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Research Paper

Experimental investigation and numerical modelling of a compact wet air-to-air plate heat exchanger

S. Lowrey *, Z. Sun

Department of Physics, University of Otago, PO Box 56, Dunedin, New Zealand

HIGHLIGHTS

- A compact air-to-air heat exchanger is investigated under two-phase conditions.
- Thermal effectiveness is investigated for a range of by-pass configurations.
- Moisture extraction rate optimisation is investigated for tilt and airflow rate.
- A numerical model is validated for duct by-pass and aspect ratio configurations.

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ABSTRACT

Air-to-air plate heat exchangers are widely used for domestic and industrial scale HVAC applications. The wide use of plate heat exchangers makes their control and optimisation critical for improving overall system performance. The application of plate heat exchangers has recently been demonstrated in domestic scale dehumidifier and heat pump clothes dryer systems. In the case of the domestic dehumidifier, the plate heat exchanger, referred to as an evaporator economizer, reduces the sensible load of moist air, and raises the drying efficiency of the system. However, they have shown to contribute to the latent cooling process also. There is little information regarding control and optimisation of air-to-air plate heat exchangers operating under wet operating conditions typical of domestic scale dehumidification and heat pump clothes drying.

In this work, a plate heat exchanger was designed and constructed to experimentally investigate several different conditions for optimizing the moisture extraction rate for application of a plate heat exchanger as an energy recovery device. The test configurations include varied tilt angle, thermal effectiveness control, varied duct aspect ratio and varied air volume flow rate and moist air conditions that are applicable to domestic scale dehumidification systems. The research findings show that a modest tilt angle of up to $-20°$, relative to a reference setting of the hot-side face area being parallel to the horizontal, the pressure drop can be reduced by 33% with a corresponding decrease in the moisture extraction rate of 8%. Furthermore, a study of different ducting configurations shows that deactivating cold-side ducts, while fixing the active hot-side, allows for a greater range of hot-side temperature difference control, while maintaining a relatively high MER compared with deactivating hot-side ducts while keeping the active cold-side ducts fixed.

The experimental data obtained in this work has been used to extend the validity of a recently developed numerical model for a wet air-to-air plate heat exchanger expanding its working range of hot-side and cold-side moist air inlet conditions, airflow conditions and ducting configurations and duct aspect ratios. This model provides for a quantitative approach when engineering emerging advanced domestic scale dehumidification and heat pump clothes drying systems.

1. Introduction

Air-to-air plate heat exchangers (PHEs) are used in a wide number of HVAC applications at the domestic and industrial scale [1]. PHEs are used in condensing clothes dryers [2] and are being extended to other domestic systems such as the domestic dehumidifier and heat pump clothes dryer [3,4]. As of 2016, the

* Corresponding author.
E-mail address: sam.lowrey@otago.ac.nz (S. Lowrey).

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Nomenclature

A	duct height [m]
A	area [m^2]
A_{FACE}	face area of heat exchanger [m^2]
b	duct width [m]
β	A_{DUCT}/A_{FACE} [–]
C_f	friction factor [m^{-2}]
D_h	hydraulic diameter [m]
e_v	contraction loss factor [–]
$\mathbf{F}$	forces that the system exerts at the control volume surface [N]
F_f	net force acting on control volume boundaries [N]
h	specific enthalpy [J kg^{-1}]
h_m	mass transfer coefficient [m s^{-1}]
$\mathbf{I}$	identity matrix [–]
$\mathbf{j}$	molecular mass flux [kg m^{-2} s^{-1}]
$\mathbf{J}_q$	total heat flux vector [W m^{-2}]
m	mass [kg]
$\dot{m}$	mass flow rate [kg s^{-1}]
$\hat{\mathbf{n}}$	unit normal vector [–]
N_{CS}	number of cold-side ducts [–]
N_{HS}	number of hot-side ducts [–]
Nu	nusselt number [–]
P	pressure [Pa], Perimeter [m]
Pr	Prandtl number [–]
$\boldsymbol{q_c}$	pure heat flow vector [W m^{-2}]
$\dot{Q}$	heat transfer rate [W]
$\dot{Q}^{(m)}$	enthalpy flux accompanying mass transfer [W]
Re	Reynolds number [–]
S	surface
$\boldsymbol{S}$	extra stress tensor [Pa]
t	time [s]
T	temperature [°C, K]
T_{db}	dry-bulb temperature [°C]
T_{wb}	wet-bulb temperature [°C]
ΔT	temperature difference [–]
u, v	x-component, y-component of velocity
$\mathbf{v}$	velocity [m s^{-1}]
x, y, z	spatial coordinates [m]
ef	enhancement factor [–]
ρ	density [kg m^{-3}]
ϕ	relative humidity [–]
ω_k	mass fraction [kg-k/kg-mixture]

Subscripts

CS	cold-side
db	dry-bulb
D/DUCT	duct
k, a, v, w	species-k, dry-air, water-vapour, liquid-water
HS	hot-side
P	control volume central node
s	surface
S	interface
W	wall
wb	wet-bulb

U.S. Department of Energy is looking to adopt greater energy conservation standards around domestic dehumidifier products [3], and have indicated that they expect pre-cooling air-to-air PHE to improve domestic dehumidification technology [3]. An energy efficient hybrid heat pump clothes dryer was recently demonstrated which incorporated an air-to-air PHE [4] showing that PHEs are extending into new domestic drying technology applications.

Air-side energy recovery, in conjunction with the refrigerant evaporator, is a well established means of increasing dehumidification capacity [1] as it has the potential to reduce the sensible cooling load of the refrigerant evaporator, promoting greater latent cooling of the humid air at this component [5–7]. This can be carried out using heat-pipes [7,8] or air-to-air plate heat exchangers [9–12]. However, the application of air-side energy recovery in domestic scale equipment has only recently been tested experimentally for low household temperatures [10–11] where these systems are typically operated in New Zealand and Britain [13,14]. In this application, it has been shown that the ratio of the PHE condensation rate to the total system condensation rate can be as high as 19% in some ambient conditions [10]. To describe the performance of a wet economiser in this application, a numerical model was developed [15], however, the model was restricted to a single air volume flow rate and a single duct geometry.

In this experimental study, the experimental data has been obtained and used to extend the validity of the numerical model developed in reference [15] for a range of airflow rates, ambient temperature and humidity conditions, duct configurations (that allow various degrees of air flow by-pass) and new plate heat exchanger duct aspect ratios. In addition, we have experimentally investigated means for improving condensate drainage and reducing air-side pressure drop in an air-to-air plate heat exchanger under two-phase flow conditions. There is little research on the effect of tilting an air-to-air plate heat exchanger [16] and this investigation looks at a range of PHE tilt conditions for reducing pressure drop and for enhancing the vapour condensation. In addition, a wide range of duct blocking configurations have been studied to control the PHE's thermal effectiveness and enhance the condensation rate. Finally, different duct aspect ratios were studied to determine how they perform under relatively high latent cooling conditions. We emphasise that the validated numerical model results show that this model can be applied for the design and investigation of an air-to-air PHE for a wide range of HVAC applications but with a specific focus on domestic scale applications such as dehumidification and heat pump clothes drying technology. The size of the compact plate heat exchanger investigated in this study, and the conditions of moist air investigated in this study, have been guided by our previous work into geared dehumidification technology. In particular the inlet conditions tested on the hot and cold sides of the plate heat exchanger used in this work are guided by the measured data presented in reference [11] in which a plate heat exchanger of a similar size was used as an evaporator economizer.

2. System and methods

2.1. Heat exchanger design and construction

In this experimental study, an air-to-air PHE was designed and constructed to investigate the performance of domestic scale PHEs. In order to keep the new PHE's capacity similar to the PHE investigated in reference [10], a similar number of HS and CS ducts were used and the heat exchanger internal heat transfer wall area was kept similar to the previous PHE's wall area, which was ~0.023 m^2. However, to reduce the complexity of the numerical modelling of the new PHE, the turbulence inducing ribs in the