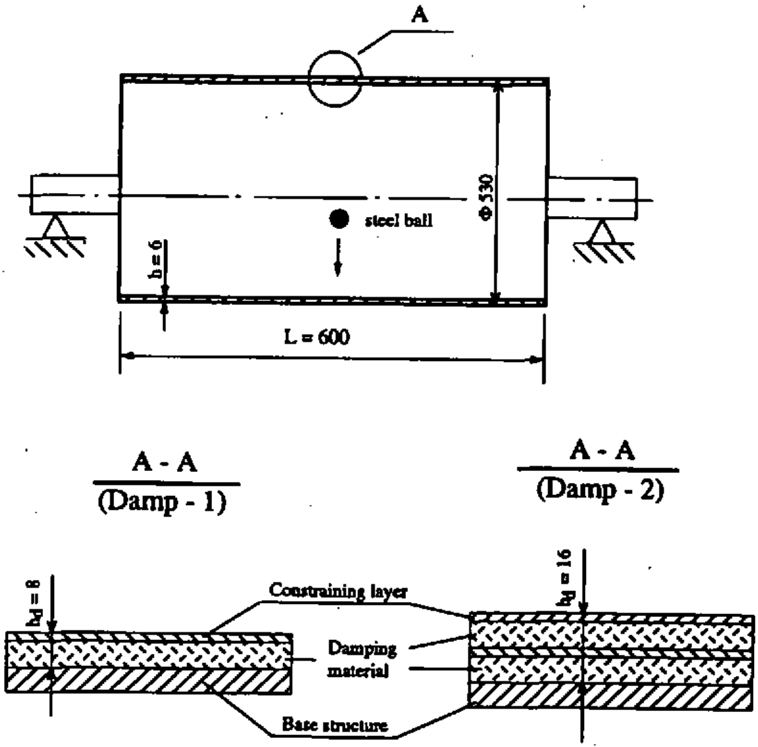

Figure 1. Test model for radiation experiment

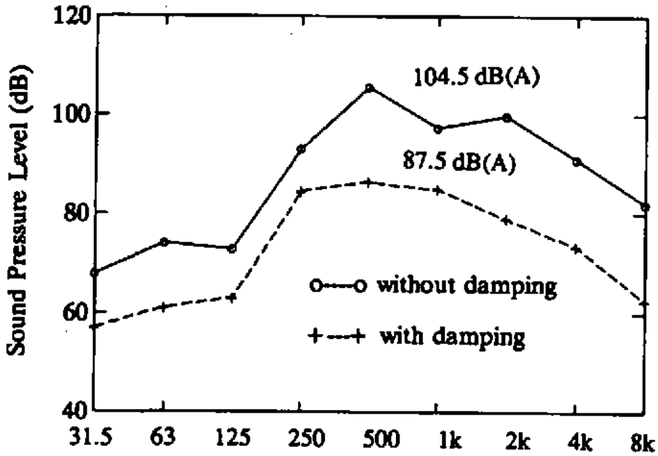

Figure 2. Measured sound pressure level with and without damping in Octave frequency bands

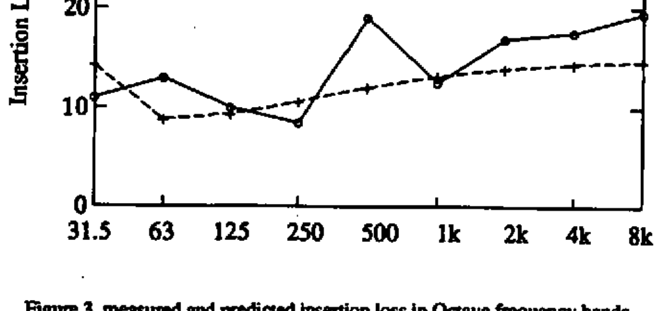

Figure 3. measured and predicted insertion loss in Octave frequency bands