Sound transmission between rooms coupled through partition with elastically restrained edges

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ABSTRACT

Sound transmission between rooms is often encountered for the acoustic design and noise control engineers. A clear understanding on the acoustical behavior of rectangular cavities connected by flexible panel structure can be of the fundamental significance. In this paper, sound transmission between rooms through the flexible partition with its edges elastically restrained is investigated. Both the translational and rotational springs are assumed along each panel edge to simulate the structural boundary conditions. The energy variational formulations in conjunction with Rayleigh-Ritz procedure are employed for this structural-acoustic coupling system modeling with the improved Fourier series constructed as the admissible functions, in which the supplementary terms are introduced to overcome potential discontinuities associated with the spatial derivatives of the conventional Fourier series on the panel-cavity interface as well as the panel elastic edge supports. Numerical examples are then presented to demonstrate the reliability and effectiveness of the current model through the comparison with the predicted data obtained by Finite Element Analysis using NASTRAN. Effect of the position and boundary restraining condition of flexible partition on the sound transmission characteristics of coupled rooms is analyzed in detail. Finally, some useful and interesting findings from this work are given in the conclusion section.

Keywords: Structural-acoustic Coupling System, Elastically Restrained Edges, Improved Fourier Series Method I-INCE Classification of Subjects Number(s): 76.9

1. INTRODUCTION

Nowadays there is an increasing desire for stability of mechanical structures and comfort of environment, indoor sound field noise control is widely concerned in various fields of engineering and technology, such as vehicular cabins, aircraft fuselages, acoustical instruments, building constructions and so on. Due to the practical importance, various studies have been proposed to predict the modal characteristics and forced response of room sound field in the past decades. Almost all the acoustic problems are related to vibro-acoustic interactions, and hence the vibro-acoustic coupling system has received more and more considerable attention.

Broadly speaking, Dowell and Voss (1) did the initial work in the vibro-acoustic coupling area. They presented the natural modes and frequencies for a vibrating plate located on one side of a rectangular box. Then in 1968, nonlinear plate stiffness and mutual interaction between the plate and external and/or internal sound field were considered by using the method of modal expansions for the plate and cavity (2). As the developing of theoretical model, Dowell et al (3) demonstrated full coupling between the wall and interior acoustic cavity based on the theory of acoustoelasticity, and simplified formulae for interior sound levels in terms of cavity, wall and external acoustic field parameters were developed, which laid the foundation for future development of vibro-acoustic coupling problems. After that, absorption material and a double wall/cavity system were introduced in the noise control of the cavity interior sound field (4), they used the same method to predict the noise transmission through a single wall or a double wall/cavity system into a cavity. Therefore an accurate characterization of the sound-structure interaction plays a key role in the prediction of acoustic field.

The mode superposition method usually used in structural modeling for dynamics in these past, it had been widely used to obtain governing equations since it was extended to the coupling system (5-8). In the paper given by Pan and Bies (9, 10), a solution for the decay time and resonance frequency of the free-vibration mode of a panel-cavity system was determined from both theoretical and experimental ways, in which the boundary condition of the panel was set to simply supported. Scarpa and Curti (11) analyzed a rectangular acoustic cavity closed at one end by a simply supported plate with isotropic material. They employed Kirchoff-Vaslov equations to describe the modal vectors of the uncoupled structures without considering rotary inertia or shear deformations. Al-Bassyiouni and Balachandran (12) took the sound field outside the enclosure into account to construct the model of a rectangular enclosure with rigid walls and one flexible panel, and applied Active Structural Acoustical Control on the system.

Beyond the work done for cavities with simple geometry, Li and Cheng (13) introduced a leaning wall in a rectangular-like cavity. Acoustic modes and the coupling characteristics of the irregular-shaped system were investigated by using the combined integro-modal approach. Hong and Kim (14,15) proposed a formulation of a one-dimensional acoustic pipe whose one end was closed and the other end was coupled to a one-degree-of-freedom mass-spring system as the source. Then, the equations were solved by the modal expansion method using uncoupled natural modes of the sub-systems. Furthermore, they investigated the effects of acoustic absorbing material applied on all or part of the structure, as well as the viscous or structural damping elements in the system. Lee et al (16,17) proposed a new coupled structural–acoustic model composed of double cavities connected by a neck simulating a passenger compartment with a trunk compartment, the vibro-acoustical behavior had been theoretically and experimentally discussed.

In recent years, a structural-acoustic model of a rectangular acoustic cavity bounded by a flexible plate with elastically restrained edges was developed by Du et al (18) in which the 2D and 3D improved Fourier series were used to represent the displacement on the plate and the sound pressure inside the cavity, respectively. The numerical results matched very well with the comparison results. This work is an extension of the work carried out by reference (18), where the vibro-acoustic analysis of a three-dimensional acoustic cavity bounded by a flexible panel with general elastically restrained boundary conditions was presented by using the improved Fourier series method. By contrast, the overall purpose of the current work is to study the indoor sound field between two rectangular rooms coupled through a flexible partition. There are three components in the whole coupled system, the admissible functions of a panel structure and two rooms will be presented by two-dimensional and three-dimensional improved Fourier series, respectively. The use of this method can overcome the discontinuities of the admissible functions in the whole solution domain. Then the Rayleigh-Ritz method will be employed to solve the eigenvalue problem of the coupled system. To validate the current approach, numerical results are presented and discussed by comparing with the results calculated by finite element method from Nastran. It is noteworthy that the boundary conditions of the partition are mostly restricted to classical types in existing literatures, but in engineering practice, the boundary restrained should be more complex. Therefore, the elastically boundary restrained partition will be considered. In addition, the different positions of the panel structure in the whole enclosure are also studied to deeply understand its influence to the modal properties of the coupled system.

2. THEORETICAL FORMULATIONS

2.1 Problem description

To investigate the sound transmission between two rooms coupled through partition, a simplified theoretical model for the modal analysis of two rectangular rooms connected via a flexible panel is considered, as shown in Figure 1. The floor of room1 is taken at z=0. The size of two rooms is considered as $a \times b \times h_1$ and $a \times b \times h_2$, respectively. The upper side of room$_1$, also the bottom side of room$_2$, is covered by a flexible rectangular partition with a thickness of h_p. As the figure shown, room$_1$ and room$_2$ both consist of five rigid walls and one elastically boundary restrained flexible panel. Two sets of linear springs are assumed to express translational and rotational springs along each panel edge to simulate the structural boundary conditions of the panel. The stiffness of translational springs along x=0 edge is presented by the symbol k_{x0}, K_{x0} presents the rotational springs stiffness at the same time. Through adjusting the stiffness coefficient, various boundary conditions can be obtained, as the stiffness of boundary restrained springs being equal to zero or infinity, all the classical boundary conditions can be obtained as special cases.

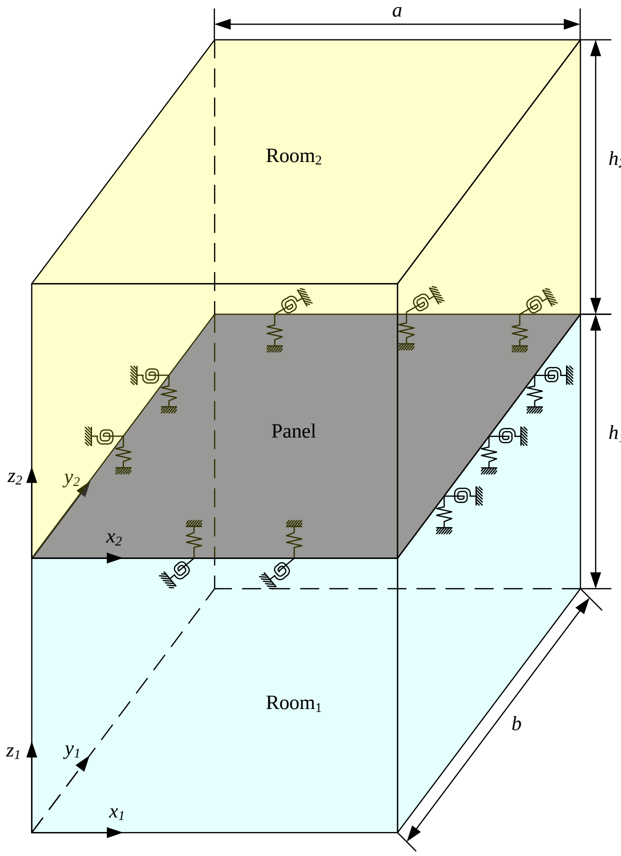


Figure 1 The coordinate system for two rooms coupled through flexible partition with elastically restrained edges

2.2 Series representation of filed functions for the coupled system

In the coupled system, the transverse bending displacement of the panel and the acoustic pressure in two rooms are interacted with each other. The transverse bending vibration of the panel radiates acoustic waves into the acoustic field in the rooms; the acoustic waves excite the panel vibration at the same time. For the transverse bending problems of the panel, the improved Fourier series method constructed in reference (19) is employed in this study, the displacement function can be expanded as

$$\begin{aligned} w(x_p,y_p) = &\sum_{m_x^p=0}^{\infty}\sum_{m_y^p=0}^{\infty} A^p_{m_x^p m_y^p}\cos\lambda_{m_x^p}x_p\cos\lambda_{m_y^p}y_p \\ &+\sum_{m_x^p=0}^{\infty}\left[a^p_{m_x^p}\zeta_{1b}(y_p)+c^p_{m_x^p}\zeta_{3b}(y_p)+b^p_{m_x^p}\zeta_{2b}(y_p)+d^p_{m_x^p}\zeta_{4b}(y_p)\right]\cos\lambda_{m_x^p}x_p \\ &+\sum_{m_y^p=0}^{\infty}\left[e^p_{m_y^p}\zeta_{1a}(x_p)+g^p_{m_y^p}\zeta_{3a}(x_p)+f^p_{m_y^p}\zeta_{2a}(x_p)+h^p_{m_y^p}\zeta_{4a}(x_p)\right]\cos\lambda_{m_y^p}y_p \end{aligned} \tag{1}$$

where $\lambda_{m_x^p} = m_x^p\pi/a$, $\lambda_{m_y^p} = m_y^p\pi/b$,and the supplementary functions are chosen in the following form

$$\zeta_{1a}(x_p) = 9a\sin(\pi x_p/2a)/4\pi - a\sin(3\pi x_p/2a)/12\pi \tag{2a}$$

$$\zeta_{2a}(x_p) = -9a\cos(\pi x_p/2a)/4\pi - a\cos(3\pi x_p/2a)/12\pi \tag{2b}$$

$$\zeta_{3a}\left(x_p\right)=a^3\sin\left(\pi x_p/2a\right)/\pi^3-a^3\sin\left(3\pi x_p/2a\right)/3\pi^3 \tag{2c}$$

$$\zeta_{4a}\left(x_p\right)=-a^3\cos\left(\pi x_p/2a\right)/\pi^3-a^3\cos\left(3\pi x_p/2a\right)/3\pi^3 \tag{2d}$$

At the edge of the panel, the first and third derivatives of supplementary functions listed above take the special value of $\zeta'_{1a}\left(0\right)=\zeta'_{2a}\left(a\right)=\zeta'''_{3a}\left(0\right)=\zeta'''_{4a}\left(a\right)=1$, all the other derivatives are identically equal to zero, which means the possible discontinuities of displacement function can be avoided through introducing the supplementary functions in the mathematical calculation. Furthermore, with all the supplementary functions, the convergence speed for different boundary conditions can also be improved.

Recently, the improved Fourier series method was extended to analyze the sound field of three-dimensional rectangular acoustic cavities with arbitrary impendence boundary conditions (20). For this study, two rooms are connected via a communal wall, namely a flexible panel with elastically boundary restrained; other walls are all rigid. In order to analyze the pressure and velocity distributions in the coupling interface correctly and conveniently, the sound pressure of two rooms can be rewritten as:

$$\begin{aligned} p_1\left(x_1,y_1,z_1\right)=&\sum_{m_x^1=0}^{\infty}\sum_{m_y^1=0}^{\infty}\sum_{m_z^1=0}^{\infty}A^1_{m_x^1m_y^1m_z^1}\cos\lambda_{m_x^1}x_1\cos\lambda_{m_y^1}y_1\cos\lambda_{m_z^1}z_1\\ &+\sum_{m_x^1=0}^{\infty}\sum_{m_y^1=0}^{\infty}b^1_{m_x^1m_y^1}\xi_{2c}\left(z_1\right)\cos\lambda_{m_x^1}x_1\cos\lambda_{m_y^1}y_1 \end{aligned} \tag{3}$$

$$\begin{aligned} p_2\left(x_2,y_2,z_2\right)=&\sum_{m_x^2=0}^{\infty}\sum_{m_y^2=0}^{\infty}\sum_{m_z^2=0}^{\infty}A^2_{m_x^2m_y^2m_z^2}\cos\lambda_{m_x^2}x_2\cos\lambda_{m_y^2}y_2\cos\lambda_{m_z^2}z_2\\ &+\sum_{m_x^2=0}^{\infty}\sum_{m_y^2=0}^{\infty}a^2_{m_x^2m_y^2}\xi_{1c}\left(z_2\right)\cos\lambda_{m_x^2}x_2\cos\lambda_{m_y^2}y_2 \end{aligned} \tag{4}$$

here, $\lambda_{m_x^1}=m_x^1\pi/a$, $\lambda_{m_y^1}=m_y^1\pi/b$, $\lambda_{m_z^1}=m_z^1\pi/h_1$, $\lambda_{m_x^2}=m_x^2\pi/a$, $\lambda_{m_y^2}=m_y^2\pi/b$, $\lambda_{m_z^2}=m_z^2\pi/h_2$, and

$$\xi_{2c}\left(z_1\right)=z_1^2\left(z_1/h_1-1\right)/h_1\ ,\ \ \xi_{1c}\left(z_2\right)=z_2\left(z_2/h_2-1\right)^2 \tag{5a, b}$$

The relevant derivatives of these supplementary functions across the end surface can be obtained as

$$\xi_{2c}\left(0\right)=\xi_{2c}\left(h_1\right)=\xi'_{2c}\left(0\right)=0\ ,\ \ \xi'_{2c}\left(0\right)=1 \tag{6a, b}$$

$$\xi_{1c}\left(0\right)=\xi_{1c}\left(h_2\right)=\xi'_{1c}\left(h_2\right)=0\ ,\ \ \xi'_{1c}\left(h_2\right)=1 \tag{7a, b}$$

It is noteworthy that with all unknown Fourier series coefficients being solved, the displacement and the sound pressure functions are capable of representing the exact solution to the problem.

2.3 Solution of the room-panel-room coupling system

In order to solve the vibration and acoustic coupling problem, the energy formulation description will be used to drive the governing equation for three coupled systems. Ordinarily, this method is considered as an approximate calculation method in solving structural dynamic problems (*weak form*).

In this work, since the displacement and sound pressure functions of the coupling system are all sufficiently smooth in the whole solving domain including the edges and surfaces of the structure, the weak solution is also seen as accurately enough.

Firstly, for the flexible panel partition, its Lagrangian of the plate structure is defined as

$$L_p = U_p - T_p + W_{c1\&p} + W_{c2\&p} \tag{8}$$

in which, U_p is the elastic strain energy of the panel due to the transverse vibration; T_p is the kinetic energy of the vibrating panel; $W_{c1\&p}$ is the work associated with the sound pressure of room$_1$; and the last term $W_{c2\&p}$ is the work associated with the sound pressure of room$_2$.

For small amplitude vibration, the total potential energy of the flexible panel including elastic strain energy as well as the potential energy stored in the elastic boundary restraints can be written down as

$$\begin{aligned} U_p = {} & \frac{D}{2}\int_0^b\int_0^a\left[\left(\frac{\partial^2 w}{\partial x_p^2}\right)^2+\left(\frac{\partial^2 w}{\partial y_p^2}\right)^2+2\mu\frac{\partial^2 w}{\partial x_p^2}\frac{\partial^2 w}{\partial y_p^2}+2(1-\mu)\left(\frac{\partial^2 w}{\partial x_p\partial y_p}\right)^2\right]dx_p dy_p \\ & +\frac{1}{2}\int_0^b\left[k_{x0}w^2+K_{x0}\left(\frac{\partial w}{\partial x_p}\right)^2\right]_{x_p=0}dy_p+\frac{1}{2}\int_0^b\left[k_{xa}w^2+K_{xa}\left(\frac{\partial w}{\partial x_p}\right)^2\right]_{x_p=a}dy_p \\ & +\frac{1}{2}\int_0^a\left[k_{y0}w^2+K_{y0}\left(\frac{\partial w}{\partial y_p}\right)^2\right]_{y_p=0}dx_p+\frac{1}{2}\int_0^a\left[k_{yb}w^2+K_{yb}\left(\frac{\partial w}{\partial y_p}\right)^2\right]_{y_p=b}dx_p \end{aligned} \tag{9}$$

where, $D = Ec^3\big/\left[12\left(1-\mu^2\right)\right]$ denotes the flexural rigidity, and μ is the Poisson's ratio.

The kinetic energy associated with the vibrating panel can be expressed as

$$T_p = \frac{1}{2}\int_0^b\int_0^a \rho_p h_p\left(\partial w/\partial t\right)^2 dx_p dy_p = \frac{1}{2}\rho_p h_p \omega^2\int_0^b\int_0^a w^2 dx_p dy_p \tag{10}$$

in which ρ_p is the mass density of the panel; h_p is the thickness of the panel.

The work associated with the sound pressure of rooms is

$$W_{c1\&p} = \int_0^b\int_0^a w p_1 dx_p dy_p \tag{11}$$

and

$$W_{c2\&p} = \int_0^b\int_0^a w p_2 dx_p dy_p \tag{12}$$

Then, for the coupling acoustical cavities, the Lagrangian function of two rooms are defined as

$$L_1 = U_1 - T_1 - W_{p\&c1} \tag{13}$$

$$L_2 = U_2 - T_2 - W_{p\&c2} \tag{14}$$

in which the term U_1 and U_2 are the acoustic potential energy of two rooms; T_1 and T_2 are the kinetic energy of two rooms; $W_{p\&c1}$ and $W_{p\&c2}$ are the work associated with the vibration of the panel.

The acoustic potential energy of two rooms can be calculated from

$$U_n = \frac{1}{2\rho_0 c_0^2}\int_{V_n} p_n^2 dV_n \qquad (n=1,2) \tag{15}$$

in which ρ_0 and c_0 are the mass density and sound speed of the acoustic medium in rooms, respectively.

The kinetic energy of two rooms is

$$T_n = \frac{1}{2\rho_0\omega^2}\int_{V_n}\left[\left(\frac{\partial p_n}{\partial x_n}\right)^2 + \left(\frac{\partial p_n}{\partial y_n}\right)^2 + \left(\frac{\partial p_n}{\partial z_n}\right)^2\right]dV_n \qquad (n=1,2) \tag{16}$$

The work associated with panel vibration is equal to the work associated with the sound pressure of two rooms, that is

$$W_{p\&c1} = W_{c1\&p}, \quad W_{p\&c2} = W_{c2\&p} \tag{17a, b}$$

To find the extremum of the Lagrangian functions, the Rayleigh-Ritz procedure is used. In this case, minimizing the Lagrangian functions with respect to each unknown Fourier series coefficient. Following the Rayleigh-Ritz procedure, substituting the expansion of displacement function in equation (1) and the sound pressure functions in equation (3, 4) into the above three Lagrangian functions, minimizing equations will lead to the following matrix equation of the structural-acoustic coupling system:

$$\left\{\begin{bmatrix} \mathbf{K}_1 & 0 & 0 \\ \mathbf{C}_{c1\&p} & \mathbf{K}_p & \mathbf{C}_{c2\&p} \\ 0 & 0 & \mathbf{K}_2 \end{bmatrix} - \omega^2 \begin{bmatrix} \mathbf{M}_1 & -\rho_0\mathbf{C}^{\mathrm{T}}_{c1\&p} & 0 \\ 0 & \mathbf{M}_p & 0 \\ 0 & -\rho_0\mathbf{C}^{\mathrm{T}}_{c2\&p} & \mathbf{M}_2 \end{bmatrix}\right\} \begin{bmatrix} \mathbf{P}_1 \\ \mathbf{W}_p \\ \mathbf{P}_2 \end{bmatrix} = \begin{Bmatrix} 0 \\ 0 \\ 0 \end{Bmatrix} \tag{19}$$

The solution of equation (19) gives all the unknown Fourier series coefficients of the displacement and sound pressure functions of two rooms coupled through partition system, we can also obtain the system natural frequencies and mode shapes by solving the normal eigenvalue problem in equation (19).

3. NUMERICAL RESULTS AND DISCUSSIONS

In this section, several numerical examples will be presented to demonstrate the accuracy and reliability of the theoretical model constructed before studying the structural-acoustical behavior of rectangular rooms coupled through flexible panel structure. Since little data can be found in the literature for the room-panel-room model, the results by Finite Element Analysis from NASTRAN is employed for comparison. In the following cases, all the geometrical and material properties of analytical model are given in Table 1.

Table 1 – Geometrical and material parameters used in the simulation model of room-panel-room

Room_1	Panel	Room_2	Air
a=0.3 m b=0.38 m h_1=0.65 m	Young's modulus=71 GPa Density=2700 Kg/m^3 Poisson's ratio=0.3 Thickness=3 mm	a=0.3 m b=0.38 m h_2=0.45 m	Density=1.21 Kg/m^3 Sound velocity=340 m/s

For the purpose of verification, three uncoupled systems are firstly studied. The natural frequencies are calculated for two rigid acoustic rooms and a single flexible panel structure with simply supported (which will be denoted by SSSS) boundary conditions. For the present model, the SSSS boundary condition can be obtained by setting the stiffness of translational springs as infinity (5×10^9 in the numerical calculation), the stiffness of rotational springs as zero at the same time. Table 2 and Table 3 tabulate the first six natural frequencies for three uncoupled systems and the system after coupling.

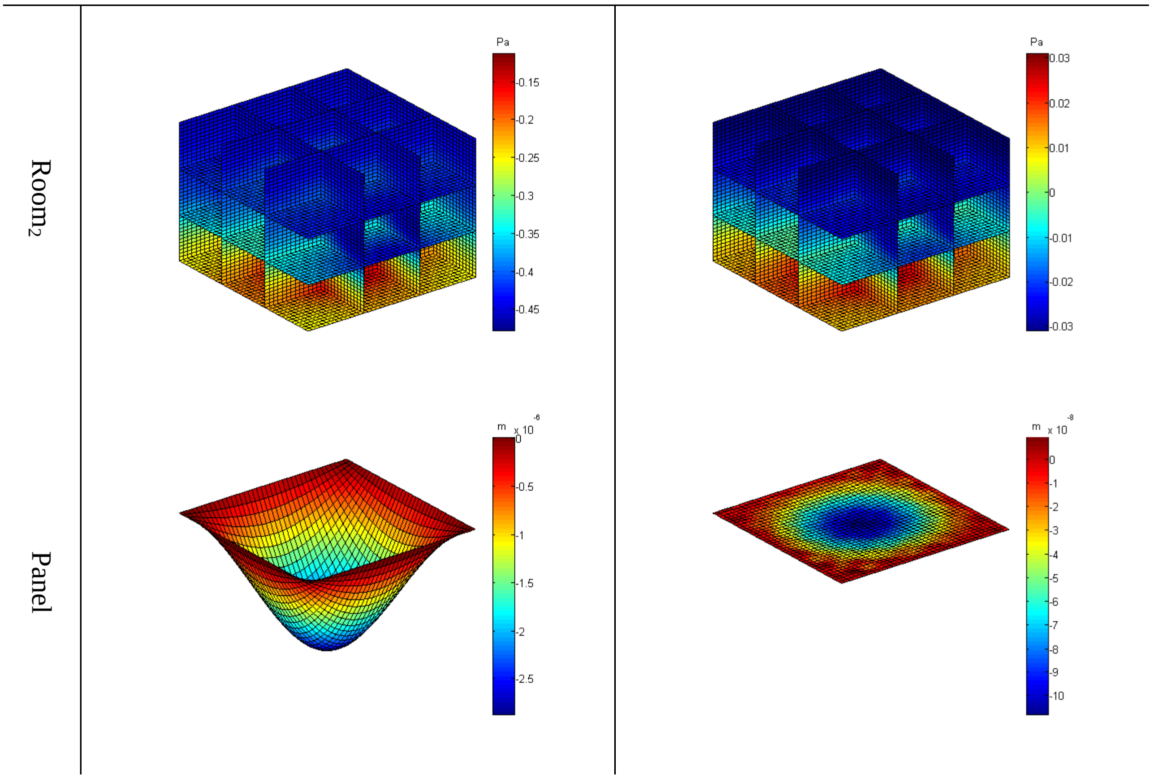
Table 2 – The first six natural frequencies of two rigid acoustical cavities and an single SSSS panel

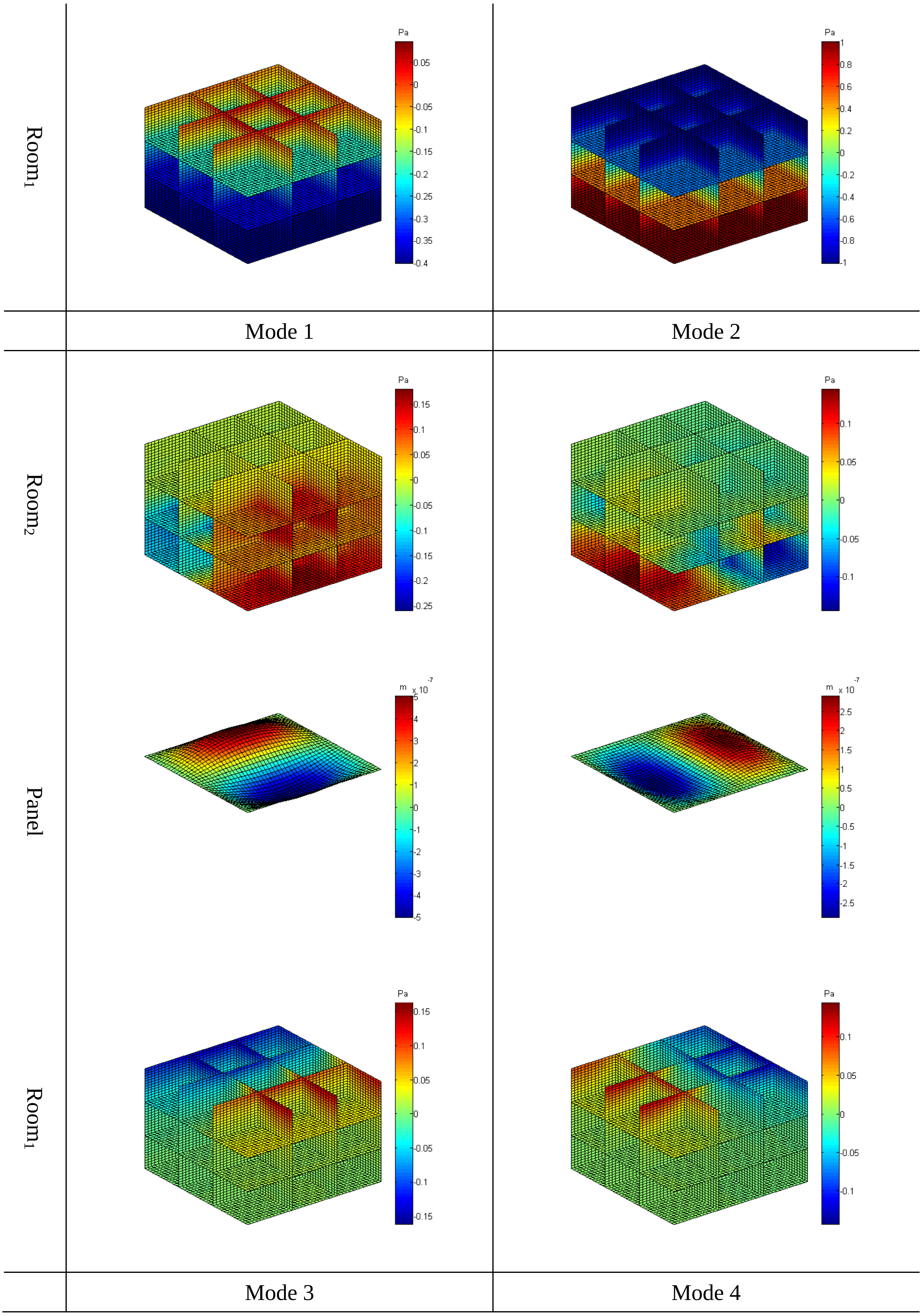
Mode	Room_1		Panel		Room_2	
	Present	FEM	Present	FEM	Present	FEM

1	261.539	261.538	131.986	131.618	377.778	377.778
2	447.368	447.368	283.947	283.178	447.368	447.368
3	518.209	518.209	375.721	375.007	566.667	566.667
4	523.077	523.077	527.751	525.859	585.538	585.538
5	566.667	566.667	537.153	535.845	681.049	681.049
6	624.110	624.110	781.062	777.609	721.976	721.976

Table 3 – The first six natural frequencies of the room-panel-room coupling system

Mode	Present	FEM	Difference (%)	Rigid body mode
1	132.987	133.297	0.23	Panel
2	263.630	263.660	0.01	Room1
3	279.770	280.248	0.17	Panel
4	371.014	372.027	0.27	Panel
5	379.417	379.446	0.01	Room2
6	447.368	447.368	0.00	Room1




Room$_1$
Mode 1
Mode 2
Room$_2$
Panel
Room$_1$
Mode 3
Mode 4
Pa
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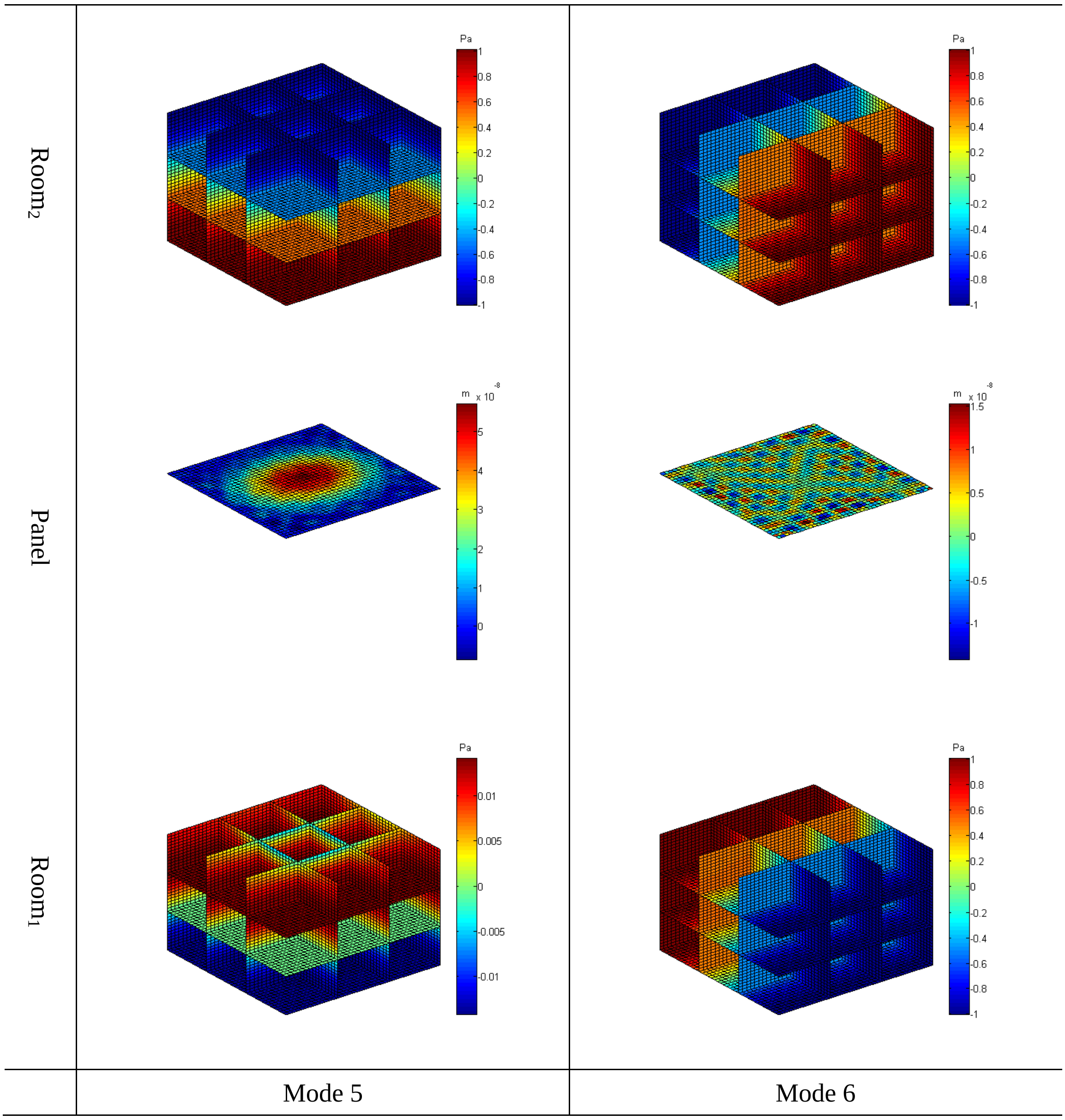

Figure 2 – The first six mode shapes for the room-panel-room coupling system:
(a) the first mode; (b) second; (c) third; (d) fourth; (e) fifth; and (f) sixth.

The data tabulated in Table 2 and Table 3 shows that good agreements can be obtained between the current model and finite element analysis. It can be observed that for each natural frequency of the coupled system, it is very near to natural frequency of each uncoupled component. The structural-acoustic coupling between the cavity, panel as well as cavity leads to slight shift for the resonant frequency. In order to better understand the vibro-acoustic coupling phenomena, the first six mode shapes for each component of room-panel-room coupling system are also plotted in Fig. 2. It can be clearly seen that the eigen-distributions of displacement and sound pressure in such coupling system.

Take room$_1$ and room$_2$ as a whole room with height h=h_1+h_2=1.1 m, the position of the panel in the room can influence the dynamic behavior of the coupled system. To investigate the effect of the position of the panel in the room, five different positions are considered in Table 4.

Table 4 – Natural frequencies for the coupled system with different positions of the panel

Mode	Frequency (Hz)									
position of the plate	h_1=0.95 m h2=0.15m		h_1=0.85 m h2=0.25m		h_1=0.75 m h2=0.35m		h_1=0.65 m h2=0.45m		h_1=0.55 m h2=0.55m	
	Present	FEM	Present	FEM	Present	FEM	Present	FEM	Present	FEM

1	135.693	136.167	133.902	134.245	133.253	133.715	132.987	133.297	132.911	133.371
2	182.736	182.829	202.906	202.957	229.092	229.138	263.630	263.660	279.774	280.404
3	279.201	279.956	279.648	280.187	279.748	280.378	279.770	280.248	309.091	309.091
4	358.762	358.781	370.960	372.829	371.009	372.406	371.014	372.027	312.838	312.909
5	370.648	372.013	400.773	400.790	447.368	447.368	379.417	379.446	371.013	372.817
6	447.368	447.368	447.368	447.368	449.133	449.167	447.368	447.368	447.368	447.368
7	450.592	450.660	449.543	449.582	453.919	453.941	448.953	448.981	448.901	448.931
8	482.659	482.679	490.891	490.908	486.457	486.533	519.151	519.170	522.907	523.728
9	522.686	523.235	522.903	524.126	502.410	502.429	521.875	522.003	533.606	535.627
10	531.339	532.620	533.609	535.522	522.918	524.318	522.910	523.357	543.761	543.761

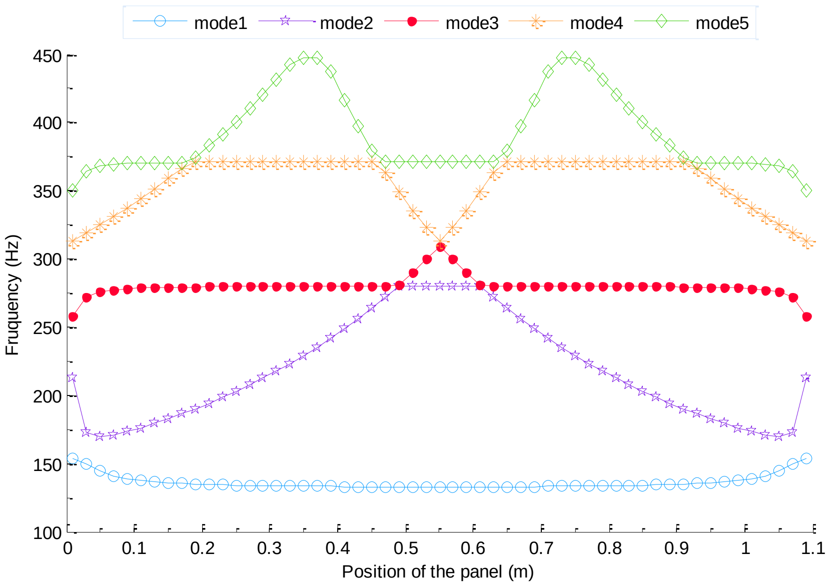

Figure 3 – Natural modes of coupling system with different position of the panel

Figure 3 shows the first five natural frequencies of coupling system varying with different positions of the panel. It can be observed that different positions of the panel do not have a meaningful impact on the first order natural frequency. This is due to the first order rigid body mode is distributed by the panel itself. From the second to fifth order natural frequency, the frequencies are sensitive to the increasing position of the panel, which is caused by the dynamic behavior of different components of the coupled system.

It is further noticed that the boundary conditions of the panel have influence on the mode characteristics of the coupled system. Therefore, the effect of boundary restraining condition of the panel on the sound transmission characteristics of coupled rooms will be analyzed. As mentioned before, it is convenient to obtain all kinds of boundary conditions for the panel through adjusting the stiffness of translational and rotational springs along each panel edge. Different boundary conditions of the panel are considered in Table 5 and Table 6. It is interesting to find that the stiffness of translational panel boundary restrained springs have more influence to the natural frequencies of the coupled system than the rotational springs.

Table 5 – Natural frequencies for the coupled system with different stiffness of translational panel boundary restrained springs

Mode	Frequency (Hz)									
restrained	k=5e9 K=0		k=5e5 K=0		k=5e4 K=0		k=5e3 K=0		k=0 K=0	
stiffness	Present	FEM	Present	FEM	Present	FEM	Present	FEM	Present	FEM

1	132.987	133.297	93.114	90.901	55.435	55.609	18.376	18.846	0.001	-
2	263.630	263.660	153.346	152.571	57.083	58.333	19.575	20.236	0.005	-
3	279.770	280.248	174.108	176.010	61.167	63.082	41.621	41.947	39.479	-
4	371.014	372.027	223.813	227.393	111.318	113.432	89.259	90.167	86.415	87.137
5	379.417	379.446	231.501	238.711	127.202	130.036	110.201	111.340	108.167	109.075
6	447.368	447.368	263.628	264.137	196.150	199.097	184.196	185.569	182.843	184.058
7	448.953	448.981	292.670	302.662	221.205	223.624	209.393	210.679	208.037	209.185
8	519.151	519.170	312.727	319.988	258.477	261.153	247.768	249.075	246.547	247.690
9	521.875	522.003	344.795	355.202	264.104	264.151	264.038	264.082	264.031	264.075
10	522.910	523.357	380.861	380.957	322.001	324.816	315.619	317.380	314.911	316.554

Table 6 – Natural frequencies for the coupled system with different stiffness of rotational panel boundary restrained springs

Mode	Frequency (Hz)									
restrained	*k*=5e9 *K*=0		*k*=5e9 *K*=5e3		*k*=5e9 *K*=5e4		*k*=5e9 *K*=5e5		*k*=5e9 *K*=5e9	
stiffness	Present	FEM	Present	FEM	Present	FEM	Present	FEM	Present	FEM
1	132.987	133.297	187.948	182.065	227.941	227.629	235.148	236.212	235.996	236.988
2	263.630	263.660	264.420	264.404	266.649	266.756	267.750	268.118	267.921	268.255
3	279.770	280.248	342.149	328.739	379.699	379.756	379.741	379.805	379.746	379.800
4	371.014	372.027	379.540	379.583	400.810	398.779	413.002	414.250	414.501	416.248
5	379.417	379.446	447.368	427.828	447.368	447.368	447.368	447.368	447.368	447.368
6	447.368	447.368	447.637	447.368	451.012	451.012	452.006	452.266	452.178	452.511
7	448.953	448.981	449.427	449.397	519.499	519.538	519.589	519.645	519.602	519.656
8	519.151	519.170	519.266	519.296	523.667	523.680	523.686	523.709	523.688	523.709
9	521.875	522.003	523.506	523.335	528.580	525.680	545.114	546.552	546.990	548.672
10	522.910	523.357	566.667	566.667	566.667	566.667	566.667	566.667	566.667	566.667

4. CONCLUSIONS

The investigation of the vibro-acoustics for two coupled rooms through flexible partition has been performed based on the energy principle. Mathematically, it means that there is a complex acoustic-vibro-acoustic coupling problem on the surface of the partition, the vibration of the panel is not only coupled with the sound field of room1, but also coupled with the sound field of room2. There is considerable impedance discontinuity between the panel structure and two cavities. The improved Fourier series method is applied to construct the transverse bending displacement function of the panel structure and the sound pressure functions inside two coupled rooms. The displacement function is derived by the summation of a traditional two-dimensional Fourier series and eight items of supplementary function multiply single cosine function, and the sound pressure function is derived by the summation of a normal three-dimensional Fourier series and a set of supplementary function multiply two-dimensional Fourier series.

Rayleigh-Ritz procedure is employed to derive the eigenvalue equation of the coupled system. The solution can be also extended to derive the forced response functions. High accuracy, stable numerical computation has been observed in the analysis. For demonstration, the natural frequencies of the coupled system of different panel positions and boundary conditions have been studied in detail. The results computed by finite element method using NASTRAN are used as comparison results for the proposed analysis. The comparison work shows that he proposed model are reliable and efficient. The first six mode shapes for the coupled system with SSSS boundary restrained of the panel are given. The method is also applicable to each single component in the system.

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REFERENCES

1. Dowell EH, Voss HM. The effect of a cavity on panel vibration. AIAA Journal. 1963;1(2):476-477.
2. Dowell EH. Transmission of noise from a turbulent boundary layer through a flexible plate into a closed cavity. Journal of Sound and Vibration. 1969;46:238-252.
3. Dowell EH, Gorman GF, Smith DA. Acoustoelasticity: General theory, acoustic natural modes and forced response to sinusoidal excitation, including comparisons with experiment. Journal of Sound and Vibration. 1977;52(4):519-542.
4. Chao CF, Dowell EH, Bliss DB. Modal analysis of interior noise field. Vib Conf of Des Eng Tech Conf; 20-23 September 1981; Hartford, CT, USA 1981. p. 29-56.
5. Lyon RH. Noise reduction of rectangular enclosures with one flexible wall. J Acoust Soc Am. 1963;35:1791-1797.
6. Pretlove AJ. Free vibrations of a rectangular panel backed by a closed rectangular cavity. J Sound Vib. 1965;2:197-209.
7. Guy RW. The response of a cavity backed panel to external airborne excitation: A general analysis. J Acoust Soc Am. 1979;65:719-731.
8. Naryannan S, Shanbhag RL. Sound transmission through elastically supported sandwich panels into a rectangular enclosure. J Sound Vib. 1981;77(2):251-270.
9. Pan J, Bies DA. The effect of fluid-structural coupling on sound waves in an enclosure-theoretical part. Journal of the Acoustical Society of America.1990;87(2):691-707.
10. Pan J, Bies DA. The effect of fluid-structural coupling on sound waves in an enclosure-experimental part. Journal of the Acoustical Society of America.1990;87(2):708-717.
11. Scarpa F, Curti G. A method for the parametric frequency sensitivity of interior acoustic-structural coupled systems. Applied Acoustics.1999;58:451-467.
12. Al-Bassyiouni Moustafa, Balachandran B. Sound transmission through a flexible panel into an enclosure: structural-acoustics model. Journal of Sound and Vibration. 2005;284:467-486.
13. Li YY, Cheng L. Modifications of acoustic modes and coupling due to a leaning wall in a rectangular cavity. J Acoust Soc Am. 2004;116(6):3312-3318.
14. Hong KL, Kim J. Analysis of free vibration of structural-acoustic coupled systems, Part I: development and verification of the procedure. Journal of Sound and Vibration.1995;188(4):561-572.
15. Hong KL, Kim J. New analysis method for general acoustic-structural coupled systems. Journal of Sound and Vibration. 1996;192(2):465-480.
16. Lee JW, Lee JM. Forced vibro-acoustical analysis for a theoretical model passenger compartment with a trunk-Part I: Theoretical of a part. Journal of Sound and Vibration. 2007;299:900-917.
17. Lee JW, Lee JM. Forced vibro-acoustical analysis for a theoretical model passenger compartment with a trunk-Part II: Experimental of a part, Journal of Sound and Vibration. 2007;299:918-932.
18. Du JT, Li WL, Xu HA, Liu ZG. Vibro-acoustic analysis of a rectangular cavity bounded by a flexible panel with elastically restrained edges. J Acoust Soc Am. 2012;131(4):1-12.
19. Li WL, Zhang XF, Du JT, Liu ZG. An exact series solution for the transverse vibration of rectangular plates with general elastic boundary supports. Journal of Sound and Vibration. 2009;321:254-269.
20. Du JT, Li WL, Liu ZG, Xu HA. Acoustic analysis of a rectangular cavity with general impedance boundary conditions. J Acoust Soc Am. 2011;130(2):807-817.