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Journal of Sound and Vibration 282 (2005) 1009–1023
JOURNAL OF SOUND AND VIBRATION
www.elsevier.com/locate/jsvi

An experimental comparison of the sea power injection method and the power coefficient method

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Received 9 October 2003; received in revised form 20 February 2004; accepted 22 March 2004
Available online 14 October 2004

Abstract

A new method, the power coefficient measurement method, was proposed recently by Fahy to measure power transfer and dissipation coefficients. In this paper, the feasibility of this new method is experimentally tested based on a two-plate system. The power injection method is used as a reference to assess the accuracy of the new method. Measurements were carried out under two types of source: continuous steady (shaker) excitation and repeated impact (hammer) excitation. The estimation was made based on two formats of data: velocity autospectra and velocity transfer functions. It is shown that the new method can give comparable results to the power injection method at frequencies where the modal overlap factor is larger than 1. Either the data-recording format or the source type has little impact on the measurement accuracy. The data can be recorded in the format of either velocity autospectrum or transfer function. At frequencies where the modal overlap factor is less than 1, the approximation error of the new method is large resulting in an overestimation.

1. Introduction

Statistical energy analysis (SEA) is a useful tool for predicting sound and vibration transmissions in complex structures such as vessels and buildings. However, the reliability of the prediction is dependant on the accurate estimation of the SEA parameters such as modal

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0022-460X/$ - see front matter
doi:10.1016/j.jsv.2004.03.064

densities, dissipation and coupling loss factors. The theories for predicting SEA parameters are limited only to some simple structures such as uniform plates, beams or cylindrical shells, and under some ideal boundary conditions such as semi-infinitely or simply supported boundaries. In practice, however, most structures are not in simple shapes and their boundary conditions may be difficult to be mathematically described. The best way for accurately estimating the SEA parameters is to measure them in situ.

The accuracy of SEA prediction has been investigated by many authors. Two parameters, modal number (N_m) and modal overlap factor (M_o), are commonly used as the indicators of the reliability of SEA prediction. Computational experiments by Fahy and Mohammed on beam and plate systems [1] indicated that a reliable SEA prediction can be made only if $M_o \geqslant 1$. At low values of modal overlap factor, the distribution functions of the normalized power flow and of the derived coupling loss factors are highly non-normal and it is impossible to estimate confidence limits from the knowledge of standard deviation alone. However, subsequent studies show that these distributions are close to log-normal, therefore allowing confidence estimates to be made [2]. For plates, Fahy and Mohammed [1] applied an extra condition that to obtain a reliable SEA prediction the modal number should be larger than 5 in the frequency bands of analysis. Hopkins [2] recently suggested that for plate systems the conditions of $M_o \geqslant 1$ and $N_m \geqslant 5$ can be used to judge if the SEA prediction is reliable. When these conditions are not met, SEA prediction may contain errors of unknown magnitude.

The power injection (PI) method [3] is universally employed for the in situ determination of SEA loss factors. In this method, both the powers sequentially injected into each SEA subsystem and the corresponding averaged responses of all SEA subsystems are required to be measured. This method however has some disadvantages: (1) input power measurements are prone to errors which could be large; (2) practical physical constraints may prevent the attachment of exciters at appropriate locations; (3) it is very time consuming if the total number (N) of SEA subsystems is large; and (4) the inversion of a large dimension (N^2) energy matrix could lead to numerical problems. To overcome a possible ill-condition of the energy matrix, Cuschieri and Sun [4] separated the estimations of internal and coupling loss factors.

Recently, Fahy [5,6] argued that SEA loss factors are not the most appropriate form of coefficients for relating energy dissipation in and transfer between SEA subsystems. He suggested an alternative formulation in terms of "power coefficients", which are proportional to the product of modal densities and loss factors, and most significantly, he proposed a new experimental method, the power coefficient measurement (PC) method, for the determination of the power coefficients without involving in any input power measurement. Similar to the energy level difference method [7,8] and the reduced analysis technique [9], this new method theoretically only gives approximate results. In comparison with the power injection method, this new method has the advantages of:

- much easier to implement, especially for practical and complex structures;
- less measurement errors because it does not involve in any input power measurement.

The power coefficient measurement method only requires measuring the velocity of source positions, while the power injection method needs to measure the input power. The measurement of velocity can be easily realized using accelerometers. However, the measurement of input power may be very difficult or impossible in some cases. For example, no effective techniques are available yet to measure the input power of distributive force or moment excitations. Even for the point force excitations, the attachment of an impedance head requires more space than that of an accelerometer. Therefore, the PC method has broader application than the PI method.

Up to now, however, the feasibility of the power coefficient measurement method has not yet been experimentally tested. This paper presents the first attempt to carry out this task. For simplicity, the test model consists of two uniform plates. Therefore, the hypothesis in the original proposal that the PC method might also be applicable to non-uniform structures is not tested. The PI method is taken as a reference. Both the continuous and repeated impact excitation sources are employed to examine the effect of source type on the measurement accuracy.

2. Theory

2.1. The power injection method

For a structure consisting of N SEA subsystems, the power balance equation can be written as follows:

$$\omega \begin{bmatrix} \sum \eta_{1i} & -\eta_{21} & \cdots & -\eta_{N1} \\ -\eta_{12} & \sum \eta_{2i} & \cdots & -\eta_{N2} \\ \vdots & & & \vdots \\ -\eta_{1N} & -\eta_{2N} & \cdots & \sum \eta_{Ni} \end{bmatrix} \begin{bmatrix} E_1 \\ E_2 \\ \vdots \\ E_N \end{bmatrix} = \begin{bmatrix} P_1 \\ P_2 \\ \vdots \\ P_N \end{bmatrix}, \tag{1}$$

where $\omega = 2\pi f$ is the radian frequency; η_{ii} is the internal loss factor of subsystem i; η_{ij} is the coupling loss factor from subsystem i to j; E_i is the total energy stored in subsystem i and P_i is the power input to subsystem i. For the purpose of loss factor measurements, the above equation may be written as

$$[E][\eta] = [P/\omega], \tag{2}$$

where $[E]$ is $N \times N^2$ matrix; $[\eta] = [\eta_{11}\ \eta_{12}\ \eta_{13}\ \ldots\ \eta_{N(N-1)}\ \eta_{NN}]^{\mathrm{T}}$ is $N^2 \times 1$ matrix and $[P/\omega] = [P_1/\omega\ P_2/\omega\ \ldots\ P_N/\omega]^{\mathrm{T}}$. The superscript T represents matrix transpose. For the PI method, the power is sequentially injected into each subsystem. When each subsystem is excited, the responses in all subsystems are measured and a set of equations can be formed from Eq. (2). After all subsystems are excited, the following matrix equation can be obtained:

$$\begin{bmatrix} [E]_1 \\ [E]_2 \\ \vdots \\ [E]_N \end{bmatrix} [\eta] = \begin{bmatrix} [P/\omega]_1 \\ [P/\omega]_2 \\ \vdots \\ [P/\omega]_N \end{bmatrix}, \tag{3}$$

where the subscript i ($i = 1, 2, \ldots, N$) represents that the ith subsystem is excited. For $N = 2$, the above equation becomes

$$\begin{bmatrix} E_{1,1} & E_{1,1} & -E_{2,1} & 0 \\ 0 & -E_{1,1} & E_{2,1} & E_{2,1} \\ E_{1,2} & E_{1,2} & -E_{2,2} & 0 \\ 0 & -E_{1,2} & E_{2,2} & E_{2,2} \end{bmatrix} \begin{bmatrix} \eta_{11} \\ \eta_{12} \\ \eta_{21} \\ \eta_{22} \end{bmatrix} = \begin{bmatrix} P_1/\omega \\ 0 \\ 0 \\ P_2/\omega \end{bmatrix}, \tag{4}$$

where $E_{i,j}$ is the total energy stored in subsystem i when subsystem j is excited. When N becomes large, the coefficient matrix in Eq. (4) may be ill conditioned. To overcome such a problem, Cuschieri and Sun [4] rearranged Eq. (4) into two separate matrix equations.

2.2. Power coefficient method

The power coefficient is defined as [5]

$$M_{ij} = \omega n_i(\omega)\eta_{ij} = f n_i(f)\eta_{ij}, \tag{5a,5b}$$

where M_{ij} is called as "power transfer coefficient" if $i \neq j$, or "power dissipation coefficient" if $i = j$. From the SEA reciprocity relation, $M_{ij} = M_{ji}$ holds. $n_i(f) = 2\pi n_i(\omega)$ is the modal density of subsystem i and, for spatially uniform structures, can be estimated from the real part of measured mobility Y_i [10]

$$n_i(f) \approx 4m_i\langle \mathrm{Re}\{Y_i\}\rangle, \tag{6a}$$

where m_i is the total mass of subsystem i; Re{ } represents the real part and $\langle\,\rangle$ denotes the average over the frequency band and the surface area. If subsystem i is a plate, the asymptotic is theoretically given by [11]

$$n_i(f) = \frac{\sqrt{3}S_i}{h_i c_{il}}, \tag{6b}$$

where c_{il} is the longitudinal wavespeed; S_i is the surface area and h_i is the thickness of the plate.

If the vibration velocity at the driving position has an uniform spectral density and the frequency band of interests contains a number of modal resonance frequencies, the input power, averaged over all excitation positions, may be approximated as [5,6]

$$\langle P_{in}\rangle_{\Delta\omega} \approx \frac{2m}{\pi n(\omega)}\langle v_{in}^2\rangle_{\Delta\omega}, \tag{7}$$

where m is the total mass of the source subsystem; v_{in} is the vibration velocity at the source position and $\langle\,\rangle_{\Delta\omega}$ represents the average over the frequency band $\Delta\omega$. Fahy suggested [5] that the above equation also may hold for non-uniform structures if the actual mass m in the above equation is replaced by an effective mass related to the choice of excitation points.

Substituting Eqs. (5a) and (7) into Eq. (1) gives

$$\begin{bmatrix} \sum M_{1i} & -\alpha_{12}M_{12} & \dots & -\alpha_{1N}M_{1N} \\ -\alpha_{21}M_{12} & \sum M_{2i} & & \vdots \\ \vdots & & & \vdots \\ -\alpha_{N1}M_{1N} & \dots & \dots & \sum M_{Ni} \end{bmatrix} \begin{bmatrix} \langle v_1^2 \rangle \\ \langle v_2^2 \rangle \\ \vdots \\ \langle v_N^2 \rangle \end{bmatrix} = \frac{2}{\pi} \begin{bmatrix} \langle v_{1in}^2 \rangle_{\Delta\omega} \\ \langle v_{2in}^2 \rangle_{\Delta\omega} \\ \vdots \\ \langle v_{Nin}^2 \rangle_{\Delta\omega} \end{bmatrix}, \tag{8}$$

where $\alpha_{ij} = n_i m_j / n_j m_i$ and $\alpha_{ji} = 1/\alpha_{ij}$. For the purpose of power coefficient measurements, the above equation may be rewritten as

$$[V][M] = \frac{2}{\pi}[V_{in}], \tag{9}$$

where $[M] = [M_{11}\ M_{12}\ M_{13}\ \dots\ M_{1N}\ M_{22}\ M_{23}\ \dots\ M_{(N-1)N}\ M_{NN}]^{\mathrm{T}}$ is $(N(N+1)/2) \times 1$ matrix; $[V_{in}] = [\langle v_{1in}^2 \rangle_{\Delta\omega}\ \langle v_{2in}^2 \rangle_{\Delta\omega}\ \dots\ \langle v_{Nin}^2 \rangle_{\Delta\omega}]^{\mathrm{T}}$ and $[V]$ is $N \times (N(N+1)/2)$ matrix. To obtain the values of power coefficients, the same procedure as that for the power injection method should be implemented. That is, every SEA subsystem is sequentially excited and the responses of all subsystems are measured. For each excitation, a matrix equation can be written based on Eq. (9). After obtaining the data from N sequential excitations on each subsystem, the power coefficients can be estimated using the following equation:

$$[M] = \frac{2}{\pi} \left(\begin{bmatrix} [V]_1 \\ [V]_2 \\ \vdots \\ [V]_N \end{bmatrix}^{\mathrm{T}} \begin{bmatrix} [V]_1 \\ [V]_2 \\ \vdots \\ [V]_N \end{bmatrix} \right)^{-1} \begin{bmatrix} [V]_1 \\ [V]_2 \\ \vdots \\ [V]_N \end{bmatrix}^{\mathrm{T}} \begin{bmatrix} [V_{in}]_1 \\ [V_{in}]_2 \\ \vdots \\ [V_{in}]_N \end{bmatrix}. \tag{10}$$

For $N = 2$, $[[V_{in}]_1 [V_{in}]_2]^{\mathrm{T}} = [\langle v_{1in}^2 \rangle_{\Delta\omega}\ 0\ 0\ \langle v_{2in}^2 \rangle_{\Delta\omega}]$ and

$$\begin{bmatrix} [V]_1 \\ [V]_2 \end{bmatrix} = \begin{bmatrix} \langle v_1^2 \rangle_1 & \langle v_1^2 \rangle_1 - \alpha_{12}\langle v_2^2 \rangle_1 & 0 \\ 0 & \langle v_2^2 \rangle_1 - \alpha_{21}\langle v_1^2 \rangle_1 & \langle v_2^2 \rangle_1 \\ \langle v_1^2 \rangle_2 & \langle v_1^2 \rangle_2 - \alpha_{12}\langle v_2^2 \rangle_2 & 0 \\ 0 & \langle v_2^2 \rangle_2 - \alpha_{21}\langle v_1^2 \rangle_2 & \langle v_2^2 \rangle_2 \end{bmatrix}.$$

In practice, however, a velocity of uniform spectral density may be difficult to be realized at driving positions. If the velocity at the driving position has non-uniform spectral density, Fahy suggested [6]: "the procedure may be correctly implemented by normalizing response velocity spectra by the associated driving point velocity autospectrum. Alternatively, the frequency average of the square of the modulus of the velocity transfer functions may be derived using either continuous or repeated impact excitations". In this case, the elements of matrix $[V_{in}]$ in Eqs. (9) and (10) should be unity and those of $[V]$ should be the square of magnitude of averaged velocity transfer functions if they are not zeros.

2.3. Normalized difference

Theoretically, Eq. (10) only gives an approximation estimation of power coefficients because Eq. (7) is used during its derivation. The approximation error may be difficult to mathematically formulate and will not be discussed here. In the following, the accuracy of measured results will be assessed based on the normalized difference between the results obtained using the PI (reference) method and the PC method. The normalized difference (ε) is defined as

$$\varepsilon = \frac{M_{PC} - M_{PI}}{M_{PC}} \times 100\% \,, \tag{11}$$

where M_{PC} is the results obtained using the PC method and M_{PI} is the reference value obtained using the PI method. The normalized difference is not equal to the normalized error because the reference value (M_{PI}) may contain measurement errors. In the following figures, all the normalized differences are calculated using the above equation.

3. Experimental verification

3.1. Setup

The test model consisted of two uniform rectangular aluminum plates, which were joined together by 14 bolts at a right angle along one common edge (0.8 m), and was freely hung by 4 metal wires, as shown in Fig. 1. The surface area was $0.9 \times 0.8\,\mathrm{m}^2$ for plate 1 and $1.1 \times 0.8\,\mathrm{m}^2$ for plate 2. The thicknesses of plates 1 and 2 were 3 and 2 mm, respectively. Self-adhesive damping strips (Idikell, $2.6\,\mathrm{kg/m}^2$) were attached to the plate surfaces (about 27.5% coverage for both plates) to reduce the reactive component of power flow for improving the measurement accuracy of the PI method.

Fig. 1. Test plate system.

Two types of excitations were used: continuous steady (shaker) excitation and repeated impact (hammer) excitation. During the measurements, the analyzer was set in an FFT mode with a frequency resolution of 8 Hz (800 lines over the frequency range from 0 to 6.4 kHz) and the responses were measured using *BK4375* accelerometers. The square of narrow frequency band transfer function magnitudes of the velocities at the source location and at the randomly selected locations of plates was obtained from the corresponding measured narrow frequency band velocity autospectra (before they were converted into one-third octave frequency bands). During the data processing, the narrow frequency band data were firstly converted into the one-third octave frequency band ones and then the calculations were made based on the converted results.

For the case of continuous steady excitation, the plates were excited by a *BK3800* shaker, fed with pseudorandom noise signals. Since the PI method requires the measurement of input power, a *BK8001* impedance head was mounted at the driving location. The shaker and the impedance head were connected by a steel rood of 30 mm in length and 1 mm in diameter to avoid the transmission of possible transverse force which could cause errors in the input power measurement. The input power was calculated from the imaginary part of the cross spectrum, $G(F, a, \omega)$, where F and a are the force and acceleration signals output from the impedance head

$$P_{\text{input}} = \frac{1}{\omega} \operatorname{Im}\{G(F, a, \omega)\}. \tag{12}$$

Three driving locations were randomly chosen in each plate [3]. For each excitation, 11 accelerometer positions were randomly selected in each plate to measure the responses. The individual values of the SEA loss factors determined by the PI method and the power coefficients determined by the PC method were estimated from the data obtained under a pair of excitations (one is on plate 1 and another on plate 2). The average values are obtained over three individual results.

For the case of repeated impact excitation, an impact hammer was used with a plastic tip for the low-frequency range upto 500 Hz and a steel tip for the frequency range above 500 Hz. During the measurements, the hammer repeatedly hit the locations, which were around and close (as possible) to the source location accelerometer. Care was taken to keep the applied hammer forces in a constant level as possible. The signals were rejected if any channel was overloaded. For the use of a steel tip, a high-frequency pass filter was used to filter the low-frequency components of hammer location response signals because they usually caused the overloading. For each hammer excitation, the responses at the hammer location and at a randomly selected position of each plate were recorded simultaneously. Both the signal-recording and hammer-hitting time periods were set to 40 s. Totally 30 hammer (excitation) positions were randomly selected in each plate. The hammer position number was chosen to ensure the maximum standard deviation of the measured accelerations on the plates was less than a pre-selected value. It was noticed that the standard deviation decreases with the increase of hammer position number. The individual power coefficients were estimated using Eq. (10) from the data obtained under a pair of hammer excitations (one is on plate 1 and another on plate 2). The average values are obtained over 30 individual results.

4. Results and discussions

Fig. 2 shows the comparisons of the modal numbers estimated from the predicted (using Eq. (6b)) and measured (using Eq. (6a)) modal densities for plates 1 and 2. It can be seen that the modal number of both plates increases with frequency and is over-predicted at low frequencies. The modal number in plate 1 is smaller than that of plate 2 in all the frequency bands. This is because plate 2 is thinner and has a larger surface area. The modal number is larger than 5 in the frequency bands above 160 Hz for plate 2 and above 315 Hz (if predicted values are used) or 500 Hz (if measured values are used) for plate 1.

Figs. 3 and 4 show the comparisons (a) between the input power for the continuous steady (shaker) excitations and its approximation $(2m(\pi n(\omega))\langle v_{in}^2\rangle_{\Delta\omega})$, and the corresponding normalized differences (b) for plates 1 and 2. The normalized differences are calculated using Eq. (11) where M_{PC} is replaced by the input power and M_{PI} by the approximation. The input power was estimated using Eq. (12) from the data measured using the impedance head. It is shown that the normalized differences decreases with increasing frequency. At frequencies where the modal number is larger than 5, the difference becomes small and the use of Eq. (6a) to estimate modal density results in an overestimation of the power flow while that of Eq. (6b) in an underestimation. At frequencies where the modal number is less than 5, the difference is large and Eq. (7) overestimates the input power. This is because one of the requirements [5] for deriving Eq. (7) is not met in these frequency bands.

Figs. 5–7 show the comparisons of the power dissipation and transfer coefficients measured using the PC method and estimated from the SEA loss factors measured using the PI method under the conditions of continuous steady (shaker) excitation. It can be seen that the results obtained using the PC method are insensitive to the choice of estimation (using Eq. (6a) or (6b)) of modal densities at all the frequency bands of interests while those using the PI method are. This is because the power coefficients obtained using the PI method are proportional to the modal

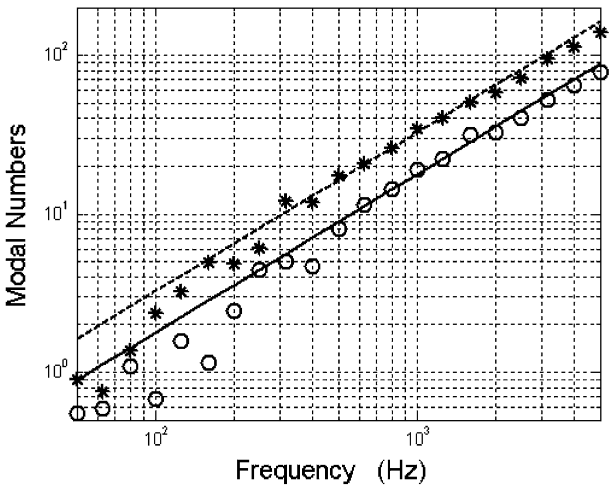

Fig. 2. Comparisons of the modal numbers estimated from the predicted (plate 1: —; plate 2: - - -) and measured (plate 1: ○; plate 2: ✱) modal densities.

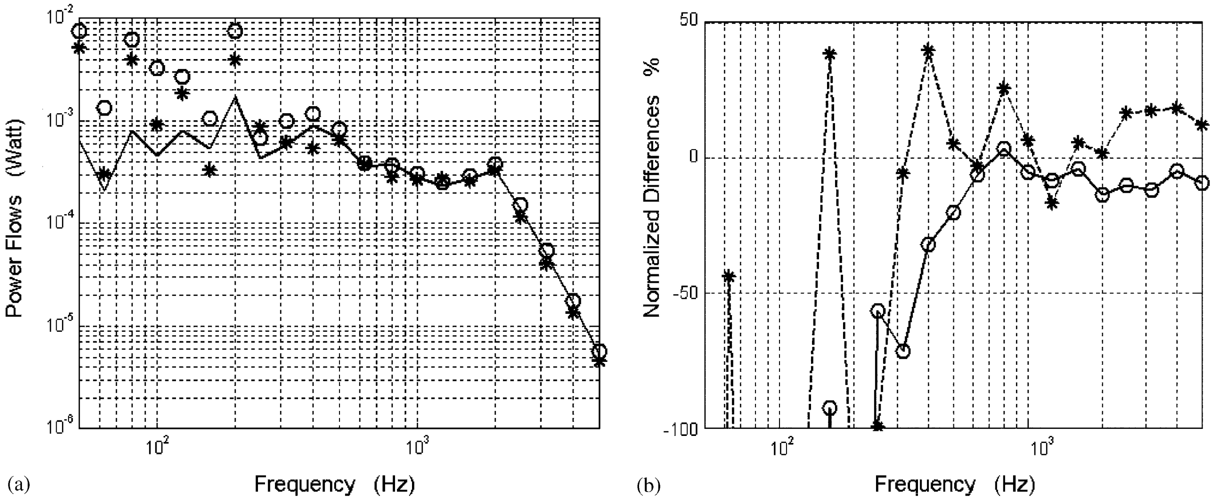

Fig. 3. Comparison (a) and normalized differences (%) (b) of the input power in plate 1 (—) and its approximation where the modal density is estimated using Eq. (6a): ○; or (6b): ✱.

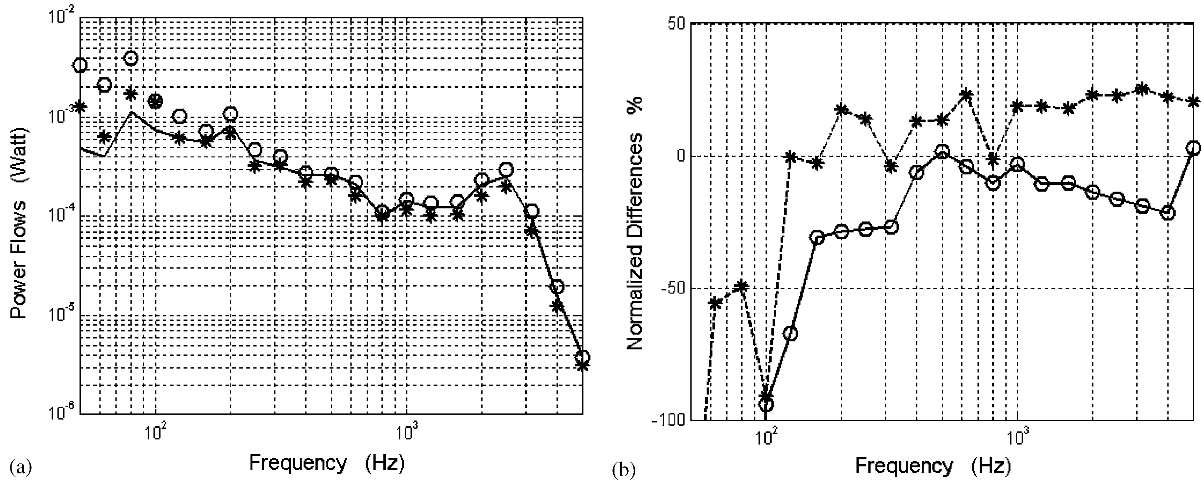

Fig. 4. Comparison (a) and normalized differences (%) (b) of input power in plate 2 (—) and its approximation where the modal density is estimated using Eq. (6a): ○; or (6b): ✱).

densities of subsystems while those using the PC method are not. The PC method only requires the knowledge of modal density ratios rather than the values of modal densities. In some cases it is possible that the errors of individual modal densities are cancelled each other to some degrees resulting in a less error of a modal density ratio. This is one of the advantages of the PC method.

If the estimation is made from the SEA loss factors obtained using the PI method, the power dissipation coefficient is larger than 1 at frequencies above 160 Hz for plate 2 and above 315 Hz (if Eq. (6b) is used) or 500 Hz (if Eq. (6a) is used) for plate 1. The power dissipation coefficient is the

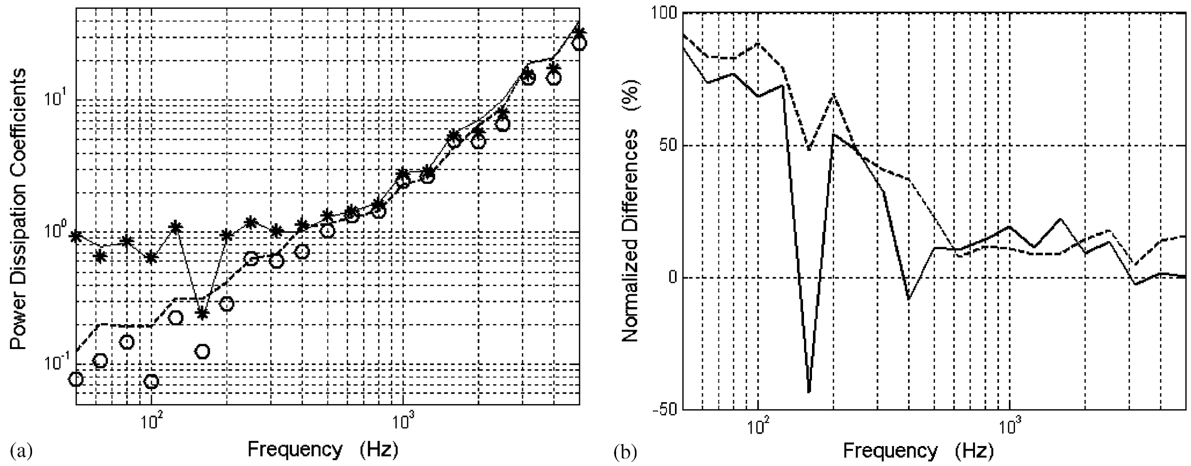

Fig. 5. Comparison (a) and normalized differences (%) (b) of M_{11} directly measured using the PC method ($n(f)$ is estimated using Eq. (6a): —— or (6b): - - -) and estimated from η_{11} measured using the PI method ($n(f)$ is estimated using Eq. (6a): ○; or (6b): ✱) under the conditions of continuous steady excitation.

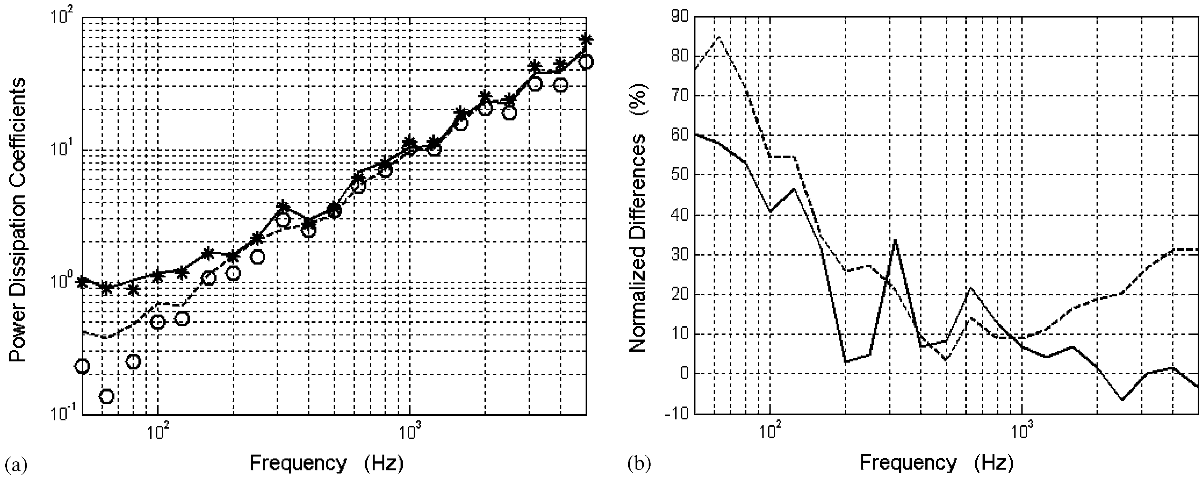

Fig. 6. Comparison (a) and normalized differences (%) (b) of M_{22} directly measured using the PC method ($n(f)$ is estimated using Eq. (6a): —— or (6b): - - -) and estimated from η_{22} measured using the PI method ($n(f)$ is estimated using Eq. (6a): ○; or (6b): - - -) under the conditions of continuous steady excitation.

modal overlap factor (M_o), which represents the degree of modal overlap of the uncoupled modes of each subsystem [5]. For this test model, the condition of $M_o \geqslant 1$ is satisfied when $N_m \geqslant 5$ (see Fig. 2). It can be seen that at frequencies where the power dissipation coefficient is larger than 1 ($M_o \geqslant 1$), the results obtained using the PI method and the PC method agree well and, from an experimental point of view, these two methods give identical results. At frequencies where the power dissipation coefficient is smaller than 1 ($M_o \leqslant 1$), the PC method gives higher values than

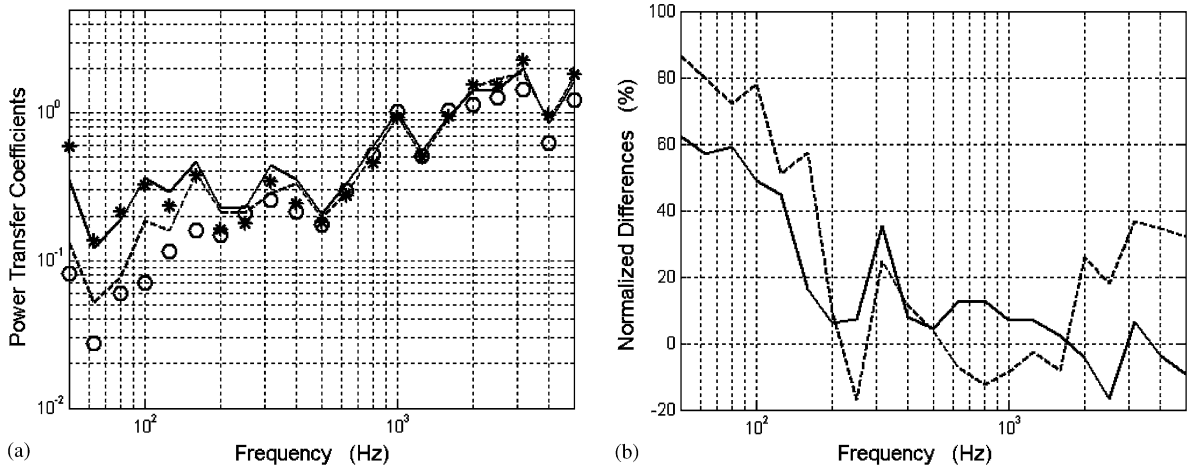

Fig. 7. Comparison (a) and normalized differences (%) (b) of M_{12} directly measured using the PC method ($n(f)$ is estimated using Eq. (6a): — or (6b): - - -) and estimated from η_{12} measured using the PI method ($n(f)$ is estimated using Eq. (6a): ○; or (6b): ✱) under the conditions of continuous steady excitation.

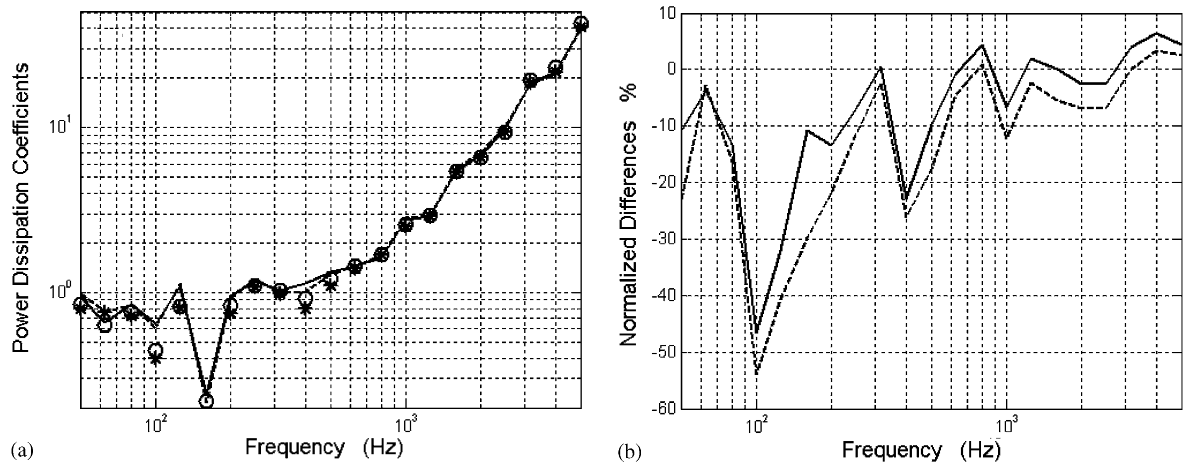

Fig. 8. Comparison (a) and normalized differences (%) (b) of M_{11} estimated from the velocity autospectra ($n(f)$ is estimated using Eq. (6a): — or (6b): - - -) and from the velocity transfer functions ($n(f)$ is estimated using Eq. (6a): ○; or (6b): ✱).

the PI method and the difference between the two results increases with decreasing modal overlap factor. It may be concluded from these results that the PC method can give reliable results if the conditions for ensuring an accurate SEA prediction ($M_o \geqslant 1$ and $N_m \geqslant 5$) are satisfied.

Figs. 8–10 show the comparisons (a) of the power dissipation and transfer coefficients estimated from different data formats, velocity autospectra and velocity transfer functions, and the corresponding normalized differences (b), which are calculated using Eq. (11) where M_{PC} represents the results obtained from velocity autospectra and M_{PI} from the velocity transfer

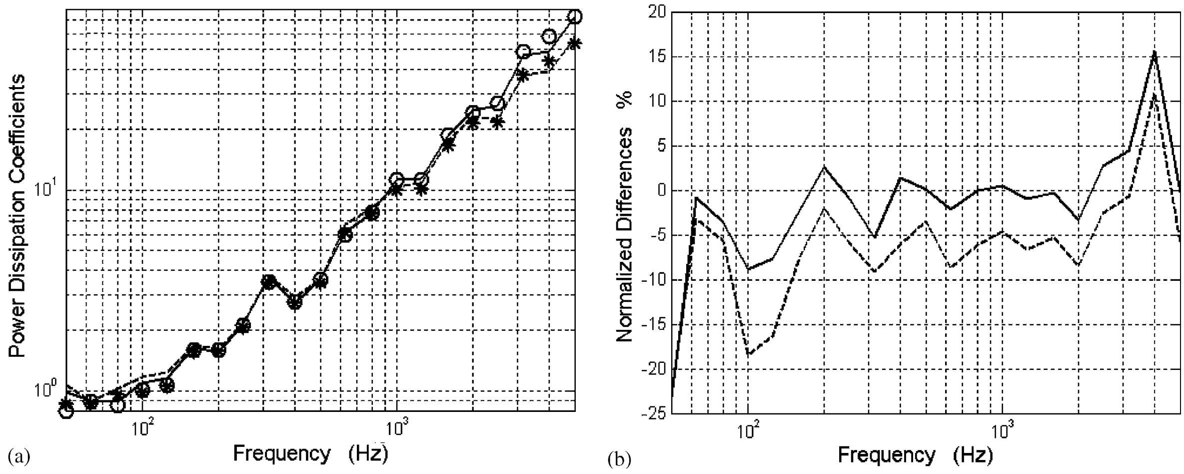

Fig. 9. Comparison (a) and normalized differences (%) (b) of M_{22} estimated from the velocity autospectra ($n(f)$ is estimated using Eq. (6a): — or (6b): - - -) and from the velocity transfer functions ($n(f)$ is estimated using Eq. (6a): O; or (6b): ✱).

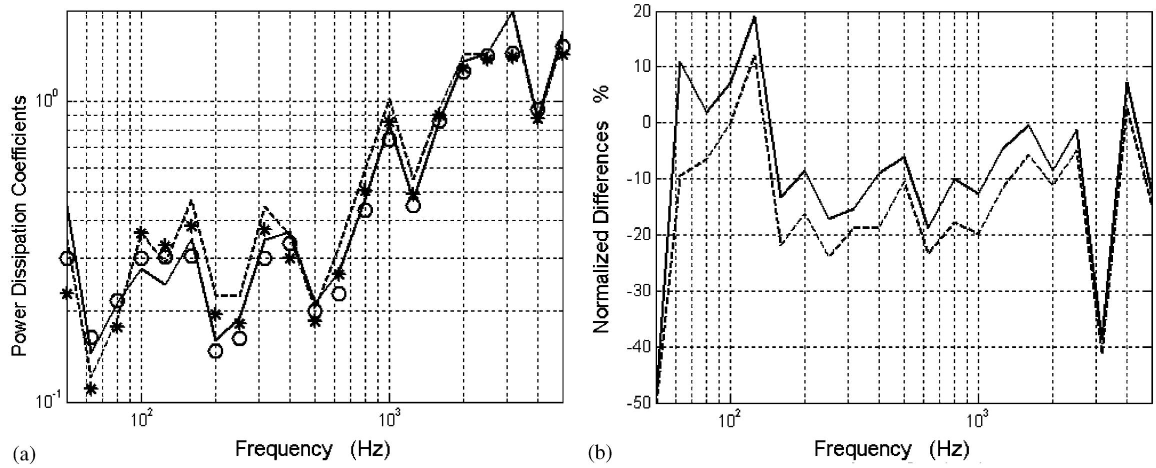

Fig. 10. Comparison (a) and normalized differences (%) (b) of M_{12} estimated from the velocity autospectra ($n(f)$ is estimated using Eq. (6a): — or (6b): - - -) and from the velocity transfer functions ($n(f)$ is estimated using Eq. (6a): O; or (6b): ✱).

functions. The plates were driven by a shaker fed with pseudorandom noise signals (under the continuous steady excitation conditions). It is shown that two estimations are very similar in most frequency bands regardless of the values of modal overlap factor and modal number. A large normalized difference appears at 3.15 Hz for M_{12} and at 4.0 kHz for M_{22}. It was found by examining the narrow frequency band data that the velocity measured at some receiving locations was very low and close to the background noise at some frequencies above 3.5 kHz. Though a

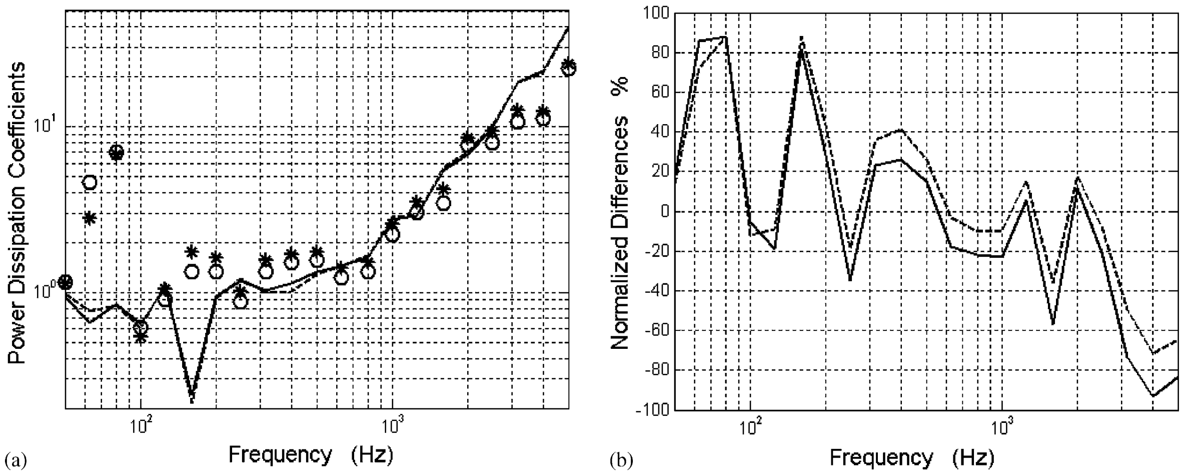

Fig. 11. Comparison (a) and normalized differences (%) (b) of M_{11} measured using the PC method under the conditions of continuous steady excitation ($n(f)$ is estimated using Eq. (6a): —— or (6b): - - -) and under the repeated impact excitation ($n(f)$ is estimated using Eq. (6a): ○ or (6b): ✱).

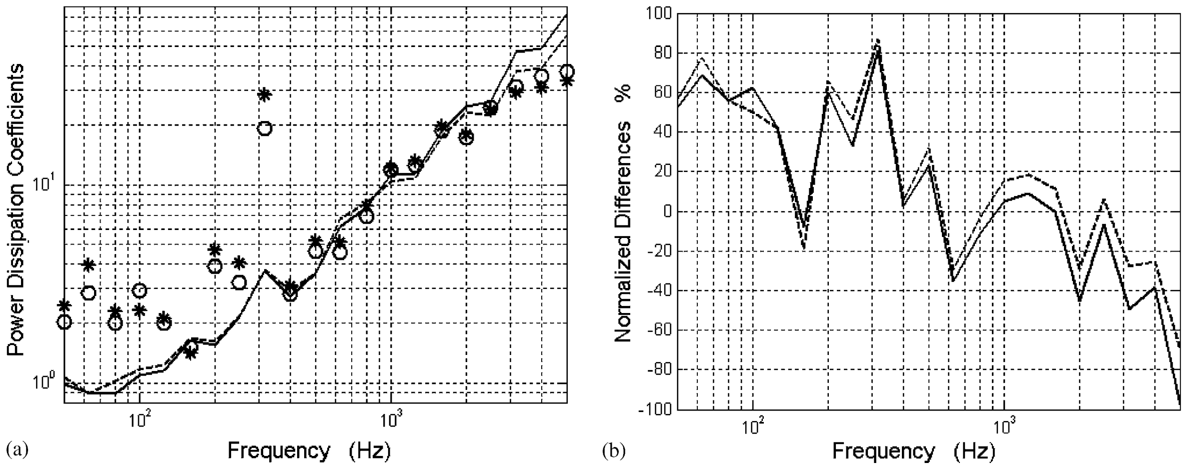

Fig. 12. Comparison (a) and normalized differences (%) (b) of M_{22} measured using the PC method under the conditions of continuous steady excitation ($n(f)$ is estimated using Eq. (6a): —— or (6b): - - -) and under the repeated impact excitation ($n(f)$ is estimated using Eq. (6a): ○ or (6b): ✱).

correction has been made to eliminate the effects of the background noise during the data processing, the corrected data may still not be accurate resulting in a large normalized difference. This indicates that if the measured velocity signal is not corrupted, the measurement of the PC method can be performed by recording either the velocity spectra or the transfer functions.

Figs. 11–13 show the comparisons of the power dissipation and transfer coefficients measured under different source conditions: continuous (shaker) and repeated impact (hammer) excitations.

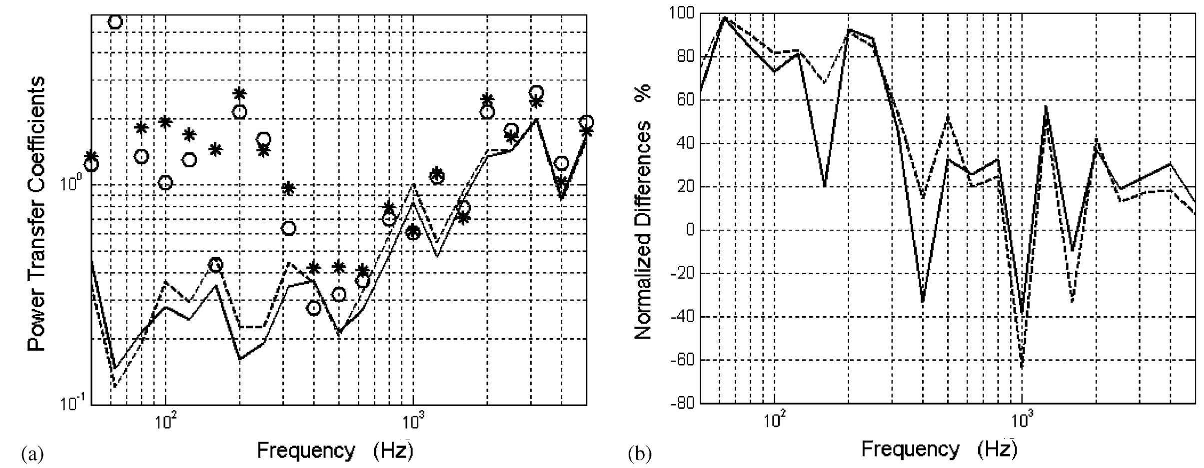

Fig. 13. Comparison (a) and normalized differences (%) (b) of M_{12} measured using the PC method under the conditions of continuous steady excitation ($n(f)$ is estimated using Eq. (6a): —— or (6b): - - -) and under the repeated impact excitation ($n(f)$ is estimated using Eq. (6a): ○ or (6b): ✱).

At most low frequencies, the results obtained under the hammer excitation are higher than those under the shaker excitation and the difference between them is large. At frequencies where $M_o \geqslant 1$, the agreement between the two results is reasonable good (most of the normalized differences are less than 1.5 dB) except for the power dissipation coefficients at frequencies above 3 kHz, for which there are two possible reasons: one is stated in the above and another is that the plates were heavily damped and, for the case of hammer excitations where the input power was low at high frequencies, the vibration field in the receiving plates was not reverberant. It was noticed during the impact measurements that at frequencies above 2.6 kHz the response of the receiving plates was low and close to the background noise level and its level decreased with the distance from the joint (source). Although the effects of background noises were corrected during the data processing, the energy of the receiving plates may not be correctly estimated from the measured data at frequencies above 2.6 kHz. If the plates were heavily hit, the whole system was swung with large amplitudes and the spots were noticeable at the hammer positions of plate surfaces. In practice, the boundaries of structures are usually restrained and a required force (or input power) could be made by a hammer without causing any swing of the whole system. Therefore, it is possible in practice to obtain reliable results using the PC method under the impact hammer excitation at high frequencies.

5. Conclusions

The feasibility of the PC method proposed by Fahy has been experimentally tested based on a two-plate system. It is shown that the accuracy of the PC method is comparable with that of the PI method at frequencies where SEA prediction is accurate ($M_o \geqslant 1$), and is little affected by the data-recording format and by the source type. The data can be recorded in the format of either velocity autospectrum or transfer function. At frequencies where a large variance of SEA prediction is possible ($M_o \leqslant 1$), the approximation error of the PC method is large resulting in an overestimation.

Acknowledgments

Financial support for this work from the Australian Research Council and Strategy Marine Ltd is acknowledged. Many thanks to Prof. F.J. Fahy, the Emeritus Professor of ISVR, Southampton University, UK, for his valuable comments on the first draft of this paper. Thanks also to Prof. J. Pan of the University of Western Australia for his useful suggestions.

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