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Laboratory measurement of the acoustical and airflow performance of interior natural-ventilation openings and silencers

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ABSTRACT

This paper discusses laboratory measurements of the acoustical and airflow performance of interior natural-ventilation openings and silencers ('ventilators'). The objective was to create and characterize a purpose-built test facility, and use it to measure the combined acoustical and airflow performance of a number of ventilators of interest, to understand and optimize it, and provide design guidelines to practitioners. The paper discusses the characterization of ventilator performance, and methods and theory for measuring it. The design and performance of a purpose-built, two-room laboratory facility are described. The facility was used to investigate the performance of a non-acoustical grille, an acoustical louver, slot ventilators, crosstalk silencers and a novel door-vent silencer. The results identify a number of best practices for successful ventilator design: non-acoustical grilles should be avoided; the addition of a glass-fiber absorptive liner to the surface adjacent to a slot ventilator increases acoustical performance by STC 3–6; acoustically-lined crosstalk silencers can be very effective – the straight configuration is best and performance increases with the length of the flow path; acoustical liners should be at least 50-mm thick. A prototype door-vent silencer showed very promising performance, but needs to be optimized.
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1. Introduction

The use of natural, as opposed to mechanical, ventilation is one method for making buildings more sustainable – that is, reducing energy consumption, and providing a healthier, more comfortable and productive environment for their occupants. Natural ventilation uses the stack effect to drive ventilation air through a building. This involves large openings in partitions, which prove detrimental to the noise isolation between the spaces. A number of studies have investigated this problem theoretically and experimentally, and how to resolve it by the design of natural-ventilation silencers. A great deal of literature exists on the airflow performance of natural-ventilation systems. However, the bulk of it focuses on external building façades. Research and professional practice that consider interior cross-ventilation airflows commonly describe the airflow by way of a discharge coefficient and equivalent opening area, or the values of equivalent measures, such as the effective free area. See Ref. [1] for a full review of the background and literature behind this issue.

There is a need for a greater understanding of interior natural-ventilation openings, and of silencers implemented to improve their acoustical performance. Here, natural-ventilation openings and silencers are collectively referred to as 'ventilators'. This paper discusses experimental aspects of a study [1] undertaken to satisfy this need. It provides a brief summary discussion of the characterization of ventilator performance, the definition of a combined sound and airflow optimization performance metric, and the theory of measuring ventilator performance; a more comprehensive discussion of these issues is published elsewhere [1,2]. It then discusses experimental aspects of the study – in particular, the design and testing of a purpose-built laboratory facility, and measurements made on ventilators installed in it. The objective was to create and characterize a purpose-built test facility, and use it to measure the combined acoustical and airflow performance of a number of ventilators of interest, to understand and optimize it and provide design guidelines to practitioners.

2. Performance characterization

As discussed fully in [1,2], the combined acoustical and airflow performance of ventilators was described by the open area ratio, $OAR = EOA_f/EOA_s$ where EOA_f is the equivalent open area for airflow and EOA_s is the equivalent open area for sound. With reference to Fig. 2 in [2], which presents the theory and methods in detail, acoustical performance is measured following the ASTM

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E90-09 standard [3] in a two-room sound-transmission facility, The acoustical performance of the ventilator is defined by its frequency-varying transmission loss TL_v – the power-transmission coefficient τ_v expressed in decibels – which can be converted into values of sound-transmission class (STC) [4] and EOA_s. Assuming the sound fields in the two rooms are diffuse, diffuse-field theory is used to show that [5]:

$$TL_v = L_1 - L_2 + 10\log\left(\frac{S_v}{A_2}\right) \tag{1}$$

in which L_1 and L_2 are the reverberant sound-pressure levels in the source and receiver rooms, respectively, S_v is the ventilator area and A_2 is the receiver-room sound-absorption area calculated from the measured reverberation time. As discussed in detail elsewhere [1,2], since transmission loss varies with frequency, to express TL_v as a single number value the frequency components were weighted by a speech-source spectrum and that of the hearing sensitivity of a human listener (A-weighting) to obtain the transmitted speech spectrum ($-TL_{A\_speech}$). The equivalent open area for sound (EOA_s) is then [1,2]:

$$EOA_s = S_v\tau_{A\_speech} = S_v \cdot 10^{\frac{-TL_{A\_speech}}{10}} \tag{2}$$

Again referring to Fig. 2 in [2], and following the ASTM E779-10 standard [6], the airflow performance of a ventilator is also measured in a two-room facility and is typically described by its discharge coefficient C_d [7], from which EOA_f is calculated. Assuming high-Reynolds-number flow and C_d independent of flow rate it can be shown that:

$$EOA_f = S_v\frac{C_d}{0.61} \tag{3}$$

The OAR optimization parameter for ventilators is simple to use and based on common, standardized measurement and analysis techniques. A simple aperture has a value of one; higher values indicate better performance, lower values worse performance. However, it is useful to introduce 'specific' equivalent open areas for sound and flow as non-dimensional performance metrics that are normalized to, and therefore independent of, the ventilator area S_v:

$$SEOA_s = \frac{EOA_s}{S_v} \quad \text{and} \quad SEOA_f = \frac{EOA_f}{S_v} \tag{4}$$

It can be seen from Eqs. (2) and (4) that the specific equivalent open area for sound is equal to the transmission coefficient and, from Eqs. (3) and (4), that the specific equivalent open area for flow is equal to the normalized discharge coefficient.

When using these performance metrics, it is important to remember the assumptions being made, the most significant of which are that the room sound fields are diffuse, and that the equivalent open area for flow is independent of flow rate. A detailed discussion of these limitations of the methods can be found elsewhere [1,2].

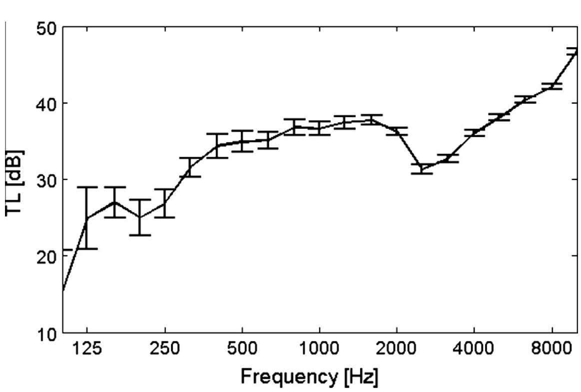

Fig. 1. Measured third-octave-band transmission loss (TL) of the laboratory-facility partition, with 95% confidence limits.

Fig. 2. Non-acoustical test grille.

3. Measurement methods

3.1. Acoustical performance

The laboratory design and operation, detailed below, took into account guidelines outlined in ASTM E90-09 [3], which are based on diffuse-field source and receiver rooms separated by a partition containing the test ventilator, and on measuring the sound-pressure level difference between the rooms. First a full partition was constructed in the laboratory to determine the transmission loss and flanking limits, characterizing the maximum sound isolation obtainable between the two rooms. Average sound-pressure levels were obtained by energy-averaging sound-pressure levels measured at nine positions in the room. Each measurement was a ten-second average. Spot measurements were made instead of scanning measurements, because the operator cannot be in the room. Reverberation times were averaged over measurements made at the nine positions in the receiver room.

3.2. Airflow performance

Flow rate was measured using a blower door [Model 2000, Retrotec Inc., Everson, WA, USA; www.retrotec.com], a calibrated fan unit designed for testing the air-tightness of buildings, as per ASTM E779-10 [6]. It provides a method for calculating the flow rate based on the difference between the ambient pressure and the pressure at a tap in the fan, as well as the pressure differential across the fan. The flow rate (Q in m^3/s) is measured at a number of different pressures; a log-linear regression is used to fit the flow rate as a function of pressure, and determine confidence intervals for the curve fit. The associated expression is:

$$Q = C\Delta p^n \tag{5}$$

in which C and n are the flow coefficient and flow exponent, respectively. Letting $x = \ln(\Delta P)$ and $y = \ln(Q)$, linear regression can be used to determine C and n from the variance and covariance of x and y [6]. The standard deviations of n and $\ln(C)$ can also be found [6].

To obtain accurate results, any obvious flow paths exiting the room, besides the ventilator, are blocked. Then the ventilator is completely blocked for the first set of measurements, to allow calculation of the flow rate exiting the room through paths other than the ventilator (the 'leakage'). Measurements are then repeated with the ventilator open, to allow calculation of the airflow exiting all flow paths. If the flow rate with the ventilator

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