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Source substitution method for obtaining the power transmission from vibrating sources in buildings

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ABSTRACT

This paper describes a method analogous to the airborne sound source substitution method, to estimate the vibrational power injected by a structure-borne sound source into the supporting building element. The injected vibrational power is required for prediction of the structure-borne sound pressure from vibrating equipment in buildings. The paper focuses on high-mobility sources connected to low-mobility receivers, a situation which is commonly encountered in heavyweight construction. The mobility mismatch simplifies the transformation of laboratory measurement data to prediction of transmitted power in-situ. Three case studies were performed. In the first study, the power injected by a simple test source into a resiliently supported aluminium plate was determined using direct and indirect methods. Source substitution was investigated with different calibration options: steady-state excitation, transient excitation, and spatial averaging. The source power could be determined within 4 dB, compared with direct measurements of the injected power. In the second study, the power injected by a second source into a concrete transmission suite floor was determined. The third study was of a combined heating and power unit on a masonry wall. In this study, a reference sound pressure level in a receiver room was calculated and compared with a criterion curve for the assessment of low-frequency noise complaints. The case studies demonstrate that structure-borne sound source substitution is a promising development of the reception plate method. While the latter can be used if a free reception plate is available, the former circumvents problems of determining the transmitted power into coupled plates and therefore has application to real building conditions. The use of the instrumented hammer for the calibration and the use of spatial averaging significantly simplify the method.

1. Introduction

The most important quantity for the calculation of structure-borne sound from vibrating sources into buildings is the power transmitted into supporting and other connected building elements [1,2]. The transmitted power provides the input into energy-based prediction models, such as Statistical Energy Analysis [3,4] or standardized procedures based on it [5], used for the prediction of sound pressure levels in buildings due to service equipment. For airborne sound power, there is a range of national and international standards using different measurement methods [6]: from sound pressure in reverberation chambers [7,8] or anechoic chambers [9,10]; by intensity methods [11,12], and by source substitution [13,14]. This paper describes a method analogous to the airborne source substitution method, and discusses its advantages and disadvantages. It focuses on high-mobility sources connected to low-mobility receivers, a situation which is common in heavyweight buildings. The mobility mismatch allows the assumption that the source behaves similarly on different receiver structures and simplifies the transformation of laboratory measurement data to prediction of transmitted power in-situ. This paper also considers laboratory methods using isolated reception plates, for comparison with the source substitution method.

2. Isolated reception plates

A vibrating device, connected to an isolated (i.e. resiliently supported) reception plate, transmits power into the plate. When a steady state is reached, the transmitted power from the source into the reception plate is equal to the energy loss of the plate [15]. By plate energy is meant that determined by the bending wave field; other components of vibration are assumed secondary. Fig. 1, left, illustrates the Statistical Energy Analysis (SEA) model of the process. Since the

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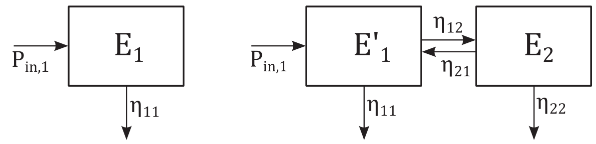

Fig. 1. SEA model of an isolated reception plate, left, and of two connected plates, right.

isolated reception plate is not connected to any other structures, there is only one subsystem in the model.

The power injected into the plate of area A and mass per area m'' can be calculated as:

$$P_{in,1} = \omega\eta_{11} m'' A \langle v^2 \rangle. \quad (1)$$

The frequency variable ω in Eq. (1) indicates that the power injected into the plate is generally frequency-dependent. In an SEA framework, calculation and results are often captured in one-third octave band levels. The mean square velocity $\langle v^2 \rangle$ in Eq. (1) is approximated by sampling the bending velocity field on the plate. The total loss factor η_{11} can be estimated from the structural reverberation time T_s [16]:

$$\eta \approx \frac{2.2}{fT_s}. \quad (2)$$

To minimise errors due to dominant plate eigenmodes, a minimum number of modes per frequency band is recommended, e.g. five or more modes in the frequency band of interest. An alternative indicator is the modal density (the number of modes per Hertz). The asymptotic modal density n_∞ of a thin plate of bending stiffness B is [16]

$$n_\infty = \frac{\Delta N}{\Delta f} = \frac{A}{2}\sqrt{\frac{m''}{B}}. \quad (3)$$

Modal density should be considered with total loss factor. If the total loss factor is low, the individual modes have a high Q factor and do not overlap sufficiently. A more appropriate measure is the modal overlap factor M, which is the ratio of the half-power bandwidth to the average frequency spacing between eigenfrequencies. For plate-like structures in buildings, Hopkins suggests a modal overlap factor of unity [16] as a lower limit for applying SEA. The modal density can be increased by increasing the plate area A and reducing the thickness h. An increase of the plate area has practical limitations. Reducing the thickness causes an unwanted increase in the plate mobility. Reference is made to the infinite plate mobility [15]:

$$Y_\infty = \frac{1}{8\sqrt{Bm''}}. \quad (4)$$

The infinite plate mobility is the mobility of a plate of the same properties as the actual plate but of infinite extent. It is real-valued and frequency-invariant. The asymptotic values in Eqs. (3) and (4) are simply related as

$$Y_\infty = \frac{n_\infty}{4Am''}. \quad (5)$$

Therefore, the requirement for a low mobility plate with a high modal density conflict.

A further challenge for the determination of the mean square velocity concerns the distribution of the plate bending energy. A large proportion of the energy of a free plate is stored along the edges and in the corners, especially at low frequencies. This is equivalent to the increase in sound pressure level at the walls and in the corners of reverberation chambers. For airborne sound sources, the Waterhouse [17] correction compensates for this systematic variation. For reception plates, Vogel et al. consider the edge effect, to find an equivalent correction factor [18,19]. Since the plate velocity is measured at only a limited number of response positions, the selection of appropriate positions assumes importance. Späh and Gibbs investigate appropriate sampling strategies [20].

The total loss factor of the reception plate can be determined by measuring the structural reverberation time – the procedure is similar to that for measuring the reverberation time in rooms, but the energy decays are generally shorter. On an isolated plate, the total loss factor equals the internal loss factor, as the coupling losses and radiation losses are assumed negligible. The energy decay curve has a single gradient and estimation of the loss factor is straightforward.

3. Connected reception plates

If the reception plate is connected to other plates, such as floors bonded into walls, part of the injected source power is lost to the other plates. Fig. 1, right, illustrates a reception plate with a single connected plate as a SEA model with two subsystems. The power balance equations of the coupled plate system as shown in Fig. 1 are

$$P_{in,1} = \omega\eta_{11}E_1' + \omega\eta_{12}E_1' - \omega\eta_{21}E_2 \quad (6)$$

$$0 = \omega\eta_{22}E_2 + \omega\eta_{21}E_2 - \omega\eta_{12}E_1'. \quad (7)$$

The first terms on the right-hand side in Eqs. (6) and (7) describe the internal losses. The second terms describes the energy lost to the other subsystem. The third term represents the power returning from the other subsystem. For a visual representation of these power flows, see Fig. 1. To obtain the injected source power, the energy in both subsystems must be known as well as the internal loss factor and the coupling loss factors. Using only the internal loss factor and energy of subsystem 1 will give an incorrect estimate of the transmitted source power. Hopkins and Robinson [21] found that vibration levels can increase, due to returning energy from connected building elements. A typical energy decay curve of a free plate measurement is a straight line with a single gradient (on a log-linear scale). For a connected element, the energy decay curve typically shows a changing gradient. Fig. 2 shows the idealized energy decay curves of a free plate and of the same plate connected to a second plate, calculated using Transient SEA [21].

Single gradient fits of such energy decay curves result in an overestimate of the structural reverberation time and consequently in an under-estimate of the loss factor and the source power. However, when combined with the overestimate of energy, due to the returning energy component, Hopkins and Robinson show that the two effects can partly compensate each other, but this depends on the building situation.

A further complication is in estimating the mass of real building elements, where it is not obvious how much of the support structure should be included. In addition, real building elements may have a composite nature. The modal behaviour of coupled plates differs from that of free plates and the sampling strategy therefore must be modified. Using connected walls and floors as reception plates can incur significant errors and alternative approaches are required.

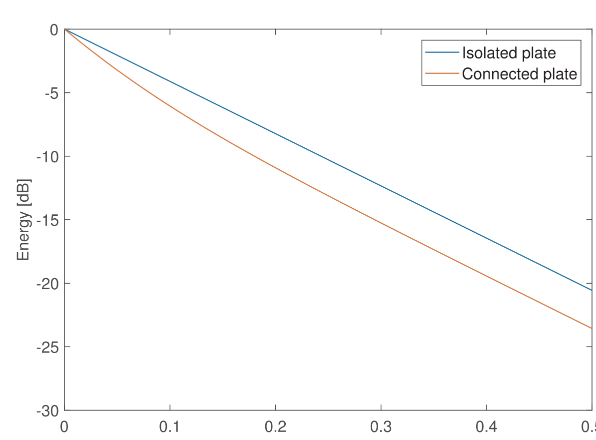

Fig. 2. Idealized energy decay curves for isolated and connected plate.