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Research Article

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Experimental Study on the Damping Loss Factor of a Plate Cavity Structure with a Complex Boundary

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Abstract

In acoustic–vibration coupling modeling and analysis, the damping loss factor (DLF) and coupling loss factor (CLF) are important parameters that can measure the damping characteristics of a system and determine its vibration energy dissipation capacity. Their accuracy directly affects the accuracy and reliability of model prediction. To improve the calculation accuracy and efficiency of the system loss factor, this study proposed a method for calculating the loss factor and CLF in a subsystem based on experimental measurement data. First, the quantitative relationship among the total loss factor, DLF, and CLF of a subsystem was deduced and established on the basis of an energy balance equation. Then, the total loss factor of the subsystem and the expressions between the average vibration energy and vibration signal under single-frequency excitation were derived in accordance with mechanical impedance theory. Second, the spectrum function of multipoint response under single-frequency excitation was measured with an experimental identification method, and the total loss factor of each subsystem of the coupling model and the CLF were calculated using the derived formula. Results show that the proposed method cannot only fully utilize the advantages of the transient attenuation and mean admittance methods, but can also avoid the error of response signal changing with time under steady-state excitation and reflect on the loss characteristics of a subsystem more comprehensively. This study provides a good reference for improving the prediction accuracy of the acoustic–vibration coupling of complex structures and guiding structural acoustic design.

Keywords: Plate cavity structure, Damping loss factor, Coupling loss factor, Statistical energy method

1. Introduction

In the modeling analysis of complex acoustic–solid coupling systems, the damping loss factor (DLF) and coupling loss factor (CLF) are important parameters, and their accuracy cannot be disregarded. DLF reflects the damping characteristics of subsystems, while CLF is the only important parameter that characterizes energy exchange between coupled systems in statistical energy analysis (SEA) [1]. However, for complex large-scale composite structures, such as high-speed rail bodies, spacecraft, and ships, the use of theoretical calculation methods to obtain parameters, such as loss factor, is unrealistic because of their large size and relatively complex structure. The finite element (FE) analysis method requires a relatively accurate structural model, and relevant parameters are obtained through experiments to verify its accuracy. Therefore, the use of FE analysis in a wide frequency range considerably increases calculation cost, and it is also time-consuming and labor-intensive. Given these issues, scholars have conducted numerous studies to calculate the loss factors of complex coupled systems [2-3].

Existing commonly used experimental methods for measuring structural loss factors (including DLF and CLF) can be classified into two categories: steady-state energy flow and transient attenuation methods [2]. The former uses continuous and stable broadband random excitation to estimate DLF. This method is required to obtain accurate input power of a structure, inevitably leading to measurement errors. The latter calculates the average loss factor of a structure within any frequency range by using the energy attenuation envelope of the response signal in accordance with the attenuation characteristics of the free vibration signal. However, this method is required to separate subsystems. When the subsystems of complex structures are assembled and inseparable, DLF cannot be obtained directly. In addition, data selection and segmentation are frequently subjective when fitting the attenuation curve. For composite structure systems, the DLF of each substructure after separation can generally be determined using an empirical formula or experimental method. When substructures are connected, the DLF of each substructure changes due to the interaction of the joint boundary, damping, and connection technology. Such change can hardly be calculated and should be solved using field measurement, which is difficult to perform during the preliminary design stage of a product. In addition, the study of power flow analysis in classical SEA is limited to directly connected systems; that is, CLF refers to that between directly coupled systems. In practical engineering, direct and indirect couplings occur between structural subsystems under the influence of the near-field sound of vibrating structures; that is, indirect CLF exists between structures, and it cannot be disregarded in the transmission of structural vibration or sound [3,4]. Therefore, efficiently and accurately calculating the loss factor of complex coupling systems is an urgent problem.

The current study proposed a new method for measuring loss factor in accordance with the relationship among the total loss factor, DLF, and CLF of subsystems. DLF and CLF can

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be directly obtained by measuring the total loss factor of a subsystem and the energy ratio between subsystems. This method does not necessitate simplifying the conditions under the assumption of satisfying the average energy distribution of subsystems. The calculation results can reflect the loss characteristics of subsystems more comprehensively.

2. State of the art

To date, scholars have conducted numerous studies on the calculation, measurement, and accuracy of loss factor, involving theoretical calculations or methods combined with experimental tests to obtain parameters [2,3].

Sun [5] proposed a theoretical calculation method and an experimental measurement method for indirect CLF and clarified the relationship between indirect CLF and DLF, and input admittance and transfer admittance. However, difficulties are experienced in solving large numerical matrices in practical applications because of the ill-conditioned matrix problem. In accordance with the attenuation characteristics of free vibration signals, Li et al. comprehensively applied transient envelope technology and an attenuation method. They obtained the envelope function of the response signal via Hilbert transform and measured the frequency average loss factor of the structure within any frequency range [6]. However, multiple modes frequently exist in a specific analysis bandwidth, making the envelope amplitude curve of the response signal complicated and causing difficulties in determining the attenuation rate. Oliazadeh et al. proposed to calculate the structural loss factor by measuring the sound intensity and sound pressure of a sound field generated by a plate under acoustic excitation; however, this method introduces additional mass when measuring the surface vibration acceleration of the plate [1]. Fahy and Ruivo presented an input power adjustment method for overcoming the shortcomings of the steady-state energy method; their method reduced workload and measurement error [7]. However, this method imposes high requirements for conditions and its practical application is uneconomical, limiting its popularization and application. Gu reported that the loss factor of a steady-state excitation system mostly depends on the mode with fast vibration attenuation, and the components with rapid attenuation at the initial time must be fully considered when using an attenuation method to test the loss factor [8]. Wang et al. calculated CLF between a flat plate and an elastic periodically stiffened plate by using a wave propagation method; they revealed the origin of the CLF between simple structures, but they disregarded the influence of stiffener width on the coordination conditions of the adjacent elements of the periodically stiffened plate [9]. Wang et al. calculated the loss factor of damping materials through structural modal analysis under pulse excitation; they discussed the factors that influenced test accuracy, proving that the measured results can be used to evaluate the performance of damping at room temperature [10]. Oliazadeh et al. estimated the DLF of a honeycomb sandwich plate and calculated input power by using the formulas of excitation force and honeycomb modal density [11]. Moreover, the influence of the additional mass of the impedance head on the excitation measurement force was studied, and the correction factor formula was derived, improving the estimation accuracy of DLF. Lin et al. obtained the DLF of viscoelastic materials via least-squares curve fitting by measuring the complex frequency response function of viscoelastic materials, but they did not consider the nonconservative coupling loss effect between subsystems [12]. Chen et al. [13] proposed an FE power input method for calculating the CLF of complex structures in a thermal environment; this method exhibits the problem of poor stability. Liu [14] presented an analytical power input method based on a numerical method. This method uses input point admittance and transfer admittance to calculate DLF, but its calculation results directly depend on the accuracy of the FE model. Treszkai [15] measured the DLF of a micro damping plate by using the half-power bandwidth method. The results demonstrated that the measured DLF was considerably influenced by frequency response quality. In accordance with power input method theory, Treszkai [16] experimentally measured the DLF and CLF of 19 combined structures with different connection modes and then compared them with the simulation results of a statistical energy method. The conclusion showed that the experimental results exhibited good agreement with the simulation results. The two aforementioned methods only analyze the middle frequency band. Bhagwan et al. [17] estimated the CLF of a composite structure by using an energy-level difference method. They studied the influence law of different tightening torques of bolted connection on the coupling factor of a composite structure. Their experimental results demonstrated that CLF increased with an increase in torque. Borgaonkar [18] performed experiments and numerical calculations on the modal density and DLF of rectangular plates composed of different materials and analyzed the influence of boundary conditions, such as free, simply supported, and cantilever, on CLF, but the study model was too idealistic. Nieradka et al. [19] corrected the negative value of CLF in experimental measurements by using the Monte Carlo filtering method, and the result was further verified through experiments.

Although the attenuation and power input methods have been successfully applied to the experimental identification of the DLF of some structures, and occasionally, consistent results have been obtained, they have their limited scope of applications. On the basis of the original energy balance equation, the current study derived the relational expressions of the total loss factor, DLF, and CLF of a subsystem. The total loss factor of a subsystem and the expressions between the average vibration energy and the vibration signal under single-frequency excitation were derived in accordance with mechanical impedance theory. The spectrum function of multipoint response under single-frequency excitation was measured with an experimental identification method, and the total loss factor of each subsystem of the coupling model and CLF were calculated using the derived formula. This study provides a reference for improving the prediction accuracy of the acoustic–vibration characteristics of complex coupling.

The remainder of this paper is structured as follows. Section 3 describes the plate cavity structure model with different boundaries, deduces the theoretical principle of the experimental measurement, and constructs the experimental scheme. In Section 4, the experimental data are calculated and analyzed using the derived theoretical formula, and the damping parameters and characteristics of different subsystems are obtained. Finally, Section 5 summarizes this study and draws the conclusion.

3. Methodology

3.1 Physical model

As shown in Fig. 1, the model is a closed cavity structure surrounded by six elastic plates with varying stiffness. The top plate (Subsystem 1) is made of steel, while the five other plates are made of wood with the same properties. The steel plate is connected to the wooden wallboard through a set of springs, while the wooden wallboards are fixedly connected, and the joints are sealed with viscoelastic materials. The cavity size is $L_x \times L_y \times L_z = 880 \times 1135 \times 715$ mm . The material properties of each subsystem are provided in Table 1.

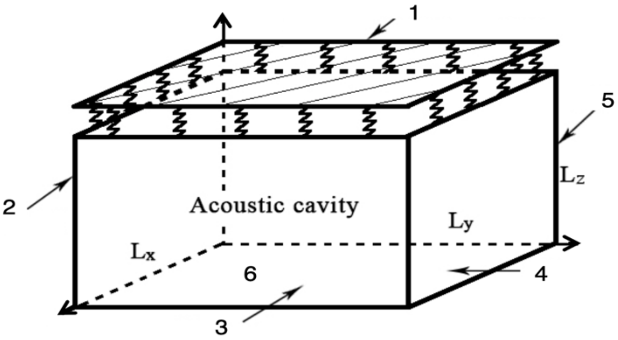


Fig. 1. Structural model of plate cavity with different boundaries

Table 1. Subsystem material properties

Materials	Thickness (mm)	Density (kg/m^3)	Elastic modulus (Pa)	Poisson's ratio
Steel plate (Subsystem 1)	1.5	7800	2.1E+11	0.3125
Plywood (Subsystems 2–6)	17	700	6E+9	0.25

3.2 Experimental theory principle

In accordance with the definition of loss factor [2], its size is equal to the ratio of unit circle frequency input power to the energy of the excited subsystem, i.e.,

$$\eta_s = \frac{P_{in}}{\omega E}, \tag{1}$$

where η_s is the total loss factor of the structure, which consists of DLF and CLF, $\omega = 2\pi f$.

When the structure reaches steady-state vibration under random excitation with a stable connection, the input power is equal to the loss power:

$$E = M \cdot \overline{\langle v^2(t) \rangle}, P_{in} = \langle f(t) \cdot v_0(t) \rangle, \tag{2}$$

where $f(t)$ is the input point excitation force, $v_0(t)$ is the response speed of the subsystem, and $\overline{\langle\ \rangle}$ is the average time and space.

Vibration source is frequently simplified as a point source in practical engineering. When the system structure is excited by a single-frequency source, the input power of the point source to any receiving system can be derived by using mechanical impedance theory.

Assuming that excitation is an ideal force source, then the input power is:

$$P_{in} = \frac{1}{2}|F(\omega)|^2 \operatorname{Re}[Y_R(\omega)], \tag{3}$$

where $Y_R(\omega)$ is the velocity admittance at the input point.

By using the expression

$$H_v(\omega) = \frac{2}{j\omega} H_a(\omega), \tag{4}$$

the input power of the structure can be expressed as follows:

$$P_{in} = \frac{1}{2}|F(f)|^2 \operatorname{Re}[H_{v0}(f)] = \frac{1}{2}|F(f)|^2 \frac{\operatorname{Im}[H_{a0}(f)]}{2\pi f} \tag{5}$$

$$E = M \cdot \overline{\langle v^2(t) \rangle} = \frac{1}{2} M \cdot |F(f)|^2 \frac{\sum_{i=1}^{N} \left|H_a^{(i)}(f)\right|^2}{N(2\pi f)^2}, \tag{6}$$

where $H_{v0}(f)$ is the origin frequency response function of the velocity, $H_{a0}(f)$ is the origin frequency response function of the acceleration, $H_a^{(i)}(f)$ corresponds to the acceleration frequency response function of the ith measuring point, and N is the measuring point.

Formulas (5) and (6) are substituted into Formula (1):

$$\eta_s = \frac{N \cdot \operatorname{Im}[H_{a0}(f)]}{M \cdot \sum_{i=1}^{N} \left|H_a^{(i)}(f)\right|^2}, \tag{7}$$

Therefore, the total loss factor of the subsystem can only be obtained by measuring the acceleration frequency response function of an excitation point of the substructure to multiple measuring points under single-frequency excitation with structure quality M .

In accordance with classical SEA theory [4], when a subsystem is excited by external force, the subsystem will produce energy loss, and the remaining energy will be transferred to the adjacent subsystems in the form of energy. If the system is weakly coupled and only subsystem *n* is excited, then the following power flow balance equation is obtained:

$$\omega E_n \eta_n + \sum_{j=1, j\neq n}^{K} \omega E_n \eta_{nj} - \sum_{j=1, j\neq n}^{K} \omega E_j \eta_{jn} = P_n,\ n = 1,2,\cdots K , \tag{8}$$

$$\omega E_i \eta_i + \sum_{j=1, j\neq i}^{K} \omega E_i \eta_{ij} - \sum_{j=1, j\neq i}^{K} \omega E_j \eta_{ji} = 0,\ i = 1,2,\cdots K, i \neq n , \tag{9}$$

where η_n is the DLF of subsystem n , η_{ij} is the CLF between i and j subsystems, ω_n denotes one-third frequency doubling of the center frequency, E_i is the space–time average total energy of subsystem i , P_n is average input power to subsystem n for excitation, and K is the number of subsystems.

Formula (9) is a system of equations that consists of $(K-1)$ equations. Combined with Formula (1), the variants of Formulas (8) and (9) are as follows:

$$\eta_n + \sum_{j=1, j\neq n}^{K} \eta_{nj} - \sum_{j=1, j\neq n}^{K} E_{jn} \eta_{jn} = \eta_{sn},\ n = 1,2,\cdots K , \tag{10}$$

$$E_{in}\eta_i + \sum_{j=1, j\neq i}^{K} E_{in}\eta_{ij} - \sum_{j=1, j\neq i}^{K} E_{jn}\eta_{ji} = 0,\ i = 1,2,\cdots K, i \neq n, \quad (11)$$

where $E_{in} = E_i / E_n$ and $E_{jn} = E_j / E_n$.

The research object is the composite structure of six subsystems, and Formulas (10) and (11) can be rewritten into matrix equations as follows:

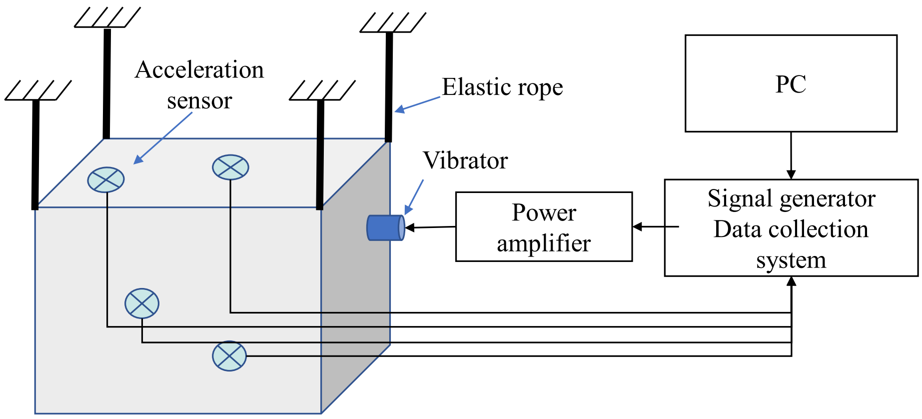
$$\mathbf{H}\varepsilon = \varepsilon_s, \quad (12)$$

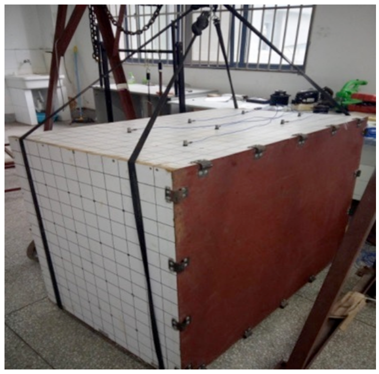
With six subsystems as examples, matrix H is a square matrix with the corresponding energy ratios as elements.

As indicated in Formulas (10) and (11), the DLF of each subsystem and the CLF of each subsystem can be obtained as long as the total loss factor of each system and the energy ratio between subsystems are measured.

3.3 Experimental identification

The energy ratio method was used in this experiment, and the plate was not constrained. The vibration exciter was applied for multipoint excitation input, and the statistically significant input power and energy of each subsystem were obtained by calculating the average of multiple measuring points. As shown in Fig. 2, the experimental object was suspended by an elastic soft rope during measurement to meet the free boundary conditions as much as possible and reduce the influence of boundary conditions on energy transfer. To improve the accuracy of the experiment result, the average energy of the subsystem can obtain sufficient speed response information by increasing the excitation points and reducing the response points. In turn, the excitation points were excited by the vibration exciter to obtain adequate vibration response information and more accurate average vibration energy of the subsystem. In accordance with the experimental principle, measurement was completed in two steps. First, the total loss factor of each subsystem was measured. Second, the energy ratio between subsystems was measured.





Experimental model

Fig. 2. Experimental scheme

When measuring the total loss factor, each subsystem was equipped with one excitation point and nine response measuring points in accordance with the structure of the acoustic cavity, as shown in Fig. 3. The measuring point closest to the excitation point was defined as the origin response, and the acceleration frequency response functions of each measuring point that corresponded to the origin excitation were recorded. The total loss factor of the subsystem was calculated by substituting the imaginary part of the origin frequency response function and the amplitudes of all the frequency response functions into Formula (7).

In measuring the average vibration energy of a subsystem, this study used the multi-excitation point input method to obtain the average vibration energy of each subsystem. As shown in Fig. 4, each subsystem is equipped with 12 pulse excitation points and 1 response measuring point.

The excitation and vibration directions of a subsystem are normal to the plane of the plate structure. After the excitation of a subsystem was completed, each subsystem received a response, and six equations were obtained after each excitation. In accordance with Formula (6), the average energy of the six subsystems can be obtained; that is, the energy ratio relationship between each excitation surface and receiving surface can be determined. In accordance with the measured total loss factor of each subsystem and the energy ratio of each subsystem, 36 unknowns, including 30 CLFs and 6 DLFs, can be obtained by solving Formula (12).

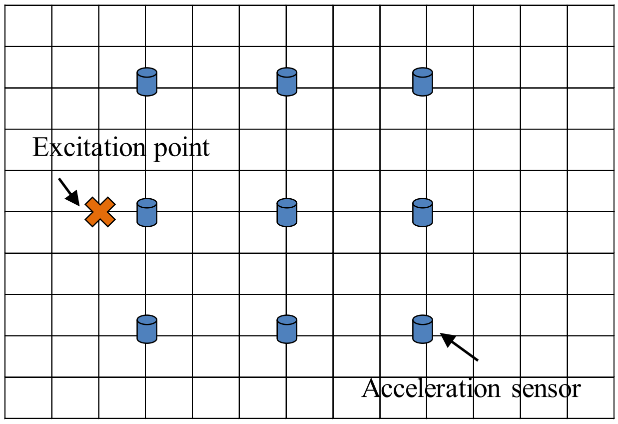


Fig.3. Total loss factor measuring point arrangement of a subsystem

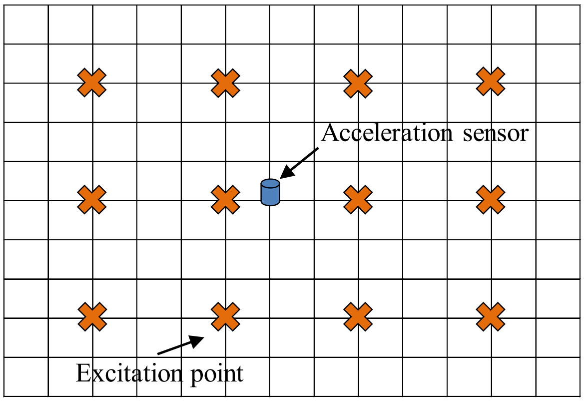


Fig.4. Average vibration energy measuring point arrangement of a subsystem

4. Result Analysis and Discussion

The DLF η_i and CLF η_{ij} of one-third of the frequency of each subsystem are shown in Figs. 5 and 6. In accordance with the loss factor curve in a subsystem, except for the elastic plate, the loss factors of the wallboards composed of composite plates are similar, which gradually decrease with a change in frequency. The first few overall bending modes of the box body mostly appear within 200 Hz; hence, overall resonance occurs in the frequency band when an external force is excited, and the energy dissipation of the structure is large. In addition, considering that the system has a symmetrical structure, the DLFs of side plates 2 and 4 (η_2,η_4), 3 and 5 (η_3,η_5), and the changing trend are similar. The values are generally above 0.01. The loss factor η_1 of elastic plate 1 also gradually decreases with an increase in frequency, and its value is above 0.0002. Under mutual coupling between the structure and the acoustic cavity, the DLFs of the rigid and elastic subsystems are larger than the structural loss factor of the material (the structural loss factor of the composite plate is 10^{-2}, while the structural loss factor of the steel is 10^{-4}). The connection boundary of the composite structure is complex and the reverberation effect of the closed acoustic cavity is evident; hence, the acoustic radiation loss and boundary loss effect of the subsystem are large, leading to a large DLF. A conclusion can be drawn that the calculation result of the DLF test method based on single-frequency excitation is close to the real value, which is in line with the measurement experience of the statistical energy method in engineering applications.

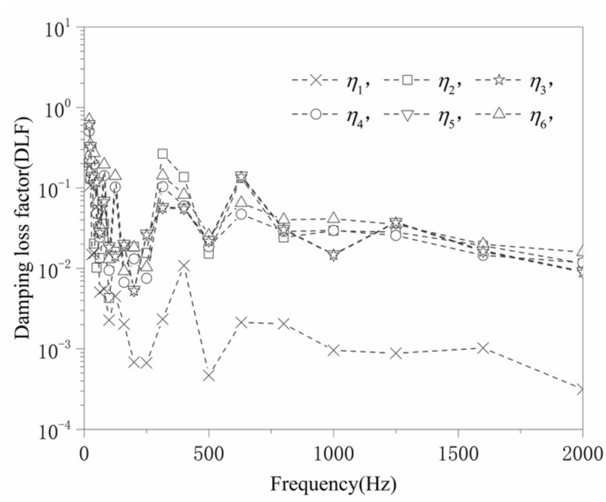
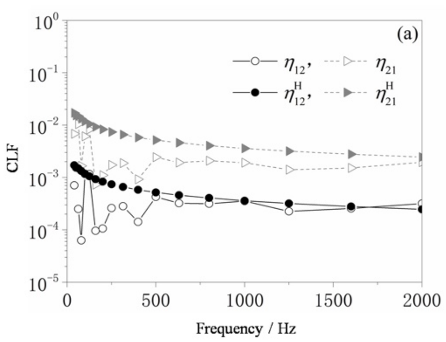
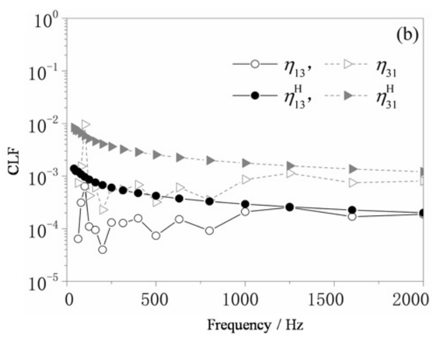


Fig.5. DLFs of each subsystem

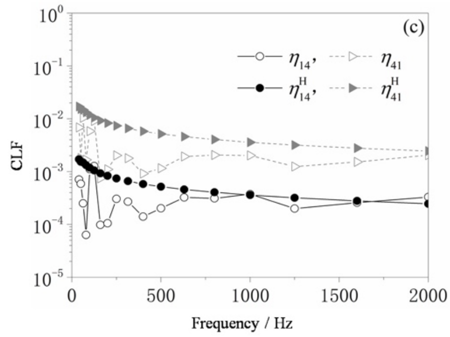
As shown in Fig. 6, coupling between the subsystems of the cavity structure is unequal, i.e., $\eta_{ij} \neq \eta_{ji}$. η_{ij} indicates that the energy transfer direction is from subsystem *i* to subsystem *j*. From the graphic results, mutual energy transfer between subsystems is irreversible. In particular, the mutual coupling loss difference between elastic plate 1 and the rigid side plates is more significant, indicating that the cavity structure is a nonconservative coupling system. Energy dissipation units, such as damping, exist at the junction of subsystems. As illustrated in Figs. 6(a)–6(d), when two substructures with different dynamic characteristics are coupled through elastic boundaries, energy loss transmitted from the subsystem with greater elasticity to the subsystem with less elasticity is considerably less than that transmitted from the subsystem with greater elasticity by about one order of magnitude.



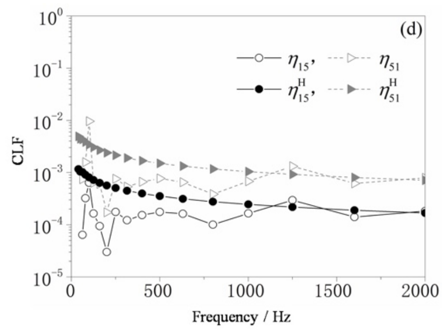
(a)



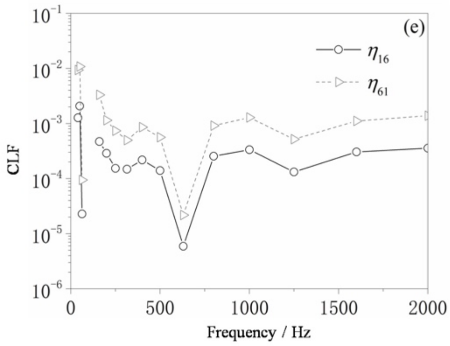
(b)


(c)
η_{14}, η_{41}
η_{14}^{H}, η_{41}^{H}
CLF
Frequency / Hz

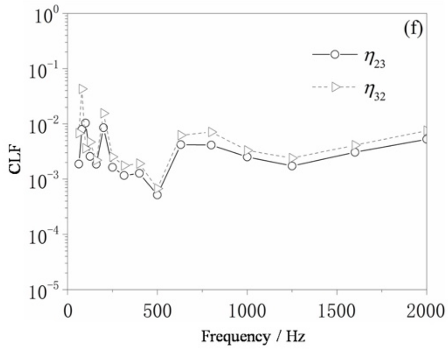
(c)


(d)
η_{15}, η_{51}
η_{15}^{H}, η_{51}^{H}
CLF
Frequency / Hz

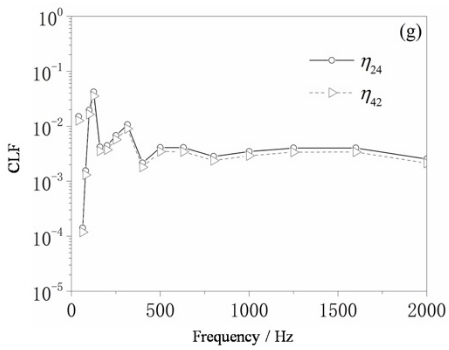
(d)


(e)
η_{16}
η_{61}
CLF
Frequency / Hz

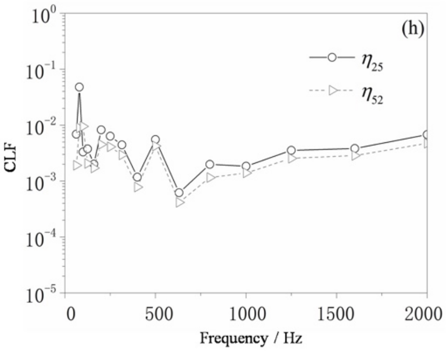
(e)


(f)
η_{23}
η_{32}
CLF
Frequency / Hz

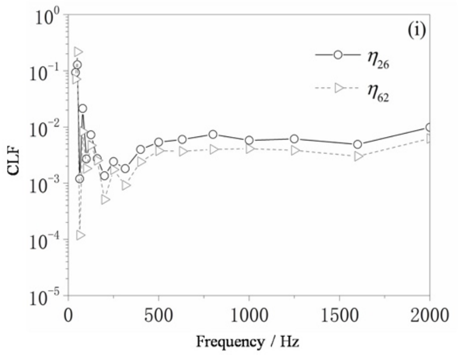
(f)


(g)
η_{24}
η_{42}
CLF
Frequency / Hz

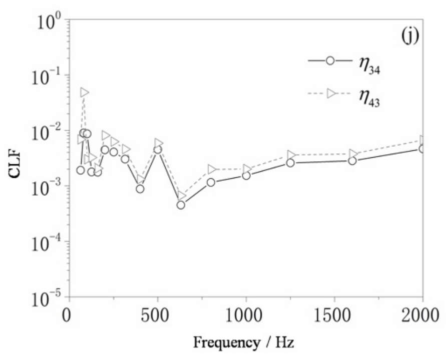
(g)


(h)
η_{25}
η_{52}
CLF
Frequency / Hz

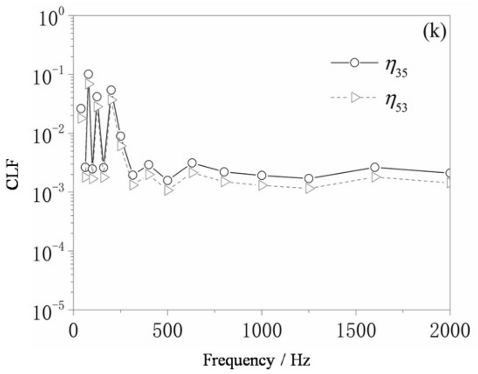
(h)


(i)
η_{26}
η_{62}
CLF
Frequency / Hz

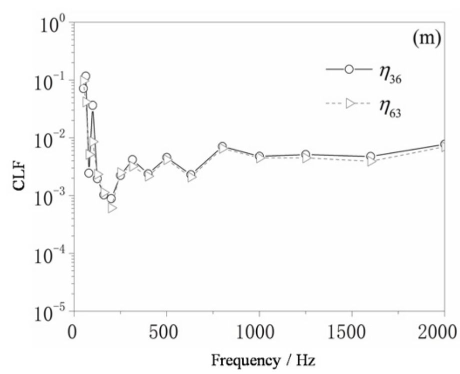
(i)


(j)
η_{34}
η_{43}
CLF
Frequency / Hz

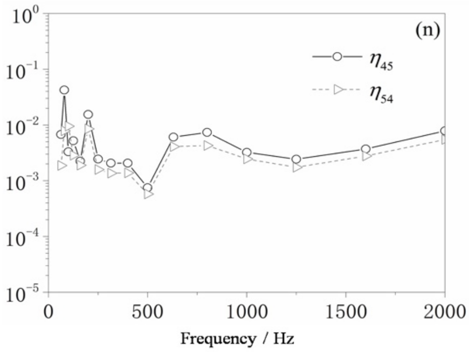
(j)


(k)
η_{35}
η_{53}
CLF
Frequency / Hz

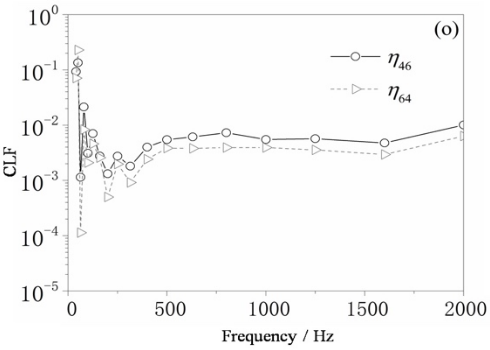
(k)


(m)
η_{36}
η_{63}
CLF
Frequency / Hz

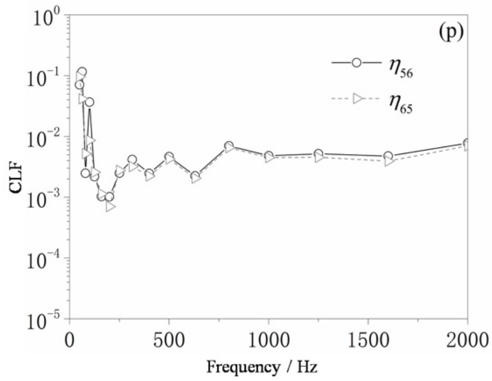
(m)



(n)



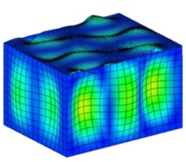
(o)



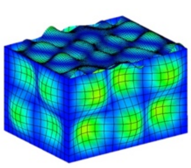
(p)

Fig.6. CLFs between subsystems: (a) subsystems 1 and 2. (b) subsystems 1 and 3. (c) subsystems 1 and 4. (d) subsystems 1 and 5. (e) subsystems 1 and 6. (f) subsystems 2 and 3. (g) subsystems 2 and 4. (h) subsystems 2 and 5. (i) subsystems 2 and 6. (j) subsystems 3 and 4. (k) subsystems 3 and 5. (m) subsystems 3 and 6. (n) subsystems 4 and 5. (o) subsystems 4 and 6. (p) subsystems 5 and 6.

The experimental study shows that the overall mode of the system exerts a considerable influence on the CLF between subsystems. For example, the peak values calculated by the experimental method at 125 Hz and 500 Hz are influenced by the overall bending mode. The overall modal shapes of the cavity structure at 125 Hz and 500 Hz extracted via FE analysis are shown in Fig. 7, and they clearly correspond to the peaks at 100 Hz and 500 Hz in the experimental test curve.



(a)



(b)

Fig.7. Overall modal shapes of the cavity structure: (a) mode shape of one-third of the frequency of the doubling center of 125 Hz, and (b) mode shape of one-third of the frequency of the doubling center of 500 Hz

Evidently, the proposed method can reflect the loss characteristics of a subsystem more comprehensively and is closer to the real value. The calculation of the loss factor of the acoustic cavity structure with complex boundary conditions is in line with the measurement experience in engineering applications.

5. Conclusions

To solve the difficulty in obtaining the loss factor of acoustic cavity structure subsystems with different boundary conditions, this study derived the relationship among the total loss factor, DLF, and CLF of a subsystem on the basis of a system energy balance equation and performed an experimental identification study based on theoretical calculation. The following conclusions can be drawn.

(1) Influenced by the complex boundary and reverberation of a closed acoustic cavity, a large acoustic radiation loss and a boundary loss effect occur, and the measured total loss factor of a subsystem is larger than the inherent loss factor of the material itself.

(2) The peak value of the CLF of a subsystem is determined by the natural frequency of the rigid body, and it gradually decreases with a change in frequency.

(3) The proposed method is not required to simplify the conditions, and the calculation results can reflect the loss characteristics of a subsystem more comprehensively. These conditions are in line with the measurement experience in engineering applications.

This study innovates by combining theoretical study and experimental identification on the basis of a steady-state energy method. The proposed method can provide a reference for the study of low-frequency vibroacoustic characteristics of complex structures in the future.

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