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Research Article

Modelling and Analysis of Global Vibroacoustic Coupling Characteristics of a Rectangular Enclosure Bounded by a Flexible Panel

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The interaction between the sound field in an enclosure and its flexible panel is a critical problem; the influence of structural parameters on global vibroacoustic coupling characteristics of the panel-enclosure system is very important. In this paper, a novel index of the global coupling level was first proposed to describe the global vibroacoustic coupling extent between multiple panel vibration modes and enclosure acoustical modes of the coupled system. Then, the influence of structural parameters on global coupling levels of the coupled system with different panel boundary conditions was obtained based on the numerical results of transfer factors. Moreover, according to the comprehensive influence of the structural parameters on global coupling levels, design methods of the coupled system with low global coupling levels were then discussed. Finally, the influence mechanism of panel boundary conditions on the coupling characteristics of the coupled system was analyzed. The results show that the structural parameters have similar effect on the coupling property of the system with different panel boundary conditions. Furthermore, the influence of the structural parameters on the coupling property of the system with a clamped panel is more sensitive than that of the system with a simply supported one. Furthermore, the structural parameters, especially the enclosure depth and panel thickness, are not completely negative correlated to the global coupling levels of the system. In conclusion, this study could provide a theoretical basis for acoustical design of the panel-enclosure system (e.g., rectangular vehicle cabins) with low global coupling level, as well as the lightweight structure design of the system.

1. Introduction

The problem of acoustic radiation caused by the vibration of the panel structures has been an important research topic in the field of acoustical engineering for a long time. The interaction between the sound field in an enclosure and its flexible boundary is a critical problem. As is known that the elastic panel structures are widely used in the practical engineering fields, such as vehicles, ships, aircraft cabins, and so on. However, in recent year, in the lightweight development process, the low-frequency noise (20–200 Hz), such as the booming noise generated by the vehicle cabin structure-acoustic coupling, has become one of the most important factors contributing to the comfort level of the occupants of a passenger car. Therefore, a good understanding of structure-acoustics coupling is particularly important to control the sound field in an enclosure.

In the past decades, considerable efforts have been devoted to the study of this classical model. Dowell et al. [1, 2] carried out some of the early research into modelling the vibrations of panel packed by a rectangular enclosure with five absorptive walls, and then he established the acoustic-structural coupling model on the sound field in an enclosure with a simply supported panel using the classical modal coupling method. Later, Pan and Bies [3] analyzed the effect of the interaction between the boundary condition of the flexible wall and the sound field in the enclosure on sound field in enclosure. Besides, in the work of Pan and Bies [3], the effect of panel modal density (corresponding to the panel thickness) variations on the resonance frequency of acoustical modes and modal decay time of the coupled system was analyzed. However, the influence of enclosure depth and absorptive walls of enclosure on the coupling degree was not considered in this work. Based on the work of Pan et al. [3, 4], Wang et al. [5] carried out a study on the influences of enclosure depth on resonance frequencies and modal decay time of the coupled system with a simply supported panel. It is known that the acoustic-structure coupling in panel-enclosure system which consists of an enclosure with a clamped flexible wall is different from that of a simply supported one. In the work of Sung and Jan [6] and Arenas [7], the vibration response and sound radiation by a clamped panel were obtained using the virtual work principle. In the current work, the mode shape function proposed by Sung and Jan [6] and Arenas [7] was used to analyze the characteristics of the panel-enclosure system with a clamped flexible wall. As an extension of the work carried out by Pan and Bies [3] and Sung and Jan [6], Wang et al. [8] analyzed the effect of different variables, including panel thickness, panel internal damping, and enclosure depth, on the coupling features of the coupled system, which include resonance frequency and modal decay time. Recently, large amount work has been carried out to study the forced response of the coupled system with different kinds of excitation [9–12] and active noise control [12–16], based on the analysis of vibroacoustic coupling. However, the existing literatures are concentrated on the coupling property between a single panel mode and an enclosure mode, and the influence of the structural parameters on the global coupling degree between multiple panel modes and multiple enclosure modes has not been considered. Therefore, it is difficult to globally evaluate the effect of structural parameters on coupling property between multiple panel modes and multiple enclosure modes, and it is impossible to choose the most suitable structural parameters that the global coupling degree of the coupled system is at the lowest level.

This paper presents a parametric study on the influence of structural parameters on the global vibroacoustic coupling characteristics of a rectangular enclosure bounded by a flexible panel with different boundary conditions. The rest of this paper is organized as follows. Section 2 introduces the analytic model of the panel-enclosure system with different panel boundary conditions. Moreover, the definition of global coupling level (GCL) is proposed to describe the coupling degree between multiple panel modes and multiple enclosure acoustical modes. Section 3 introduces the method used in this paper. In Section 4, the influence of structural parameters, as well as the panel boundary conditions, on the global coupling levels of the system is analyzed. Additionally, according to the comprehensive effect of structural parameters on global coupling level, design methods of the system with low global vibroacoustic coupling levels are then discussed. Furthermore, the influence mechanism of the panel boundary conditions on the coupling characteristics is also analyzed. Finally, this paper concludes with a discussion of future research in Section 5.

2. Analytical Model

The rectangular panel-enclosure system is shown in Figure 1. The enclosure has five small absorptive walls and one flexible wall; the dimensions of the enclosure are L_x, L_y, and L_z. The flexible panel is fixed along its edges at $z = L_z$. The sound pressure p in the enclosure can be described by an inhomogeneous wave equation:

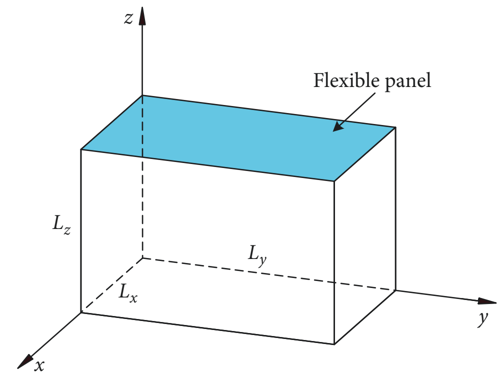


Figure 1: Panel-enclosure geometry and coordinate system.

$$\nabla^2 p - \frac{1}{c_0^2}\frac{\partial^2 p}{\partial t^2} = 0, \tag{1}$$

where p is the interior sound pressure of the enclosure and c_0 is the sound speed in air.

At the five absorptive walls inside the enclosure and the flexible enclosure interface, the boundary conditions are taken as the following form:

$$\begin{cases} \dfrac{\partial p}{\partial n} = -\rho_0 \dfrac{\partial^2 w}{\partial t^2}, & z = L_z, \text{ at absorption wall,} \\[2ex] \dfrac{\partial p}{\partial n} = -j\dfrac{\rho_0 \omega}{Z_A} p, & z \neq L_z, \text{ at flexible wall,} \end{cases} \tag{2}$$

where n indicates the normal direction of the boundary surfaces. Z_A is the specific acoustic impedance on the absorptive walls inside the enclosure. $w(x, y, t)$ is the normal displacement of the flexible panel.

The governing equation of flexible vibration of thin isotropic panel due to the internal pressure p is expressed as follows:

$$\frac{Eh^3}{12(1-\mu^2)}\nabla^4 w + \rho h \frac{\partial^2 w}{\partial t^2} = p, \tag{3}$$

where ρ, E, μ, and h are panel density, Young's modulus, Poisson's ration, and panel thickness, respectively. In this paper, the panel radiation is neglected, and the positive direction for displacement of the flexible panel is towards outside of the enclosure.

The mode shape function of rectangular enclosure with rigid walls is given by

$$\phi_N(x, y, z) = \cos\left(\frac{l\pi x}{L_x}\right)\cos\left(\frac{m\pi y}{L_y}\right)\cos\left(\frac{n\pi z}{L_z}\right), \tag{4}$$

where l, m, and n are the indices of the enclosure acoustical mode.

For a simply supported panel, the mode shape function of uncoupled panel [4] is given by

$$s_M(x,y)=\sin\left(\frac{u\pi x}{L_x}\right)\sin\left(\frac{v\pi y}{L_y}\right), \tag{5}$$

where u and v are the mode indices of the panel vibration mode.

For a clamped panel, the mode shape function of uncoupled panel is given by Pan et al. [4] and Sung and Jan [6], as shown in Equation (6), and it has been validated of high accuracy by experiments:

$$s_M(x,y)=\psi_u(x)\psi_v(y), \tag{6}$$

where $\psi_u(x)$ and $\psi_v(y)$ can be expressed by

$$\begin{aligned}\psi_u(x)&=\gamma\left(\frac{\lambda_u x}{L_x}\right)-\frac{\gamma(\lambda_u)}{H(\lambda_u)}H\left(\frac{\lambda_u x}{L_x}\right),\\ \psi_v(y)&=\gamma\left(\frac{\lambda_v y}{L_y}\right)-\frac{\gamma(\lambda_v)}{H(\lambda_v)}H\left(\frac{\lambda_v y}{L_y}\right),\end{aligned} \tag{7}$$

where $\gamma(s)=\cosh(s)-\cos(s)$ and $H(s)=\sinh(s)-\sin(s)$ and λ_v and λ_u are determined by

$$\cosh(\lambda)\cos(\lambda)-1=0. \tag{8}$$

Modal coupling coefficient represents the spatial matching degree between enclosure and panel modes on the interacting surface of the enclosure sound field and flexible panel, as shown in Equation (9). It is the integral of panel mode and enclosure mode on the surface of panel. As seen from Equation (9), the modal coupling coefficient only depends on the geometry shape and boundary condition of panel and the mode shape of enclosure acoustic field [3, 17], and it has nothing to do with panel thickness, material properties, damping of the panel, and enclosure depth [3, 17]:

$$B_{N,M}=\frac{1}{A_f}\int_{A_f}s_M\phi_N\,dA. \tag{9}$$

According to Wang's work [8], for a panel-enclosure system bound by a simply supported panel, if the coupling coefficient between a panel mode and an enclosure mode is nonzero, the modal indices must meet the following condition:

$$\begin{cases} l+u & =\text{odd number},\\ m+v & =\text{odd number}.\end{cases} \tag{10}$$

Coupling coefficient determines whether there is a coupling between the panel mode and the enclosure acoustical mode, while the coupling degree between a panel mode and an enclosure mode is decided by transfer factor [3]. The transfer factor between the N^{th} enclosure mode and M^{th} panel mode is given by

$$F_{N,M}=\left[1+\left(\frac{\omega_{aN}-\omega_{pM}}{2}\right)^2\left(\frac{\rho h L_z\Lambda_N\Lambda_M}{\rho_0 c_0^2}\right)\left(\frac{1}{B_{N,M}^2}\right)\right]^{-1}, \tag{11}$$

where $\Lambda_N=(1/V_0)\int_{V_0}\phi_N^2\,dV$, $\Lambda_M=(1/A_f)\int_{A_f}s_M^2\,dA$.

As seen from Equations (12) and (13), when the modal coupling coefficient $B_{N,M}$ is zero, the transfer factor must be zero, too. In this case, there is no energy transfer between the N^{th} enclosure mode and the M^{th} panel mode. When the modal coupling coefficient is nonzero, the transfer factor relates to many factors, such as the difference between resonance frequencies of uncoupled panel and enclosure modes and the enclosure depth. The larger the transfer factor between a panel mode and an enclosure mode is, the greater the coupling strength between them is obtained. When the transfer factor is approximately equal to 1.0, the energy transfer between the N^{th} enclosure mode and the M^{th} panel mode is strongest.

For a panel-enclosure system shown in Figure 1, the resonance frequencies of the enclosure acoustical mode and the uncoupled simply supported panel mode are given by

$$f_{aN}=\frac{c_0}{2}\left[\left(\frac{l}{L_x}\right)^2+\left(\frac{m}{L_y}\right)^2+\left(\frac{n}{L_z}\right)^2\right]^{1/2}, \tag{12}$$

$$f_{pM}=\frac{\pi}{2}\sqrt{\frac{D}{\rho h}}\left[\left(\frac{u}{L_x}\right)^2+\left(\frac{v}{L_y}\right)^2\right], \tag{13}$$

where $D=Eh^3/12(1-\mu^2)$.

Unlike the resonance frequencies of a uncoupled simply supported panel mode shown in Equation (13), the resonance frequencies of the uncoupled clamped panel mode are given by Sung and Jan [6] and Arenas [7] as follows:

$$f_{pM}=\frac{1}{2\pi}\sqrt{\frac{D}{\rho h}}\left(\left(\frac{\lambda_u}{L_x}\right)^4+\left(\frac{\lambda_v}{L_y}\right)^4+2\left(\frac{\lambda_u\lambda_v}{L_xL_y}\right)^2\frac{\kappa_u\kappa_v}{\delta_u\delta_v}\right)^{1/2}, \tag{14}$$

where κ_u, κ_v, δ_u, and δ_v are given by

$$\begin{cases}\kappa_i=\frac{1}{4}\left(1+D_i^2\right)\sinh(2\lambda_i)-\frac{1}{2}D_i\cosh(2\lambda_i)+\frac{3}{2}D_i-\frac{1}{4}\left(1-D_i^2\right)\sin(2\lambda_i)-D_i\cos^2(\lambda_i)-D_i^2\lambda_i,\\ \delta_i=\frac{1}{4}\left(1+D_i^2\right)\sinh(2\lambda_i)+\sinh(\lambda_i)\left[2D_i\sin(\lambda_i)-\left(1-D_i^2\right)\cos(\lambda_i)\right]-\left(1+D_i^2\right)\sin(\lambda_i)\cosh(\lambda_i)\\ \quad+D_i\cos^2(\lambda_i)-\frac{1}{2}D_i\left[1+\cosh(2\lambda_i)\right]+\frac{1}{4}\left(1-D_i^2\right)\sin(2\lambda_i)+\lambda_i,\end{cases} \tag{15}$$

where $D_i = r(\lambda_i)/H(\lambda_i)$ and λ_i is determined by Equation (8).

According to the definition of modal coupling coefficient and transfer factor, the coupling property between only one panel mode and an enclosure mode is described. However, the coupling degree between each panel mode and the enclosure mode is different, which makes it difficult to globally evaluate the effect of structural parameters on coupling property between multiple panel modes and multiple enclosure modes. Hence, it is impossible to choose the most suitable structural parameters that the global coupling degree of the coupled system is at the lowest level.

To solve this problem, coupling level $L_{F_{N,M}}$ is defined to evaluate the coupling degree between one single panel mode (the M^{th} order mode) and an enclosure mode (the N^{th} order mode), as expressed in Equation (16). Then, the comprehensive couplings between multiple panel modes (the first U order modes) and multiple enclosure modes (the first V order modes) are defined as the global coupling level $G_{V,U}$, as shown in Equation (17). According to Equation (17), the global coupling levels are only determined by those strong couplings, such as those transfer factors that are equal to 1, while those weak couplings will not make much contribution to the indictor of the global coupling level. For instance, a 60 dB global coupling level (corresponding to the transfer factor of 1) is larger than the global coupling level (about 57 dB) resulting from the superposition of two 54 dB coupling levels (corresponding to the transfer factor of 0.5):

$$L_{F_{N,M}} = 20\log_{10}\left(\frac{F_{N,M}}{F_{\text{ref}}}\right), \tag{16}$$

$$G_{V,U} = 10\log_{10}\left(\sum_{i=1}^{U}\sum_{j=1}^{V}\left(10^{L_{F_{j,i}}/10}\right)\right). \tag{17}$$

In this study, the minimum transfer factor is assumed to be 0.001, and let F_{ref} equal to 0.001. According to Equation (16), when the transfer factor $F_{N,M}$ equals to 1, the coupling level between the N^{th} enclosure mode and the M^{th} panel mode is 60 dB. Therefore, 60 dB can be used as the dividing line to judge whether the global coupling level of the system is strong or not. Therefore, when the global coupling level is greater than 60 dB, the system is likely to have some strong couplings; otherwise, there cannot be any strong couplings of the system. Moreover, compared with the linear addition of the transfer factors, the global coupling levels obtained by Equation (17) show a wider range of weak coupling regions. Hence, it can provide more choices for structural parameter design of the panel-enclosure system with low global coupling levels.

3. Case Study Description

According to Equation (16), the global coupling level of the panel-enclosure system is calculated based on the transfer factor between a single panel mode and an enclosure mode. Therefore, the structural parameters that affect the transfer factor could also affect the global coupling characteristics of the coupled system. For a given panel-enclosure system, it is determined by panel-enclosure depth L_z, panel thickness h, panel aspect ratio α (in this paper, $\alpha = L_x/L_y$), and panel boundary conditions. However, the existing researches have carried out considerable studies on the effect of enclosure depth and panel thickness on the coupling properties of the system; how the panel aspect ratio and the panel boundary conditions affect the coupling properties of the system is still unknown.

In order to explore the influence of the variations of each structural parameter, including enclosure depth, panel thickness, panel aspect ratio, and panel boundary conditions, on the global coupling characteristics of the coupled system, four cases were designed using the control variable method, as shown in Table 1. In this work, the area of flexible panel (A_f) was assumed to be 1 m^2, and cases 1~3 present the study on the effect of each structural parameter on the global coupling levels of the system. Case 4 analyzes the comprehensive effect of structural parameters on the global coupling levels of the system. Additionally, this paper investigates the global coupling levels between the first 8 orders of panel modes and the first 20 orders of enclosure modes, and the panel modes and enclosure modes are shown in Table 2.

Besides, the panel material properties were taken as follows: the panel material is aluminum, density ρ = 2700 kg/m^3, Young's modulus E = 71 Gpa, and Poisson's ration μ = 0.33. In this paper, the transfer factors of the first eight orders of the panel modes and the first twenty orders of the enclosure modes were calculated using Matlab software. Furthermore, the panel mode (2,1) and panel mode (2,3) were taken to discuss the influence of structural parameters, as well as panel boundary conditions on the global coupling characteristics of the coupled system.

4. Results and Analysis

4.1. Effect of Enclosure Depth on Global Coupling Level of the Coupled System. According to the theory in Section 2, the transfer factor determines the coupling degree between a panel mode and an enclosure mode. According to Equation (12), when the enclosure mode index n is equal to zero, by adjusting the enclosure depth, the resonance frequencies of panel remain unchanged. Hence, the enclosure depth determines the transfer factors between panel and enclosure modes, and the transfer factors decrease with the increase of enclosure depth. When the enclosure mode index n is nonzero, the transfer factors are determined by resonance frequency differences between panel modes and enclosure acoustical modes.

Figure 2 shows the effect of enclosure depth on coupling property between panel mode (2,1) and each enclosure mode. As seen from Figure 2(a), the transfer factor between the simply supported panel (SSP) mode (2,1) and the enclosure mode $(1, *, 0)$ $(* \neq 0)$ decreases with the increase of enclosure depth, which is consistent with the theoretical

Table 1: Parameters' range for case study.

Cases	Panel area A_f (m^2)	Enclosure depth L_z (m) Min	Enclosure depth L_z (m) Max	Panel thickness h (mm) Min	Panel thickness h (mm) Max	Panel aspect ratio α Min	Panel aspect ratio α Max	Panel boundary condition
1	1.0	0.2	5.0	2.0		1.0		Simply supported/clamped
2		1.5		0.2	5.0	1.0		
3		1.5		2.0		0.2	5.0	
4		0.2	5.0	0.52	5.0	0.2	5.0	

Table 2: Panel modes and enclosure modes.

Panel mode (u, v)	Enclosure mode (l, m, n)					
(1,1)	(1,2)	(0,0,0)	(0,1,0)	(0,0,1)	(1,0,0)	(0,1,1)
(2,1)	(2,2)	(1,1,0)	(1,0,1)	(1,1,1)	(0,2,0)	(0,0,2)
(1,3)	(3,1)	(2,0,0)	(0,2,1)	(0,1,2)	(1,2,0)	(1,0,2)
(2,3)	(3,2)	(2,1,0)	(2,0,1)	(1,1,2)	(1,2,1)	(0,3,0)

analysis above. Besides, the transfer factors between the SSP mode (2,1) and enclosure mode $(0, 0, *)$ increase as the enclosure depth increases. In addition, when the enclosure depth reaches a certain value, the transfer factors are approximately equal to 1. In this case, there is a large energy transfer between enclosure mode and panel mode. Moreover, the transfer factors between SSP mode (2,1) and enclosure mode (1,2,1) increase first and then decrease as the enclosure depth increases, and no significant coupling occurs.

By comparing Figure 2(b) with 2(a), the enclosure depth variations have similar influence on the coupling property between the clamped panel (CP) mode (2,1) and each enclosure mode. Furthermore, it is noticed that the influence of structural parameters on the coupling property of the system with a clamped panel is more sensitive than that of the system with a simply supported one. For instance, the transfer factors between CP mode (2,1) and enclosure mode $(*, *, 0)$ decrease significantly as the enclosure depth increases. What is more, as the enclosure depth increases, the transfer factors between CP mode (2,1) and enclosure mode $(*, *, *)$ $(* \geq 0)$ increase first and then decrease, which makes the strong coupling appear in a certain range of enclosure depth.

Figure 3 shows the effect of enclosure depth on coupling property between panel mode (2,3) and each enclosure mode. As seen from Figure 3, the transfer factors between panel mode (2,3) and enclosure modes $(*, *, 0)$ decrease as the enclosure depth increases, and the transfer factors between panel mode (2,3) and other enclosure modes increase first and then decrease as the enclosure depth increases. In addition, it is observed that the influence of enclosure depth on the transfer factors between panel mode (2,3) and enclosure mode $(*, 0, *)$ $(* \neq 0)$ is more sensitive than that on the transfer factors between panel mode (2,3) and enclosure mode (1,2,1). Therefore, the effect of enclosure depth on the coupling characteristics of the panel-enclosure system can be divided into two categories: one is that the coupling degree decreases with the increase of enclosure depth; another one is that the coupling extent increases first and then decreases as the enclosure depth increases, and such situations are more prone to significant couplings.

Moreover, by comparing Figure 2 with 3, it is observed that the variations of enclosure depth affect the coupling property more significantly between high-order panel modes and enclosure modes than that between low-order panel modes and enclosure modes. Furthermore, the enclosure depth ranges become narrower when the significant coupling occurs between high-order panel modes and enclosure modes. What is more, by comparing the effects of enclosure depth on the coupling characteristics of the coupled system with different panel boundary conditions, it is observed that the influence of panel boundary conditions on the coupling characteristics of the coupled system is similar.

Figure 4 shows the effect of enclosure depth on the global coupling level of the panel-enclosure system with different panel boundary conditions. As seen from Figure 4, in the enclosure depth range of 0.200~2.250 m, the global coupling level of the system with a clamped panel is greater than that of the system with a simply supported one, while in other range of enclosure depth, the global coupling level of the coupled system with a clamped panel is smaller. In addition, the maximum global coupling levels of the coupled system with different panel boundary conditions occur at the enclosure depth of 0.900 m and 1.100 m, respectively. Moreover, it is noticed that when the enclosure depth $L_z > 0.900$ m, the global coupling level of the system with a clamped panel decreases significantly as the enclosure depth increases. However, when the enclosure depth $L_z > 1.100$ m, the variations of enclosure depth weakly affect the global coupling level of the system with a simply supported panel. From a global perspective, compared to the panel-enclosure system with a simply supported panel, the global coupling level versus enclosure depth curve of the system with a clamped panel tends to move and compress towards the direction of smaller enclosure depth.

4.2. Effect of Panel Thickness on Global Coupling Level of the Coupled System. According to resonance frequencies equations, shown in Equations (13) and (14), of the simply supported panel and clamped panel, the variations of panel thickness will affect the resonance frequencies of panel modes, which thereby affect the frequency differences between the panel mode and enclosure mode. Additionally, according to Equation (11), the variations of panel thickness will also affect the transfer factor between the panel mode and enclosure mode. Therefore, the influence of panel

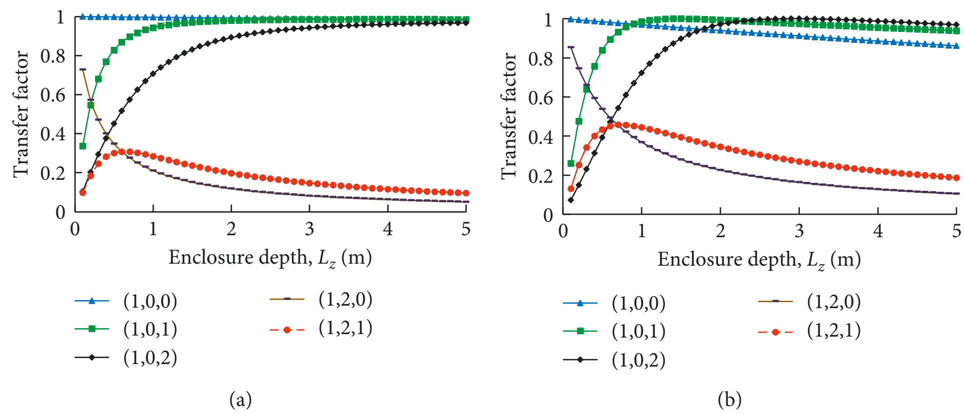


FIGURE 2: Effect of enclosure depth on the coupling property of the panel-enclosure coupled system with different panel boundary conditions: panel mode (2,1). (a) Simply supported panel mode (2,1). (b) Clamped panel mode (2,1).

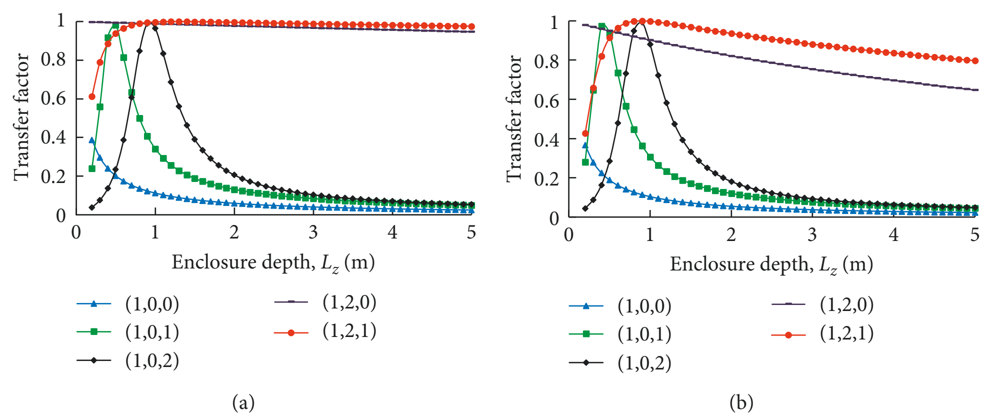


FIGURE 3: Effect of enclosure depth on the coupling property of the panel-enclosure coupled system with different panel boundary conditions: panel mode (2,3). (a) Simply supported panel mode (2,3). (b) Clamped panel mode (2,3).

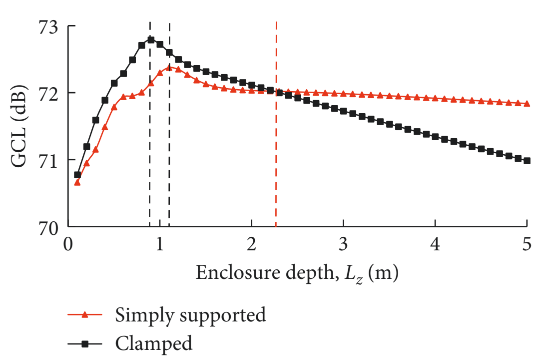


FIGURE 4: Effect of enclosure depth on the global coupling level of the panel-enclosure coupled system with different panel boundary conditions.

thickness on the coupling characteristics of the coupled system would be rather complicated.

Figure 5 shows the effect of panel thickness on the coupling property between panel mode (2,1) and each enclosure mode. As seen from Figure 5, the transfer factors between panel mode (2,1) and each enclosure mode first decrease and then increase and decrease again with the increase of panel thickness; strong couplings appear at different panel thickness. For instance, strong couplings between the SSP mode (2,1) and enclosure mode (1,0,0), (1,0,1), (1,0,2), (1,2,0) occur at the panel thickness of 2.3 mm, 2.8 mm, 3.8 mm, and 5.0 mm, respectively. In addition, in Figure 5(b), strong couplings between the CP mode (2,1) and the related enclosure mode occur at the panel thickness of 1.7 mm, 2.1 mm, 2.8 mm, and 3.7 mm, respectively. By comparing Figure 5(a) with 5(b), it can be concluded that the effects of panel thickness on the coupling characteristics of the system with different panel boundary conditions are similar. Furthermore, the variations of panel thickness affect the coupling characteristics of the system with a clamped panel more significantly than that of the system with a simply supported one. What is more, the panel boundary conditions affect the coupling properties between high-order panel mode and enclosure mode more significantly than that between low-order panel mode and enclosure mode.

Figure 6 shows the effect of panel thickness on the coupling property between panel mode (2,3) and each enclosure mode. By comparing Figure 6 with 5, it is observed that the effect of panel thickness on transfer factors between different panel modes and enclosure modes is the same. As seen from Figure 6(a), strong couplings between the SSP mode (2,3) and enclosure mode (1,0,0), (1,0,1), (1,0,2),

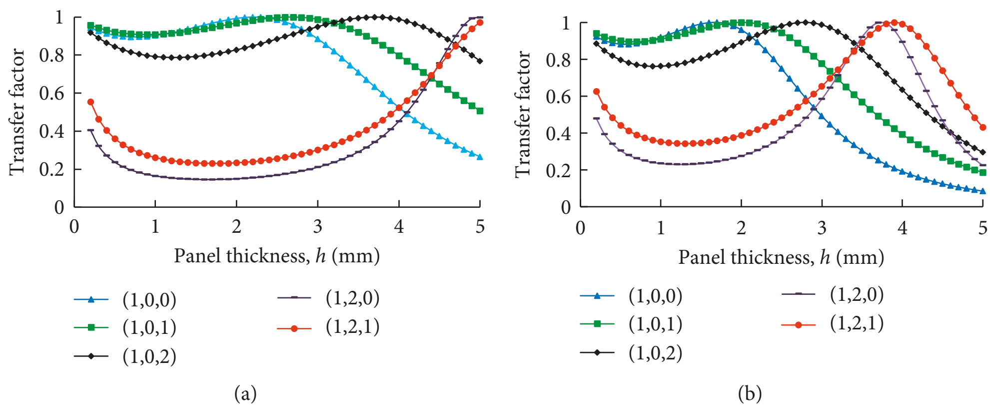


FIGURE 5: Effect of panel thickness on the coupling property of the panel-enclosure coupled system with different panel boundary conditions: panel mode (2,1). (a) Simply supported panel mode (2,1). (b) Clamped panel mode (2,1).

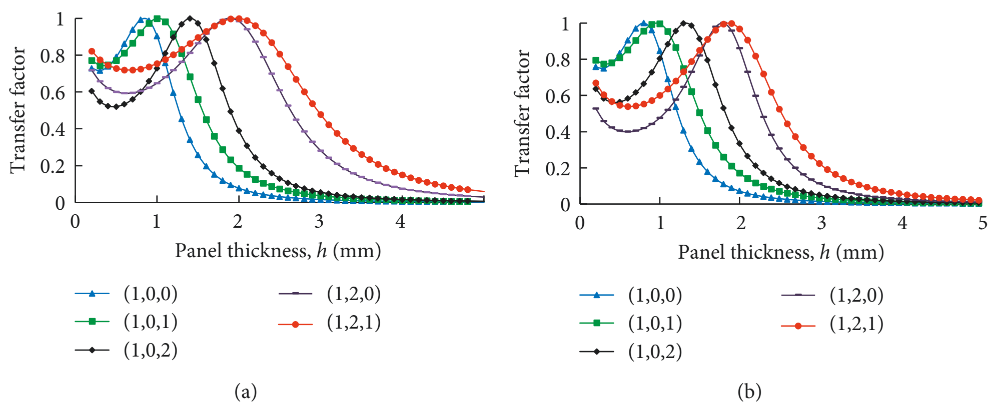


FIGURE 6: Effect of panel thickness on the coupling property of the panel-enclosure coupled system with different panel boundary conditions: panel mode (2,3). (a) Simply supported panel mode (2,3). (b) Clamped panel mode (2,3).

(1,2,0), (1,2,1) appear at the panel thickness of 0.9 mm, 1.1 mm, 1.4 mm, 1.9 mm, and 2.1 mm, respectively. In Figure 6(b), strong couplings between the CP mode (2,3) and these enclosure modes occur at the panel thickness of 0.8 mm, 0.9 mm, 1.3 mm, 1.8 mm, and 1.9 mm, respectively. There is a large energy conversion between the panel modes and enclosure modes at these panel thickness regions. Furthermore, as seen from Figure 6, when the panel thickness $h > 2.2$ mm, there are no strong couplings between the panel mode (2,3) and the enclosure modes any more.

Moreover, by comparing Figure 6(a) with 6(b), it is observed that panel boundary conditions weakly affect the coupling properties between the panel mode (2,3) and each enclosure mode. Additionally, by comparing Figure 5 with 6, it is found that the panel thickness regions where strong couplings occur become smaller as the resonance frequency of panel mode increases. Similarly, the influence of different panel boundary conditions on the coupling characteristics of the system becomes weaker with the increase of the resonance frequency of panel mode. This characteristic resembles the influence of enclosure depth on the coupling property of the system with different panel boundary conditions.

Figure 7 shows the effect of panel thickness on the global coupling level of the coupled system with different panel

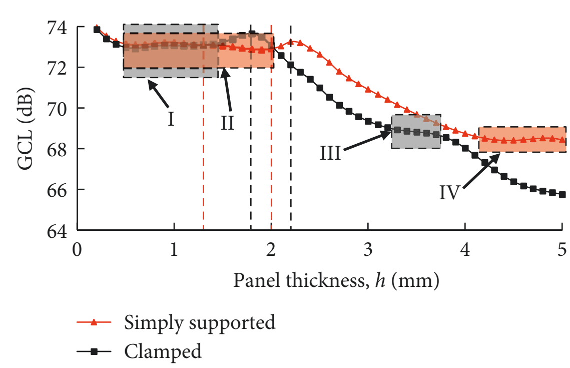


FIGURE 7: Effect of panel thickness on the global coupling level of the panel-enclosure coupled system with different panel boundary conditions.

boundary conditions. As seen from Figure 7, in the panel thickness range of 0.2~1.4 mm, the global coupling level of the system with a clamped panel is basically the same to that of the system with a simply supported one. Besides, in the panel thickness range of 1.4~2.0 mm, the global coupling level of the system with a clamped panel is larger than that of the system with a simply supported one, while in the panel thickness range of 2.0~5.0 mm, the global coupling level of the system with a clamped panel is smaller. From a global perspective, compared to the coupled system with a simply supported panel, the global coupling level versus panel thickness curve of the system with a clamped panel tends to move and compress towards the direction of smaller panel thickness.

Moreover, in the panel thickness regions I and III, the global coupling levels of the coupled system with a clamped panel remain basically unchanged with the increase of panel thickness. Similarly, in the panel thickness regions II and IV, the global coupling level of the system with a simply supported panel is weakly affected by the variations of panel thickness. Furthermore, when the thickness of clamped panel is greater than 1.8 mm, the global coupling level of the coupled system starts to decrease dramatically as the panel thickness increases. Similarly, when the thickness of simply supported panel is larger than 2.2 mm, the global coupling level of the coupled system begins to decrease with the increase of panel thickness. Therefore, it can be drawn that the global coupling level of the coupled system does not decrease exactly with the increase of panel thickness. What is more, these panel thickness regions where the global coupling level remains unchanged could also provide a basis for the lightweight structure design of the system.

4.3. Effect of Panel Aspect Ratio on Global Coupling Level of the Coupled System. As analyzed above, the enclosure depth and panel thickness affect the coupling characteristics of the system by changing the resonance frequency of panel mode or enclosure mode. Unlike these two structural parameters, when the panel area is a constant, both the resonance frequencies of the panel mode and the enclosure mode may possibly change at the same time as the panel aspect ratio changes. Therefore, the effect of panel aspect ratio on the coupling property between a panel mode and an enclosure mode is much more complicated than that of other structural parameters. However, the influence of panel aspect ratio on the coupling characteristics of the coupled system has not been considered in the existing literatures.

Figure 8 shows the effect of panel aspect ratio on the coupling property between panel mode (2,1) and each enclosure mode. As seen from Figure 8(a), the variations of panel aspect ratio weakly affect the coupling property between SSP mode (2,1) and enclosure mode (1,0,0), (1,0,1), (1,0,2). Besides, the transfer factors between SSP mode (2,1) and enclosure mode (1,0,1) are always approximately equal to 1.0 in the panel aspect ratio range of 0.5~5.0, which is not basically affected by the panel aspect ratio variations. Additionally, the transfer factors between SSP mode (2,1) and enclosure mode (1,2,0), (1,2,1) increase first and then decrease dramatically as the panel aspect ratio increases, and the strong couplings appear at the panel aspect ratio of 0.4 and 0.5, respectively. By comparing Figure 8(a) with 8(b), it is observed that the effects of panel aspect ratio on the coupling property of the coupled system with different panel boundary conditions are similar. However, the variations of panel aspect ratio affect the coupling properties of the system with a clamped panel more significantly than that of the system with a simply supported one.

Figure 9 shows the effect of panel aspect ratio on the coupling property between panel mode (2,3) and each enclosure mode. As seen from Figure 9, the transfer factors between panel mode (2,3) and each enclosure mode increase first and then decrease as the panel aspect ratio increases. In addition, with the increase of resonance frequencies of enclosure modes, the maximum amplitudes of the transfer factors between panel mode (2,3) and each enclosure mode increase dramatically. Furthermore, by comparing Figure 9(a) with 9(b), it is observed that the influence of different panel boundary conditions on the coupling properties between panel mode (2,3) and the higher-order enclosure modes is very significant. However, the amplitudes of the transfer factors between panel mode (2,3) and the lower-order enclosure modes are slightly affected by different panel boundary conditions.

Figure 10 shows the effect of panel aspect ratio on the global coupling level of the coupled system with different panel boundary conditions. As seen from Figure 10, the global coupling levels of the coupled system first increase and then decrease as the panel aspect ratio increases. In addition, the global coupling level versus panel aspect ratio curves are symmetrically distributed to $\alpha = 1$, and the maximum global coupling levels occur at the panel aspect ratio of $\alpha = 1$. Therefore, it can be concluded that when the panel length L_x is equal to the panel width L_y, the global coupling level of the coupled system is likely to be very strong. Moreover, the effect of panel aspect ratio on the global coupling level of the coupled system with a clamped panel is more significant than that of the system with a simply supported one. This characteristic resembles the effects of enclosure depth and panel thickness on the global coupling levels of the coupled system with different panel boundary conditions.

4.4. Comprehensive Effect of Structural Parameters on the Global Coupling Levels of the Coupled System. Based on the control variable method, the influence of each structural parameter on the global vibroacoustic coupling characteristics of the panel-enclosure system is analyzed above. To provide a theoretical basis for the parameters design of the panel-enclosure system with low global coupling characteristics, the comprehensive effect of all the structural parameters on the global coupling levels of the system with different panel boundary conditions is analyzed in this section.

Figure 11 shows the comprehensive effect of structural parameters on the global coupling level of the coupled system when the enclosure depth L_z is 0.4 m, 1.6 m, 3.0 m, and 4.8 m, respectively. According to the differences in the global coupling levels, the global coupling levels of the coupled system can be divided into four regions, A, B, C and D, as shown in Figure 11(a). As analyzed in Section 2, 60 dB is used as the dividing line to judge whether the global coupling characteristics of the coupled system are strong or

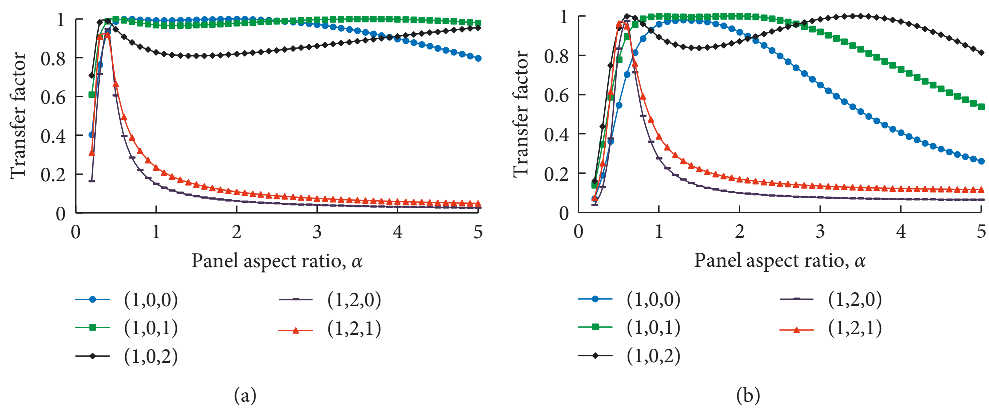


Figure 8: Effect of panel aspect ratio on the coupling property of the panel-enclosure coupled system with different panel boundary conditions: panel mode (2,1). (a) Simply supported panel mode (2,1). (b) Clamped panel mode (2,1).

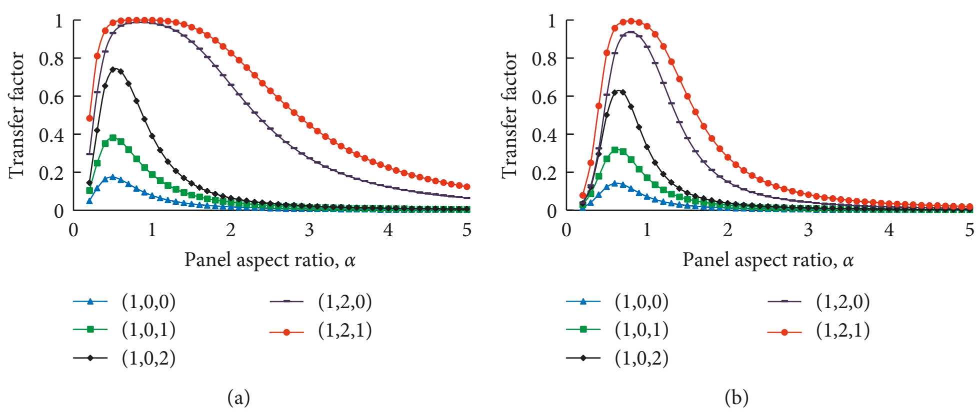


Figure 9: Effect of panel aspect ratio on the coupling property of the panel-enclosure coupled system with different panel boundary conditions: panel mode (2,3). (a) Simply supported panel mode (2,3). (b) Clamped panel mode (2,3).

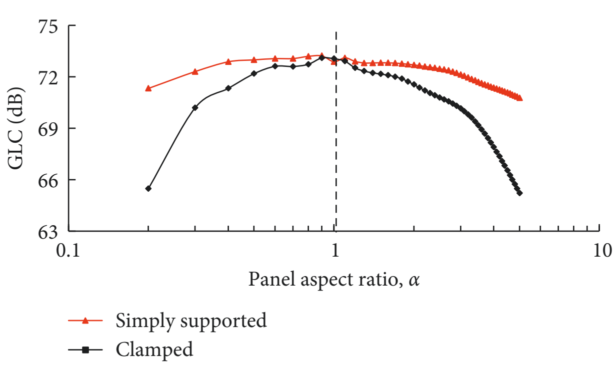


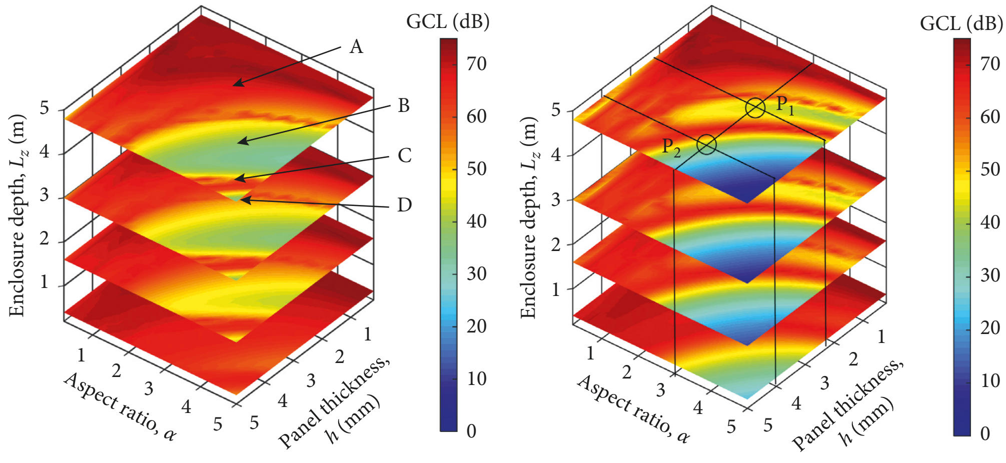
Figure 10: Effect of panel aspect ratio on the global coupling level of the panel-enclosure system with different panel boundary conditions.

not. Take Figure 11(a) for example, the global coupling levels of the coupled system in regions A and C are higher than 60 dB, indicating that there possibly exists some strong couplings, while in regions B and D, the global coupling levels are smaller than 60 dB, which reveals that there cannot be any significant couplings in these regions.

By comparing Figure 11(a) with 11(b), it is observed that the influence of panel boundary conditions on the global coupling level of the coupled system is very significant. Besides, under the same conditions, the regions where there are no strong couplings of the system with a clamped panel are greater than that of the system with a simply supported one. To design a panel-enclosure system (like a special vehicle cabin) with low global coupling level, taking Figure 11(b) for example, assume that the panel aspect ratio α is 3.0, and the enclosure depth L_z is 4.8 m, it is noticed that both the global coupling levels at P_1 and P_2 meet the requirements. Therefore, as far as the lightweight structural design and cost savings are concerned, it is better that the panel thickness is set as 2.3 mm.

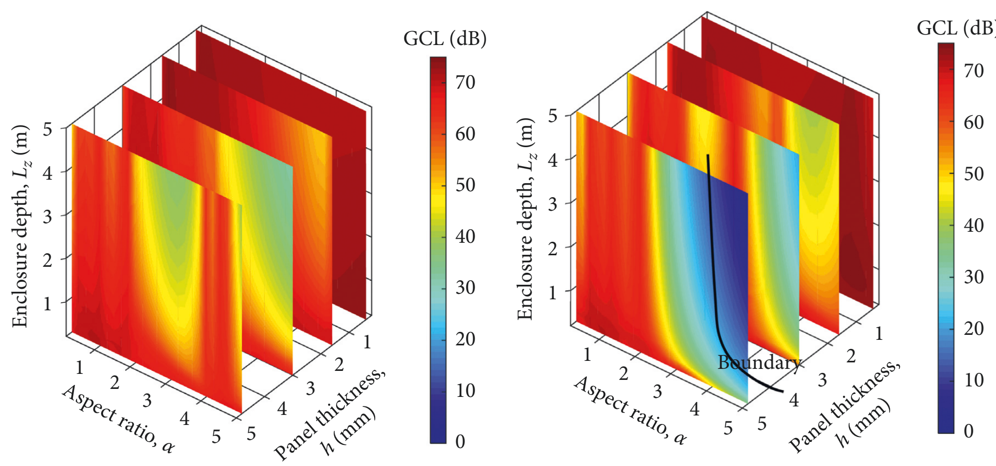
Figures 12 and 13 show the comprehensive effects of structural parameters on the global coupling levels of the coupled system with different panel boundary conditions, when the panel thickness and panel aspect ratio are at some specific values, respectively. The influence of panel thickness and panel aspect ratio on the global coupling level of the system resembles that of enclosure depth. Hence, it would not be repeated herein any more.

Figure 14 shows the comprehensive effect of structural parameters on the global coupling level of the coupled



(a) (b)

Figure 11: Comprehensive effect of structural parameters on the global coupling level of the coupled system: enclosure depth $L_z = 0.4$ m, 1.6 m, 3.0 m, and 4.8 m. (a) Simply supported panel. (b) Clamped panel.



(a) (b)

Figure 12: Comprehensive effect of structural parameters on the global coupling level of the coupled system: panel thickness $h = 0.4$ mm, 1.6 mm, 3.0 mm, and 4.8 mm. (a) Simply supported panel. (b) Clamped panel.

system with different panel boundary conditions. As seen from Figure 14, the panel boundary conditions have a significant influence on the global coupling levels of the coupled system, especially in some structural parameter regions. Furthermore, the regions where there are no strong couplings of the coupled system with a clamped panel are greater than that of the system with a simply supported panel.

Theoretically, the panel boundary conditions affect the global coupling level of the coupled system by changing the resonance frequency of the panel mode. The strong couplings occur only when the resonance frequency of panel mode is close to or equal to that of the enclosure acoustical mode. For example, assume that $\omega_{aN} = \omega_{pMs}$, thus the transfer factor between the M^{th} simply supported panel mode and the N^{th} enclosure acoustical mode will be 1. While the resonance frequency ω_{pMc} of the M^{th} clamped panel is generally greater than ω_{pMs}, therefore, by lowering the panel thickness or the enclosure depth (only when the mode index n is nonzero) to a specific value, the coupling between the M^{th} clamped panel mode and the N^{th} enclosure acoustical mode can then become strong. This is the root reason why those transfer factor versus panel thickness (or enclosure depth) curves of the system with a clamped panel tends to move and compress towards the direction of smaller structural parameters, compared to that of the system with a simply supported panel. Therefore, under the same conditions, the regions where

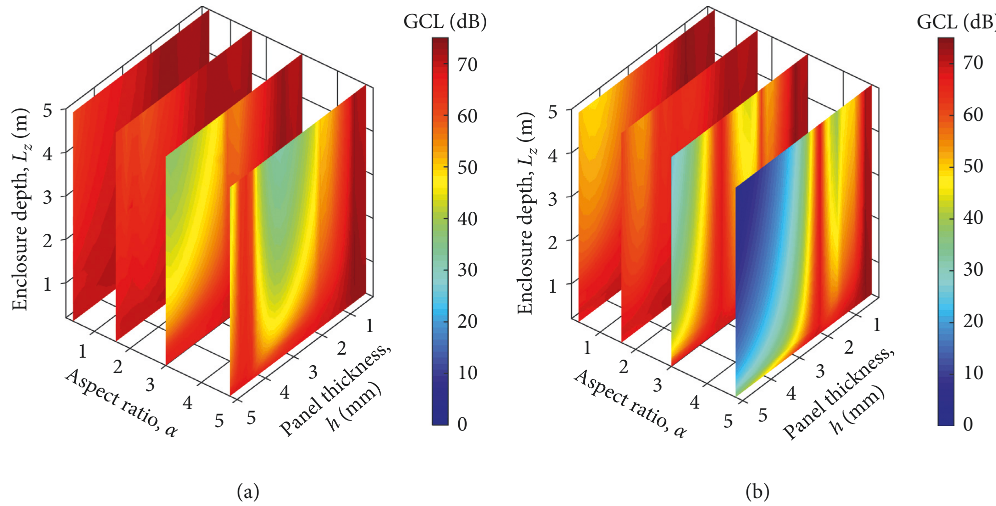


Figure 13: Comprehensive effect of structural parameters on global coupling level of the coupled system: panel aspect ratio α= 0.4, 1.6, 3.0, and 4.8. (a) Simply supported panel. (b) Clamped panel.

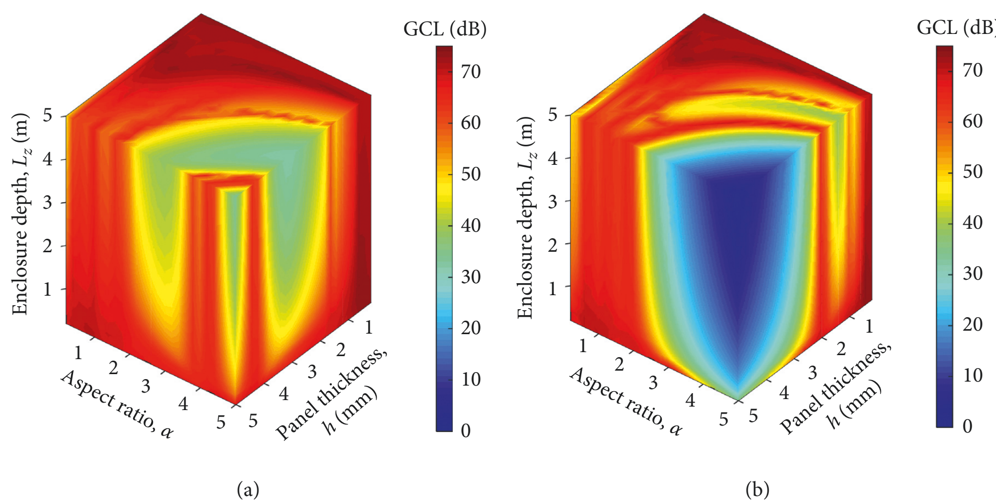


Figure 14: Comprehensive effect of structural parameters on global coupling level of the panel-enclosure system with different panel boundary conditions. (a) Simply supported panel. (b) Clamped panel.

there is no strong global coupling level of the system with a clamped panel are greater than that of the system with a simply supported panel.

5. Conclusions

This paper presents a parametric study on the vibroacoustic coupling characteristics of a rectangular enclosure bounded by a flexible panel with different boundary conditions. Firstly, a novel index of the global coupling level was proposed to describe the global vibroacoustic coupling degree between multiple panel vibration modes and the multiple enclosure acoustical modes of the coupled system. Then, the effects of structural parameters, as well as the panel boundary conditions, on the transfer factor between single panel mode and enclosure mode of the coupled system were investigated. Afterwards, the influence of structural parameters on global coupling levels of the coupled system with different panel boundary conditions was obtained based on the numerical results of transfer factors. Moreover, according to the comprehensive influence of structural parameters on global coupling levels, design methods of the coupled system with low global coupling levels were discussed. Finally, the mechanism of how the panel boundary conditions affect the coupling characteristics of the coupled system was analyzed as well.

The results show that the structural parameters, including the enclosure depth, panel thickness, and panel aspect ratio, have similar influence on the coupling properties of the coupled system with different panel boundary conditions. Additionally, the influence of structural parameters on the coupling property of the coupled system with a clamped panel is more sensitive than that of the system with a simply supported one. Moreover, the structural parameters, especially the enclosure depth and panel thickness, are not completely negative correlated to the global coupling levels of the coupled system. Furthermore, the global coupling level versus panel thickness (or enclosure depth) curve of the system with a clamped panel tends to move and compress towards the direction of smaller panel thickness (or enclosure depth), when compared to that of the coupled system with a simply supported panel. What is more, under the same conditions, the regions where there is no strong coupling of the system with a clamped panel are greater than that of the system with a simply supported one.

This study provides a solution to the evaluation problem of the global coupling characteristics of the rectangular panel-enclosure system. It also provides a theoretical basis for the acoustic design of the rectangular vehicle cabins with low global vibroacoustic coupling level, as well as the lightweight structure design of the system. Further studies may be carried out on the effect of panel aspect ratio on energy transfer, resonance frequency, and decay times of acoustical modes.

Data Availability

The data used to support the findings of this study are available from the corresponding author upon request.

Conflicts of Interest

The authors declare that they have no conflicts of interest with respect to the research, authorship, and/or publication of this article.

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References

[1] E. H. Dowell and H. M. Voss, "The effect of a cavity on panel vibration," *AIAA Journal*, vol. 1, no. 2, pp. 476-477, 1963.

[2] E. H. Dowell, G. F. Gorman, and D. A. Smith, "Acousto-elasticity: general theory, acoustic natural modes and forced response to sinusoidal excitation, including comparisons with experiment," *Journal of Sound and vibration*, vol. 52, no. 4, pp. 519–542, 1977.

[3] J. Pan and D. A. Bies, "The effect of fluid–structural coupling on sound waves in an enclosure-experimental part," *Journal of the Acoustical Society of America*, vol. 87, no. 2, pp. 708–717, 1990.

[4] J. Pan, S. J. Elliott, and K.-H. Baek, "Analysis of low frequency acoustic response in a damped rectangular enclosure," *Journal of Sound and Vibration*, vol. 223, no. 4, pp. 543–566, 1999.

[5] Y. Wang, J. Zhang, W. Gu, and V. Q. Le, "Analysis of structural-acoustic coupling characteristics of rectangular enclosure with different parameters," *Journal of Southeast University (Natural Science Edition)*, vol. 43, no. 3, pp. 503–508, 2013, in Chinese.

[6] C.-C. Sung and J. T. Jan, "The response of and sound power radiated by a clamped rectangular plate," *Journal of sound and vibration*, vol. 207, no. 3, pp. 301–317, 1997.

[7] J. P. Arenas, "On the vibration analysis of rectangular clamped plates using the virtual work principle," *Journal of Sound and Vibration*, vol. 266, no. 4, pp. 912–918, 2003.

[8] Y. Wang, J. Zhang, and V. Le, "Vibroacoustic analysis of a rectangular enclosure bounded by a flexible panel with clamped boundary condition," *Shock and Vibration*, vol. 2014, Article ID 872963, 17 pages, 2014.

[9] M. Sadri and D. Younesian, "Vibroacoustic analysis of a sandwich panel coupled with an enclosure cavity," *Composite Structures*, vol. 146, pp. 159–175, 2016.

[10] H. Cui, R. Hu, and N. Chen, "Modelling and analysis of acoustic field in a rectangular enclosure bounded by elastic plates under the excitation of different point force," *Journal of Low Frequency Noise, Vibration and Active Control*, vol. 36, no. 1, pp. 43–55, 2017.

[11] M. Sadri and D. Younesian, "Vibro-acoustic analysis of a coach platform under random excitation," *Thin-Walled Structures*, vol. 95, pp. 287–296, 2015.

[12] R. Jiao and J. Zhang, "Vibro-acoustic modeling of a rectangular enclosure with a flexible panel in broad range of frequencies and experimental investigations," *Journal of Vibroengineering*, vol. 18, no. 4, pp. 2683–2692, 2016.

[13] D. Chen and N. Chen, "Active control of structural sound radiation in an acoustic enclosure consisting of flexible structure," *Journal of Southeast University*, vol. 30, no. 3, pp. 318–322, 2014.

[14] W. Elwali, M. Li, and T. C. Lim, "Numerical analysis of active structural acoustic control in an enclosed cavity," *Noise Control Engineering Journal*, vol. 64, no. 4, pp. 459–468, 2016.

[15] J.-H. Ho and A. Berkhoff, "Comparison of various decentralized structural and cavity feedback control strategies for transmitted noise reduction through a double panel structure," *Journal of Sound and Vibration*, vol. 333, no. 7, pp. 1857–1873, 2014.

[16] Y. Sun, T. Yang, M. Zhu, and J. Pan, "The use of an active controlled enclosure to attenuate sound radiation from a heavy radiator," *Journal of Sound and Vibration*, vol. 392, pp. 1–17, 2017.

[17] U. C. Jeong, J. S. Kim, Y. D. Kim, and J. E. Oh, "Reduction of radiated exterior noise from the flexible vibrating plate of a rectangular enclosure using multi-channel active control," *Applied Acoustics*, vol. 105, pp. 45–54, 2016.