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NOISE TRANSMISSION CONTROL OF DOUBLE PARTITION WINDOWS USING T-SHAPED ACOUSTIC RESONATORS

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Abstract

This paper presents theoretical and experimental noise transmission analyses on a double-partition window with structurally integrated T-shaped acoustic resonators. The study aims at providing a new and practical solution for the building and construction industry to tackle the ever-increasing noise pollution problems for residential areas locating near the airport and high traffic areas. Conventionally, grid-stiffened windows and double-partition windows are used to reduce the noise transmission into rooms. However, the use of grid-stiffeners embedded to glass panels will sacrifice window's vision quality and double-partition windows usually loose the noise insulation efficiency in the low-frequency range. Although classical Hemlholtz resonators may be used to improve the noise control performance of double-partition windows, this treatment is however not practical in such small enclosures due to the bubble-like profile of the Helmholtz resonator. The double-partition windows proposed in this paper incorporate long T-shaped acoustic resonators. The large aspect ratio of the long T-shaped acoustic resonator makes it possible to integrate the resonator into the sash of the windows, which relaxes the space requirement in implementation. A mathematical model describing structural and acoustic interaction between the glass panels, air cavity, and resonators is presented. Based on the model, the optimal location of the resonators is determined considering multiple cavity modes in addition to the targeted mode. Series of numerical simulations are conducted to illustrate the control of a specific resonance peak. The theoretical development provides insight into the noise transmission mechanism, leading to a helpful design tool for generating solutions to reduce the magnitude of noise transmission at cavity resonances, which currently still involve an effort of trial and error. Experimental measurements are also carried out, which are compared with the theoretical predictions.

I. INTRODUCTION

Sound transmitted through windows is one of the major noise sources in rooms. The challenge for noise transmission control through a window is in that the implementation of any noise control techniques should not sacrifice the vision quality of the window, and be economical enough for mass production. Conventionally, grid-stiffened single-leaf windows and double-leaf partition windows are used for this purpose. However, the use of grid-stiffeners will affect window's vision quality and the double-partition windows are tied with an unacceptable noise transmission in the low-frequency range. The original idea of using double partition windows is to induce more mechanical filters or absorbers into windows to filter out or dissipate the undesired noise. A literature review shows that almost all existing double partition windows only use their natural filtering property to reduce noise without involving any vibro-acoustic control strategies. By inserting acoustic control components into the window, one may well attenuate the annoying noise transmission. Studies on the noise transmission control in an Expendable Launch Vehicle (ELV) payload fairing have successfully demonstrated the potential of a low-frequency noise control device - long T-shaped acoustic resonator (TAR) in small enclosures [1-2]. Compared with conventional bulb-like Helmholtz resonators, one of the biggest advantages offered by the TAR is its large aspect ratio, such providing a better energy absorption and easy integration integrated into the host structures (window sash) to reduce space requirement in the implementation.

Detailed experimental measurements of sound transmission through double partition windows were presented by Quirt [3], in which the effects of glass thickness and interpane spacing on transmission loss (TL) were systematically evaluated. Statistical energy analysis was used to model the sound transmission through double walls [4], and a new prediction model of sound transmission loss through double partitions having limited geometric dimensions using patch-mobility method was also constructed [5]. Mason and Fahy [6] proposed the first mass-air-mass coupling model for a double-panel partitions incorporating acoustically tuned Helmholtz resonators (HRs) installed in boundaries. Their focus was to investigate the effects of the resonator parameters on the sound transmission loss through the double-panel partition, thus the elastic properties of the panels and the cavity were ignored. Later, a fully coupled model was developed by Estève and Johnson [7] to investigate noise transmission control into a cylindrical structure using distributed vibration absorbers (DVAs) and HRs, which were tuned to the natural frequencies of the targeted cylindrical structure and acoustic cavity modes, respectively. In this study, a fully structural and acoustic coupled model describing structural and acoustic interaction between the glass panels, air cavity, and integrated resonators is developed. Numerical simulations are conducted to illustrate the control of noise transmission through the double partition window using one TAR located at different points. Finally, experimental measurements are carried out to validate the predicted results.

II. THEORY

A structurally and acoustically coupled system to model a double partition window is shown in Fig. 1. It is assumed that two parallel glass panels are simply supported in a rigid frame which is embedded in an infinite rigid wall. The panels located at $z = L_z$ is denoted as incident panel and another one located at $z = 0$ is refereed to as radiating panel. The two panels have the same geometrical dimensions and the same mechanical properties. Acoustic resonators are integrated into the frame at boundaries. It is also assumed that the normal displacement $w^I(\mathbf{r}_S^I, t)$ of the incident panel is positive inward, and the normal displacement $w^R(\mathbf{r}_S^R, t)$ of the radiating panel and the source volume velocity $q_m(t)$ of the resonator *m* is positive outward, and is assumed all time-dependent variables to be harmonic. There are no sources inside the cavity. The inhomogeneous wave equation governing the pressure fields inside the cavity is

$$\begin{aligned}\nabla^2 p(\mathbf{r},t) - \frac{1}{c^2}\ddot{p}(\mathbf{r},t) = -\rho\sum_{m=1}^{M}\frac{\partial q_m(\mathbf{r},t)}{\partial t}\delta(\mathbf{r}-\mathbf{r}_m) \\ -2\rho_0\frac{\partial^2 w^I(\mathbf{r},t)}{\partial t^2}\delta(\mathbf{r}-\mathbf{r}_S^I) + 2\rho_0\frac{\partial^2 w^R(\mathbf{r},t)}{\partial t^2}\delta(\mathbf{r}-\mathbf{r}_S^R)\end{aligned}, \tag{1a}$$

where $\delta(\mathbf{r}-\mathbf{r}_0)$ is the Dirac delta function; $\mathbf{r}_m$ the aperture-center location of the resonator *m*. Using acoustic impedance and averaged pressure over the aperture to express the volume velocity directed out of the resonator, i.e. $q_m = -p(\mathbf{r}_m, t)/Z_m$, equation (1a) becomes

$$\begin{aligned}\nabla^2 p(\mathbf{r},t) - \frac{1}{c^2}\ddot{p}(\mathbf{r},t) = \rho_0\sum_{m=1}^{M}\frac{\partial p(\mathbf{r},t)}{\partial t}\frac{1}{Z_m}\delta(\mathbf{r}-\mathbf{r}_m) \\ -2\rho_0\frac{\partial^2 w^I(\mathbf{r},t)}{\partial t^2}\delta(\mathbf{r}-\mathbf{r}_S^I) + 2\rho_0\frac{\partial^2 w^R(\mathbf{r},t)}{\partial t^2}\delta(\mathbf{r}-\mathbf{r}_S^R)\end{aligned}, \tag{1b}$$

The dynamic equations of the two panels are:
Incident panel:

$$m^I\frac{\partial^2 w^I(\mathbf{r}_S^I,t)}{\partial t^2} + D^I\nabla^4 w^I(\mathbf{r}_S^I,t) = 2p^{in}(\mathbf{r}_S^I,t) + p_r^I(\mathbf{r},t)\delta(\mathbf{r}-\mathbf{r}_S^I) - p(\mathbf{r},t)\delta(\mathbf{r}-\mathbf{r}_S^I), \tag{2}$$

Radiating panel:

$$m^R\frac{\partial^2 w^R(\mathbf{r}_S^R,t)}{\partial t^2} + D^R\nabla^4 w^R(\mathbf{r}_S^R,t) = p(\mathbf{r},t)\delta(\mathbf{r}-\mathbf{r}_S^R) - p_t^R(\mathbf{r},t)\delta(\mathbf{r}-\mathbf{r}_S^R), \tag{3}$$

where $p^{in}(\mathbf{r}_S^I, t)$ is the incident sound, $p_r^I(\mathbf{r},t)$ the reflection sound of the incident panel, $p_t^R(\mathbf{r},t)$ the transmission sound of the radiating panel, $m^I = \rho_S^I h$, $m^R = \rho_S^R h$, $D^I = (h^I)^3 E^I / 12[1-(\nu^I)^2]$, $D^R = (h^R)^3 E^R / 12[1-(\nu^R)^2]$, and $\nabla^4 = \left(\partial^2/\partial x^2 + \partial^2/\partial y^2\right)^2$. The terms $p_r^I(\mathbf{r},t)$ and $p_t^R(\mathbf{r},t)$ are very small when compared with $p^{in}(\mathbf{r}_S^I,t)$ and $p(\mathbf{r},t)$ in a light fluid medium, thus, they are ignored in the following analysis. The pressure in the cavity and the displacement of the panels can be expanded in terms of acoustic eigenfunctions of the rigid-walled cavity and the structural eigenfunctions *in vacuum*. Substituting the expansion expressions into Eqs. (1b, 2, 3), applying orthogonality properties of the eigenfunctions, and taking into account the viscous damping terms yield the following discretized acoustic and structural equations

Cavity:

$$\left[\ddot{P}_j(t)+2\varsigma_j^C\alpha_j\dot{P}_j(t)+\alpha_j^2P_j(t)\right]\frac{\Lambda_jV^C}{\rho_0c^2}=-\sum_{h=1}^{J}\left[\dot{P}_h(t)\sum_{i=1}^{M}\frac{\varphi_h(\mathbf{r}_i^R)\varphi_j(\mathbf{r}_i^R)}{Z_i}\right]+\sum_{n=1}^{N^I}C_{j,n}^I\ddot{W}_n^I(t)-\sum_{n=1}^{N^R}C_{j,n}^R\ddot{W}_n^R(t), \quad (4)$$

Incident panel:

$$\ddot{W}_n^I(t)+2\varsigma_n^I\beta_n^I\dot{W}_n^I(t)+\left(\beta_n^I\right)^2W_n^I(t)=\frac{2}{M_n^I}P_n^{IN}(t)-\frac{1}{M_n^I}\sum_j C_{j,n}^IP_j(t), \quad (5)$$

Radiating panel:

$$\ddot{W}_n^R(t)+2\varsigma_n^R\beta_n^R\dot{W}_n^R(t)+\left(\beta_n^R\right)^2W_n^R(t)=\frac{1}{M_n^R}\sum_{h=1}^{J}C_{h,n}^RP_h(t), \quad (6)$$

where α_j is the j^{th} acoustic eigenvalue, β_n^I the n^{th} structural eigenvalue of the incident panel, β_n^R the n^{th} structural eigenvalue of the radiating panel, $C_{j,n}^I=\int_S\varphi_j(x,y,z=L_z)\Phi_n^I(x,y)dxdy$ and $C_{j,n}^R=\int_S\varphi_j(x,y,z=0)\Phi_n^R(x,y)dxdy$ the structural and acoustic coupling coefficients, V^C the volume of the acoustic cavity, $\Lambda_j=\int_{V^C}\left[\varphi_j(\mathbf{r})\right]^2dV\Big/V^C$ the modal normalization factor, $M_n^I=m_S^I\int_S\left[\Phi_n^I(x,y)\right]^2dxdy$ and $M_n^R=m_S^R\int_S\left[\Phi_n^R(x,y)\right]^2dxdy$ the n^{th} modal mass, and $P_n^{IN}(t)=\int_S p_i(\mathbf{r}_S^I,t)\Phi_n^I(\mathbf{r}_S^I)dS$ the n^{th} generalized force. All time-independent parts $e^{i\omega t}$ in equations (4-6) can be canceled, and then, from Eqs. (5) and (6), the modal displacement response of the incident and radiating panels can be respectively expressed in terms of modal pressure response. Substituting these modal displacement expressions into Eq. (4), a set of linear equations with the unknown modal pressure responses are obtained. If the incident sound is given, responses of the cavity and the panels can be numerically computed from these linear equations and expressions.

The sound transmission is calculated from

$$TL=10\log_{10}\left(\frac{W_{inc}}{W_{rad}}\right), \quad (7)$$

where W_{inc} is the incident power to the incident panel and W_{rad} the radiated power from the radiating panel. The incident power can be computed from

$$W_{inc}=\frac{\left|P^{in}\right|^2}{2\rho_0c}L_xL_y\cos\theta, \quad (8)$$

where P^{in} and θ are the amplitude and incident angle of the incident sound pressure, respectively. After dividing a baffled vibrating-panel into a number of elements with the same area S, the radiation power from the panel can be calculated using those element radiators [8]

$$W_{rad} = \mathbf{V}^H \mathbf{R} \mathbf{V} \quad \text{and} \quad R_{ij} = \frac{\rho_0 \omega^2 S^2}{4\pi c} \frac{\sin kr_{ij}}{kr_{ij}} \tag{9, 10}$$

where $\mathbf{V}$ is the complex vector of elemental velocities, H denotes the complex conjugate transpose, and $\mathbf{R}$ is a symmetric and purely real matrix associating with acoustic transfer resistances among elemental radiators, which is computed by [8], and r_{ij} is the distance between i^{th} and j^{th} elements.

III. NUMERICAL AND EXPERIMENTAL RESULTS

In this section, the control of sound transmission through a double partition window at the lowest natural frequency of the cavity using one TAR is numerically and experimentally evaluated. The window consists of one frame and two 830 x 830 x 3 mm glass panels simply supported on the frame (see Fig. 1). The interpane spacing is 19 mm. The geometric dimensions and physical parameters used in numerical simulation are listed in Table 1. In simulation, a total of 37 acoustic cavity modes [(l, m, n) = (0 ~ 5, 0 ~ 5, 0), where l, m, n are the node number in x-, y-, and z-direction, respectively.] and a total of 36 structural modes [(p, q) = (1 ~ 6, 1 ~ 6), where p and q are the node number in x-, and y-direction of the incident and radiating panels, respectively.] were considered. The ambient temperature is 20°C, and the speed of sound at this temperature is c = 343.6 m/s. The predicted natural frequencies (below 600 Hz) of the rigid-walled acoustic cavity are listed in the two left columns of Table 2, and the predicted natural frequencies (below 300 Hz) of a simply-supported incident and radiating panels *in vacuum* are listed in the two right columns of Table 2. Before calculating the sound transmission loss, the acoustic impedance of a TAR needs to be determined.

For a typical TAR shown in Fig. 2, when considering the absorptive process within the fluid and at the walls of the resonator, the acoustic impedance of Z at the external aperture can be computed from [1]

$$Z = iz_0 \frac{1 - \dfrac{S_2}{S_1}\tan(\mathbf{k}L_1)\tan(\mathbf{k}L_2) - \dfrac{S_3}{S_1}\tan(\mathbf{k}L_1)\tan(\mathbf{k}L_3)}{S_1\tan(\mathbf{k}L_1) + S_2\tan(\mathbf{k}L_2) + S_3\tan(\mathbf{k}L_3)}, \tag{11}$$

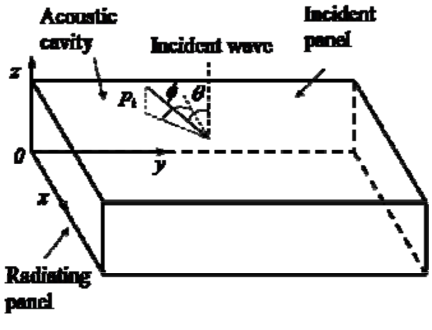

Fig.1 Double partition window system.

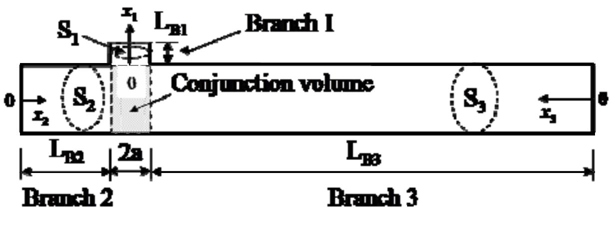

Fig. 2 T-shaped acoustic resonator.

where L_1, L_2, and L_3 are the effective lengths of Branch 1, Branch 2, and Branch 3, respectively, which can be computed using physical lengths by adding end corrections presented in [1], **k** is a complex propagation constant, which can be approximately expressed with a dispersion relation $\mathbf{k} = k\text{-}i\alpha_w$, where $k = \omega/c$ is wave number and α_w is absorption coefficient defined in [9].

Table 1. Geometric dimensions and physical parameters of double partition window.

Parameters	Incident panel	Radiating panel	Acoustic cavity
Dimension: Length $2L_x$ × Width $2L_y$ ×Thickness $2L_z$ (mm)	830×830×3	830×830×3	830×830×19
Young’s modulus (Pa)	60×10^9	60×10^9	
Poisson ratio	0.22	0.22	
Density (kg/m^3)	2373	2373	1.21
Modal damping ratio	0.015	0.015	0.0028

Table 2. Predicted natural frequencies of cavity and simply supported panel.

Acoustic mode No. (l m n)	Acoustic natural freq. (Hz)	Structural mode No. (p q)	Structural natural freq. (Hz)
000	0	11	20.36
010	206.99	12 or 21	50.89
100	206.99	22	81.43
110	292.73	13 or 31	101.78
020	413.98	23 or 32	132.32
200	413.98	14 or 41	173.04
120	462.84	33	183.22
210	462.84	24 or 42	203.58
220	585.45	34 or 43	254.47
		15 or 51	264.65
		25 or 52	295.18

When a plane wave $p_i(\mathbf{r}_S^I, t) = P^{in} e^{i(\omega t - k\sin\theta\cos\phi x - k\sin\theta\sin\phi z)}$ impinged to the incident panel, where $\theta = \pi/6$ and $\phi = \pi/4$ (see Fig. 1), the sound transmission loss through the window without inserting resonators was predicted. It was found that the predicted smallest transmission loss occurred at 208 Hz, which was dominated by the rigid-walled cavity modes (010) and (100) at 207 Hz (the lowest rigid-walled cavity natural frequency.). The TL around 208 Hz is shown in Fig. 3. One TAR, named as TAR_208 with a Helmholtz frequency 208 Hz, was used to improve the transmission loss at 208 Hz. Branch 1 of the resonator was designed using a circular cross-sectional tube with inner diameter 7.7 mm, and Branch 2 and 3 were designed using a square cross-sectional tube having width × height = 14.2 × 14.2 mm. The physical lengths of the designed TAR are: L_{B1} = 20 mm, L_{B2} = 15mm, and L_{B3} = 356.1 mm [1]. It is known that when inserting a resonator into a cavity, the control performance depends on the resonator location, which can be optimally determined through comparing transmission loss predicted at different TAR locations using the present model. In this study, the TL at following locations was predicted: **x** = [0, 99, 178, 257, 336, 415, 494, 573, 652, 731, 830] mm, y = 10 mm, and z = 9.8 mm. The computed transmission losses using the present model are also shown in Fig. 3. Only the curves at $\mathbf{x}$ = [494, 573, 652, 731, 830] mm are shown in the figure due to symmetric geometry. There is no control obtained when the resonator is located at x = 415 mm, y = 10 mm, and z = 9.8 mm (not shown). A 5.4 dB improvement in TL is observed at x = 494 and 830 mm, and 5.3 dB improvement is observed at x = 652 mm. Therefore, the optimal locations are numerically determined at x = 494 and 830 mm. Obviously, x = 830 mm (at one corner) is an optimal location for controlling the acoustic modes (100) and (010). However, for controlling higher order cavity-mode, the four corners may not be optimal locations for installing resonators, and the optimal location must be numerically determined.

Series of experiments were designed to validate the numerical predictions. The window was installed between an anechoic chamber and a reverberant chamber, and the incident sound source (white noise) was located in the reverberant chamber. The measured noise reduction (NR, defined as the total averaged input power including the reflection power from the incident panel over the averaged-output power) without a TAR was firstly measured, and the NR around the smallest rigid-walled cavity resonance frequency is shown in Fig. 4. The measured smallest TL is at 204 Hz. In order to reduce noise transmission at this frequency, a TAR, named as TAR_204 having a Helmholtz frequency 204 Hz, was designed and fabricated. The materials used to fabricate TAR_204 were the same as those used in TAR_208 above. The physical lengths of the TAR_204 are: L_{B1} = 20 mm, L_{B2} = 15 mm, L_{B3} = 322 mm. When changing the location of the TAR_204 among $\mathbf{x}$ = [494, 652, 731, 810] mm, y = 10 mm, and z = 9.8 mm, the NR were measured and were also shown in Fig. 4. Notice that the resonator can not be installed at the corner of (830, 10, 9.8) mm because there are mechanical linkages in the four corners. It is observed that improvement 6.3 dB in NR is obtained at x = 494 and 810 mm, and improvement 6.0 dB in NR is also obtained at x = 652 mm. Therefore, the optimal locations determined by the experimental measurements are the same as those obtained from numerically predicted results.

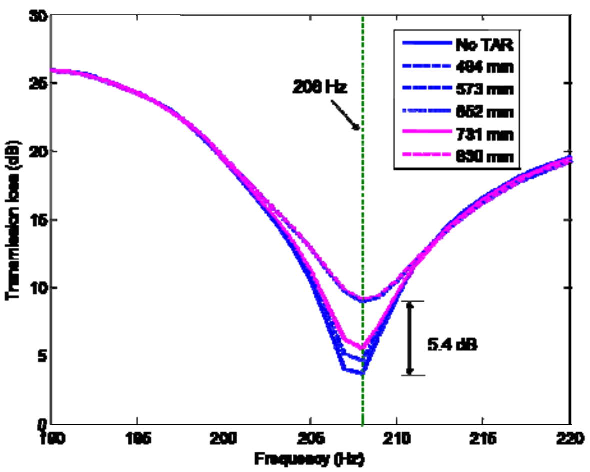

Fig. 3. Predicted noise transmission loss.

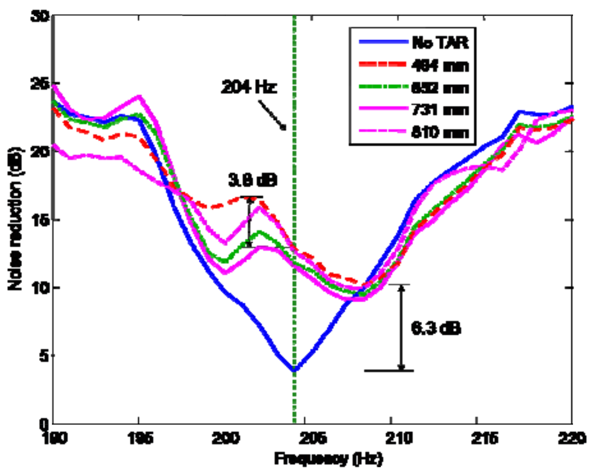

Fig. 4. Measured noise reduction

Comparing the predicted results shown in Fig. 3 and the measured results shown in Fig. 4, it is found that there is about 4 Hz frequency shift between the predicted TL and measured NR. Two coupled-frequencies can be clearly observed from the measured NR curves after inserting a resonator, but no coupled-frequencies are identified in the predicted data, which may be induced by a larger predicted absorption-coefficient α_w [9] than the actual one. After introducing the resonator to their optimal location, the predicted model predicts a 5.4 dB TL improvement, and measurement shows a 6.3 dB NR improvement.

IV. CONCLUSIONS

A fully coupled mathematical model describing noise transmission through a double partition window with integrated acoustic resonators is developed. The model is found to be useful in designing acoustic resonators, and determining the optimal placement of resonators in particular. The concept of using embedded T-shaped resonators in double-leaf windows was validated by laboratory measurement.

Acknowledgements

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