Applied Thermal Engineering 50 (2013) 868–876

Contents lists available at SciVerse ScienceDirect

Applied Thermal Engineering

journal homepage: www.elsevier.com/locate/apthermeng

Heat transfer and pressure drop in spacer-filled channels for membrane energy recovery ventilators

Jason Woods*, Eric Kozubal

National Renewable Energy Laboratory, Jason Woods, 52/213-17, 1617 Cole Blvd, Golden, CO 80401, USA

HIGHLIGHTS

► We investigated support spacers for membrane energy recovery ventilators (ERVs).
► We calculated and measured heat transfer and pressure drop for several spacers.
► New spacer designs show improved performance over simple spacers common in ERVs.
► Tests showed a transition to unsteady flow for velocity ranges expected in ERVs.
► We evaluated the use of common heat transfer performance metrics for ERV spacers.

ARTICLE INFO

Article history:
Received 23 February 2012
Accepted 8 June 2012
Available online 23 August 2012

Keywords:
Membrane spacers
Heat transfer
Mass transfer
Enhancement
Polarization
Energy recovery ventilator

ABSTRACT

This article investigates various support spacers for airflow through membrane-bound channels in energy recovery ventilators (ERVs) to enhance heat and mass transfer. Although liquid flow through membrane-bound channels has been extensively investigated, little work has looked at airflow through these channels. This article presents theoretical pressure drop and heat transfer for an open channel and for simple triangular corrugation (or plain-fin) spacers, which are common in heat exchangers and in some ERVs. It then presents the experimental pressure drop and heat transfer for two new corrugated mesh spacers, with one spacer in three orientations. Results indicate that these can improve heat transfer with little pressure-drop penalty compared to the triangular corrugation spacers. Results also show that unsteady flow occurs in the mesh spacers once a certain flow rate is reached. The optimal spacer depends on the application, which is shown with a cost savings estimate for a hypothetical ERV. Simpler performance metrics that do not require cost estimates can be used to compare two spacers, as long as their limitations are considered.

1. Introduction

We investigated spacers for enhancing heat and mass transfer for airflow in membrane-bound channels of heating, ventilation, and air conditioning (HVAC) devices. A common device is the membrane energy recovery ventilator (ERV) [1–4], which is commercially available from several manufacturers (e.g., ConsERV, RenewAire, NuAire, Fantech). Membrane ERVs consist of alternating channels of exhaust air and ventilation air separated by flat-sheet membranes, which are permeable to water vapor but nearly impermeable to air. During the winter, the ventilation air recovers sensible and latent energy from the exhaust air and returns it to the space; in the summer the exhaust air removes this energy from the ventilation air.

ERVs require support spacers to maintain air channel geometry, because the pressure on the exhaust side at one location is not, in general, the same as that of the ventilation air on the other side of the membrane. Another application requiring support spacers is liquid desiccant dehumidification using membranes, where a hygroscopic salt solution (liquid desiccant) absorbs moisture from the air through the membrane [5]. A support spacer maintains consistent air channel geometry while several forces act on the membrane.

We are interested in these spacers for two reasons: (1) the spacer increases the pressure loss of the airflow, which requires higher fan power and operating cost; and (2) the spacer influences the airside heat and mass transfer coefficients between the membrane surface and the bulk flow. If the spacer increases mixing, and therefore heat and mass transfer, it reduces the ERV size and initial cost, because

* Corresponding author. Tel.: +1 303 384 6189.
E-mail address: jason.woods@nrel.gov (J. Woods).

1359-4311/$ – see front matter © 2012 Published by Elsevier Ltd.
http://dx.doi.org/10.1016/j.applthermaleng.2012.06.052

the overall energy transfer is dominated by these airside resistances. Zhang [6] showed that the airside boundary layers were roughly 99% of the heat transfer resistance for laminar flow through open 2-mm channels. For mass transfer, the improved vapor permeability of polymer membranes compared to paper cores [2,7–9] has reduced the membrane mass transfer resistance. Zhang [6] estimated that the airside resistance is about 25% of the mass transfer resistance, while Min and Su [10] estimated the airside resistance at 10–35%. Using the membrane resistance from [11], this would increase to roughly 75%. So, although the membrane is still a significant part of the mass transfer resistance, the airside resistance is becoming more important.

Traditional heat transfer enhancements protrude from the wall and are difficult to implement when the wall is a membrane. Support spacers do not attach to the membrane and can be designed to produce these heat and mass transfer enhancements. Many researchers have investigated spacers for liquid flows in membrane-bound channels, focusing on how these spacers affect pressure drop and mass transfer in pressure-driven processes [12–24], and heat transfer in membrane distillation [25–28]. Little research has been done on heat and mass transfer for airflows in spacer-filled channels of membrane modules. While the fluid mechanics between liquid flows and airflows is similar, the different pressure requirement necessitates a different spacer design. ERV fans provide 100–300 Pa, which is orders of magnitude less than the pressure provided by liquid pumps.

Our objectives are to determine: (1) the relative performance of various airside spacers for membrane HVAC applications; (2) the transport mechanisms (e.g., unsteady flow or turbulence) that must be included in any computational fluid dynamics (CFD) simulations to accurately model the flow around these spacers; and (3) the applicability of commonly used heat and mass transfer enhancement metrics to ERV spacers. The article presents theoretical and experimental results for heat transfer and pressure drop for three spacers over a range of Reynolds numbers (Re). We compare simple triangular corrugation spacers commonly used in ERVs, with alternative, more complicated spacers. The comparison is based on their Darcy friction factors (f) for pressure drop and Colburn j factors (j) for heat transfer. When thermal conduction through a spacer is near zero, invoking the heat and mass transfer analogy enables the extension of heat transfer results to mass transfer. We then compare the spacers using heat and mass exchanger metrics from the literature based on the tradeoff between pressure drop (parasitic energy use) and heat transfer (reduced size or larger capacity). Finally, we evaluate these metrics with a more realistic, yet simple, calculation of energy-related cost savings for a hypothetical ERV.

2. Methods

Three spacers were considered (Table 1 and Fig. 1). Spacer 1 is a triangular (plain-fin) corrugation that is common in membrane ERVs and heat exchangers. These spacers are typically made from a polymer, such as polypropylene. We consider polypropylene and also hypothetical materials of zero conductivity and infinite conductivity. The simple geometry of spacer 1 enables j and f to be calculated with equations from the literature, and they are not measured experimentally. Spacers 2 and 3 are alternative designs for HVAC devices. Their complex geometries require experimental measurements. To validate the test setup, we also measured the performance of an open channel with no spacer.

Table 1
List of spacers and their properties. Filament size and pitch are indicated in Fig. 1. For spacer 1, we consider three conductivities: zero, polypropylene (typical for an ERV), and infinite.

	Spacer 1	Spacer 2	Spacer 3
Supplier	n/a	AIL Research	Permatron
Material	Varies	Aluminum	Aluminum
Thickness (mm)	3	3	3.175
Corrugation pitch (mm)	8	6	9
Porosity (open volume)	0.89	0.98	0.95
Filament size, d_f (mm)	0.2	0.2	0.9

The factors, j and f, were calculated (no spacer, spacer 1) and measured (no spacer, spacers 2 and 3) as a function of the channel Re, which is defined as:

$$\mathrm{Re} = \frac{\rho_{\mathrm{air}} V d_{\mathrm{h}}}{\mu_{\mathrm{air}}} \tag{1}$$

with ρ_{air} and μ_{air} the air density and dynamic viscosity, d_{h} the channel hydraulic diameter, and V the superficial velocity. This velocity is:

$$V = \frac{\dot{m}_{\mathrm{air}}}{\rho_{\mathrm{air}} A_{\mathrm{x\text{-}sec}}} \tag{2}$$

where $\dot{m}_{\mathrm{air}}$ is the air mass flow rate and $A_{\mathrm{x\text{-}sec}}$ the empty-channel cross-sectional area. Tests were performed for $300 < \mathrm{Re} < 800$, which is a typical ERV operating range. As defined, Re is the independent variable (independent of spacer geometry) and f and j the dependent variables (dependent on spacer geometry and Re). This enables easier comparison between f and j of different spacers for the same mass flow rate.

2.1. Theory

2.1.1. Pressure drop

For the open channel, $f = C_0/\mathrm{Re}$, where C_0 is 96 for parallel plate channels. The channels used in the experiments have a height of 3 mm and width of 250 mm, which leads to $C_0 = 94.8$ [29]. For spacer 1, f was calculated with theoretical correlations for laminar flow through triangular channels [30]. The triangular channel friction factor (f_{tri}) is $C_0/\mathrm{Re}_{\mathrm{tri}}$, where $\mathrm{Re}_{\mathrm{tri}}$ is the Reynolds number based on the triangle hydraulic diameter, and C_0 depends on the angles of the triangle ($C_0 = 51.7$ for spacer 1). The f_{tri} was converted to the friction factor for spacer 1 (f_1) based on the channel Re (Eq. (1)) by equating the pressure drops for both cases:

$$\frac{f_1}{2}\rho V^2 \frac{L}{d_{\mathrm{h,channel}}} = \frac{f_{\mathrm{tri}}}{2}\rho V_{\mathrm{tri}}^2 \frac{L}{d_{\mathrm{h,tri}}} \tag{3}$$

where L is the channel length. Some algebra leads to:

$$f_1 = f_{\mathrm{tri}} \frac{d_{\mathrm{h,channel}} V_{\mathrm{tri}}^2}{d_{\mathrm{h,tri}} V^2} \tag{4}$$

2.1.2. Heat transfer

The j factors for an open channel and for spacer 1 were calculated with constant-temperature Nusselt number (Nu) correlations. For the open channel, we used a developing flow correlation from Bejan [31]. For spacer 1, we used relations from Zhang [32] to calculate the Nu for adiabatic fins (non-conductive), fins of a typical spacer (polypropylene, 0.15 W m^{-1} K^{-1}), and assumed isothermal fins (infinitely conductive). These were derived as follows. The