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A numerical investigation on the sound insulation of ventilation windows

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A B S T R A C T

A simulation model is proposed and developed for predicting the sound insulation performance of ventilation windows in buildings, which complies with the laboratory measurement standard ISO 10140. Finite element method (FEM) with verified model definitions is implemented to characterize the airborne sound transmission. An acoustic cavity with rigid-boundaries is used to simulate the diffuse field on the source side of the window, with its diffuseness verified with the pressure field uniformity. On the receiver side, a free field with an infinite baffle is assumed to capture the transmitted sound power. The Sound Reduction Index (SRI) is calculated from the difference between the source and receiving sound power levels in the one-third octave band. Using the proposed model, different ventilation window configurations, consisting of partially open single glazing, double glazing with staggered openings and that with sound absorbers are systematically investigated. Parametric studies are carried out to investigate the effects of various window dimensions and absorber parameters. Simple formulas are proposed for estimating the SRI in the mid-to-high frequency range, providing guidelines for engineering designs. The validity of the numerical model is confirmed by comparisons with full-scale experimental results.

1. Introduction

The need of environmental sustainability calls for the development of natural ventilation technologies to enhance occupant comfort for high-performance buildings. Traditionally, casement windows, top-hung windows and single sliders are commonly adopted window designs, whose structures are simply formed by a single layer of partially open glazing. However, the ventilation openings can easily cause poor noise insulation problem, hampering their uses in densely populated and noisy areas. Hence, the design of building windows capable of achieving natural ventilation whilst warranting required noise mitigation remains an attractive and challenging topic. In 1970s, Ford and Kerry [1,2] first proposed the use of partially open double glazing with staggered inlet-outlet openings to improve the sound insulation. By conducting laboratory and field tests, they claimed the window could provide satisfactory acoustic and ventilation performance. Since then, this simple window construction has aroused continuous research interests [3–11]. For example, Kang et al. [3,4] studied the feasibility of integrating transparent micro-perforated absorbers into the air channel between the double glazing. Through extensive experiments, they demonstrated the acoustic responses were sensitive to the selection of window parameters, showing the need for a prediction model. By adopting active noise cancellation technology, Huang et al. [5] further mitigated the low-frequency noise penetrating through the air channel. More recently, Søndergaard and Olesen [7,8] prototyped a "supply air window" and attempted to optimize its acoustic performance. Tong et al. [9,10] proposed a "plenum window" and conducted both scale-down laboratory and *in-situ* field measurement. It was shown from these experimental works that open double glazing can significantly improve the sound insulation compared to open single glazing. With appropriate treatment of sound absorbing materials, the resultant SRI can even be comparable to a closed single glazing. Nevertheless, a numerical model that can systematically address the need for design and optimization is still lacking. This becomes increasingly important considering the large number of parameters involved in the system design, which, without a reliable simulation model, can hardly be entertained.

Theoretically, the Sound Reduction Index (SRI), as the basic measure of the sound insulation capability of a window, characterizes the proportion of incident sound energy that cannot transmit

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through its surface. To measure the SRI, ISO 10140 standards [12] specify the necessary requirements and practical guidelines for conducting the laboratory experiments. A schematic diagram of the test-rig is shown in Fig. 1, where the test specimen is mounted on a separation wall between a source and a receiving room. Although the test procedure has been well documented, the experiment is only useful for testing the performance of an existing window rather than for seeking a better design, mainly due to the cost of prototypes, experimental reliability and repeatability issues. To solve this problem and potentially shorten the product development cycle, many recent studies have attempted to develop numerical models facilitating the prediction of insulating structures [13–20]. For example, Papadopoulos [13,14] used a virtual laboratory tool to calculate the wall Transmission Loss (TL), where an algorithm was proposed to optimize the shape of the test rooms to obtain adequate diffuseness. Chazot and Guyader [15] formularized a computationally efficient patch-mobility method to predict the TL of a double panel coupled with an air cavity. The simulation repeatability issue caused by the variation of room dimensions and source locations was discussed by Dijckmans and Vermeir [17]. Unfortunately, despite the numerous works found on closed structures, simulations on open windows are scarce, if not inexistent, to the best knowledge of the authors.

The aim of this study is to develop a numerical model for predicting the acoustic performance of open windows, with an attempt to systematically address the effect of changing window parameters. To comply with ISO standard, the source field is modeled as a large acoustic cavity with rigid boundaries, for simulating a diffuse room condition [21,22]. The diffuseness is verified with the spatial uniformity of the pressure field within the domain using a proposed theoretical formulation. As for the radiation field on the receiver side, a free space with an infinite rigid baffle is assumed to capture the transmitted sound power, which mimics an anechoic chamber in the experiment [3,15]. The sound power levels on the source and receiving side of the window, characterized by the acoustic properties of the two fields, respectively, are obtained to calculate the SRI of the window in one-third (1/3) octave frequency band. Detailed descriptions of the proposed simulation model are presented in Section 2.

Based on the proposed numerical model, the SRI characteristics of typical ventilation window configurations will be investigated. An open single glazing is illustrated in Fig. 2(a), where the opening refers to the area which is physically open, allowing for free air passage. In practical implementations, the window can operate either by sliding or pivoting to control the degree of the opening. Note that the two operating methods will not be distinguished in this study. Instead, the dominating effect of changing the opening size will be systematically investigated. Fig. 2(b) and (c) illustrates

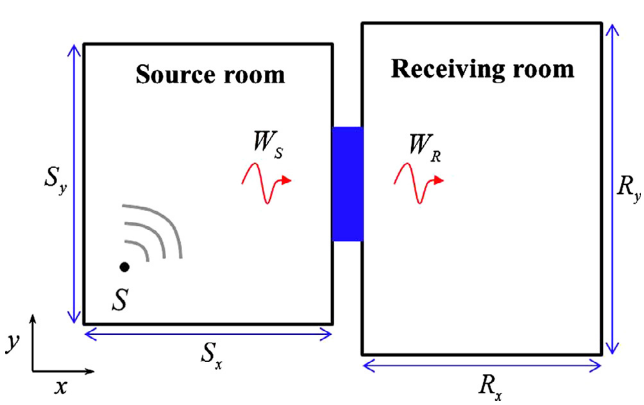

Fig. 1. Experimental evaluation of the SRI of a window, as specified by ISO 10140.

two open double glazing configurations with rigid surfaces or with sound absorbers inside. The sound absorbing material shown in Fig. 2(c) uses a piece of transparent micro-perforated panel (MPP) with honeycomb backing cavity [23]. The real three-dimensional window (3D) configurations can be considered as simple extrusions from the two-dimensional (2D) cross-sections. By assuming the sound transmission is mainly determined by the opening size and the open cavity resonances in the longitudinal and vertical directions, 2D simulations only simulating a window cross-section are performed in Section 3. The effect of changing window geometries and adding sound absorbers will be systematically discussed. Finally, an experimental validation is carried out to validate the proposed numerical model, showing its effectiveness for practical designs.

2. Simulation model

2.1. Diffuse source room

ISO 10140 suggests the use of a reverberant room to excite the test structure, so that the incident sound energy is uniformly distributed over the surface of the specimen [12]. This also enables the incident power to be characterized by averaging the sound intensity inside the source room. A large rigid-walled acoustic cavity is usually adopted. This section proposes a theoretical formulation to check whether an adequate diffuseness has established for the source room used in the simulation.

Let us consider a rectangular cavity with rigid boundary conditions as sketched in Fig. 3, which intends to simulate a diffuse source room for a two-dimensional analysis. The room dimension $S_x \times S_y$ is chosen as 5 m × 6 m, with an aspect ratio of $2^{1/3} = 1.2$ as suggested by Ref. [13]. The window to be tested is mounted on the wall at $x = 5$ m, and a sound source S is placed near the opposite corner to the test element.

For harmonic analysis conducted in the frequency domain (with time-dependent term $e^{j\omega t}$ being omitted), the Helmholtz equation governing the sound pressure distribution can be written as:

$$\nabla^2 p_c(x,y) + k^2 p_c(x,y) = q e^{j\varphi}\delta(x_s, y_s), \tag{1}$$

where p_c is the sound pressure at any point inside the cavity, k is the wavenumber with $k = \omega/c_0$, ω and c_0 are the angular frequency and the sound speed in air, respectively. $j = \sqrt{-1}$ and t is time. The air absorption effect can be accounted by using a complex sound speed $c_0\sqrt{1+\eta j}$, with η being the damping loss factor. For the source term, q describes the source amplitude and φ the phase angle; δ is the Dirac delta and coordinates (x_s, y_s) specify the source location.

Using the modal expansion approach, the pressure field can be decomposed as:

$$p_c(x,y) = \sum_m a_c^m \varphi_c^m(x,y), \tag{2}$$

where a_c^m is the m-th modal amplitude of the cavity; φ_c^m is the mode shape function. For the rigid rectangular-shaped cavity, the following analytical expression for the acoustic modes can be applied:

$$\varphi_c^m = \cos(k_x x)\cos(k_y y) = \cos\left(\frac{m_x \pi}{S_x}x\right)\cos\left(\frac{m_y \pi}{S_y}y\right), \; m_x, m_y = 0, 1, 2, \ldots, \tag{3}$$

where k_x and k_y are the wavenumbers in the x and y directions, S_x and S_y are the cavity dimensions, while m_x and m_y are the modal indices, respectively. The resonant frequencies are $f_m = c_0\sqrt{(m_x/S_x)^2 + (m_y/S_y)^2}/2$. Note that the mode shape function for a complex-shaped cavity can be obtained by using FEM [24].

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