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An optimized crossflow plate-fin membrane-based total heat exchanger

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ABSTRACT

In this paper, a test rig was constructed to measure the sensibility and enthalpy effectiveness of a plate-fin total heat exchanger (PFTHE). Combined with finite element method, the influence of airflow distribution was analyzed, and an improved PFTHE was subsequently introduced. Utilizing the test rig, the study measured the sensibility and enthalpy effectiveness of the improved PFTHEs with air deflectors and different air spreader plates. Results show that the sensibility and enthalpy effectiveness of the PFTHE are optimized at 17.4% and 7.8% airflow rates, respectively.

1. Introduction

At present, many engineering systems advocate the use of less energy while maintaining the same functions and exceeding the performance required by previously designed systems. This concept is especially true for heating, ventilating, and air conditioning (HVAC) systems. Buildings and their HVAC systems should be energy efficient while satisfying the ever-increasing demand for better indoor air quality, performance, and environmental preservation. Research shows that membrane-based total heat exchanger (MTHE) is an effective means to reduce the energy consumption of ventilation systems [1–3]. The MTHE allows ventilation systems to reduce energy consumption given the heat and moisture recovery of exhausted air of the buildings [4,5]. Literature indicates that that the MTHE is the most effective device for energy-saving, commercial/industrial buildings [6,7] and residence [8,9]. How to increase the effectiveness of heat and moisture exchanger has been the key issue in this field. Most studies focus on improving the intensification of heat and mass transfer. In the aspect of structure design of channel, the use of triangular plate-fin (PF) structure had significant improvement on heat transfer efficiency compared with the parallel plates [2,10]. Recent research shows that with corrugated plate-fin, the heat transfer was improved with little pressure-drop penalty compared to the triangular plate-fin [11]. In the other hand, the most effective mean to improve the efficiency is the utilization of novel membranes to enhance moisture and heat transfer. Different type membrane materials [12–15] was used in MTHE. The influence of properties [16,17], selective permeation [18], spacing and thickness [19] of the membrane were studied experimentally. The influence of adsorption heat, the moisture and thermal resistance of membrane were also studied theoretically [20–22]. Two excellent review articles were presented on recent membrane progress on HVAC by Woods and Zhang, respectively [23,24]. Some researchers have delved on the effects of flow arrangements by using quasi-counter flow design instead of the crossflow [25]. Noticeable advantages include high efficiency, ease of construction and no moving parts, has have made quasi-counter flow design of exchanger become a research hotspot [26–30]. However, it is still a long term for the use of this type exchanger in residential/commercial buildings.

The above researches mainly focuses on the core structure or membrane materials, whereas relatively limited attention has been devoted to the optimization of air channel of MTHE. In this study, we optimized crossflow PF MTHE with air deflector and air spreader plate. The performance of the optimized MTHEs was measured, hence indicated a significant improvement in the efficiency of heat exchange.

2. Classic exchanger

The crossflow with PF is the most commercial MTHE design in the market because of its simplicity and the ease of duct seal

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Nomenclature

Letters

c	air specific heat
C	concentration
D	mass diffusion coefficient
h	enthalpy
H	convective heat transfer coefficient
k	convective mass transfer coefficient
L	latent heat
M	cross-sectional mass transfer rate
m	average mass flow rate of air
N	nonuniform coefficient
S	humidity
T	temperature
v	velocity
δ	thickness of membrane
η	effectiveness
λ	heat conductivity coefficient of membrane
ρ	air density
U	uncertainty

subscript

e	exhaust air
f	flesh air
h	enthalpy
i	inlet air
j	flesh air flow direction (x) or exhaust air flow direction (y)
m	membrane
n	flesh in or flesh out
n	total number of channels
o	outlet air
s	latent
T	sensible
w	water

required for MTHE systems. Fig. 1 shows the crossflow PF MTHE design. The fresh air and exhaust air was injected in the MTHE. The core of MTHE will provide a crossflow arrangement, therefore, the heat and moisture transfer will occurred over the plate-fin and membrane. As a result, the amount of energy will be recovered.

2.1. Test rig

Test rig is used in measuring the steady state heat and moisture transfer through the THE by considering inlet and outlet temperatures, humidity, and airflow rates. The sensibility and the latent effectiveness are the performance indices. A schematic of the test rig is shown in Fig. 2. The rig is divided into three rooms, in which the left and the right rooms are intended for simulating outdoor and indoor conditions and comprised two 310 mm × 280 mm × 3000 mm ducts. With air conditioning and electric heating coils, the temperature and humidity can be controlled and maintained even under hot and humid ambient weather conditions. The exchanger in the middle room is connected to the ducting works with four inlet/outlet vents by flanges. Data of temperature/moisture, pressure, and air speed in the duct can be detected by different sensors. During the experiment, equal airflow rates are stored for the four ducts, which are changed from 300 m^3/h to 600 m^3/h by variable speed blowers to generate different air velocities. The airflow under such conditions is laminar, with Reynolds numbers not exceeding 8800. From these preparatory undertakings, the test rig is considered reliable. Heat and mass balance is a time-consuming process requiring more than 12 h to complete under one testing condition.

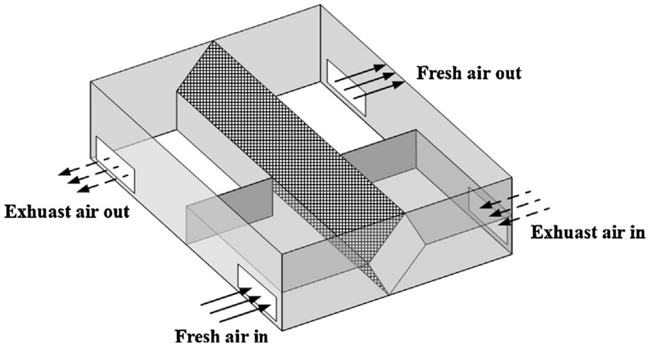

Fig. 1. Depiction of a crossflow PF MTHE design.

After measuring inlet and outlet temperature and humidity, the temperature, latent and enthalpy effectiveness are calculated by

$$\eta_T = \frac{T_{fo} - T_{fi}}{T_{eo} - T_{fi}} \quad \eta_S = \frac{S_{fo} - S_{fi}}{S_{eo} - S_{fi}} \quad \eta_h = \frac{h_{fo} - h_{fi}}{h_{eo} - h_{fi}} \tag{1}$$

where η_T, η_S, and η_h are the sensibility, latency, and enthalpy effectiveness, respectively; T is temperature; S is humidity; h is enthalpy; subscripts *fo*, *fi*, and *eo* refer to flesh out, flesh in, and exhaust out, respectively.

2.2. Uncertainty analysis

In the test rig, the accuracy of temperature sensor is ≤0.2 °C, the accuracy of moisture sensor is ≤2%. For 95% confidence interval, the uncertainty of three different temperature sensors are $U_{Tfo} = 0.203\,^\circ\mathrm{C}$, $U_{Tfi} = 0.204\,^\circ\mathrm{C}$, $U_{Teo} = 0.206\,^\circ\mathrm{C}$ and the uncertainty of three different moisture sensors are $U_{sfo} = 0.038$ g/kg, $U_{sfi} = 0.036$ g/kg, $U_{seo} = 0.045$ g/kg. Therefore, the uncertainty of enthalpy are $U_{hfo} = 0.110$ g/kg, $U_{hfi} = 0.114$ g/kg, $U_{heo} = 0.149$ g/kg.

Based on Eq. (1), the uncertainty of temperature, latent and sensibility effectiveness can be calculated by

$$\frac{U_X}{\eta_X} = \sqrt{\left(\frac{1}{X_{fo} - X_{fi}} \times U_{Tfo}\right)^2 + \left(-\frac{1}{X_{fo} - X_{fi}} \times U_{Tfi}\right)^2 + \left(-\frac{1}{X_{eo} - X_{fi}} \times U_{Teo}\right)^2} \tag{2}$$

where, X indicates temperature (T), latent (S) and sensibility(h) separately. Therefore, the uncertainty of sensibility effectiveness (U_T/η_T) is 7.73%, the uncertainty of latency effectiveness (U_S/η_S) is 5.19%, the uncertainty of enthalpy effectiveness (U_h/η_h) is 2.78%.

2.3. Mathematical control equation

To discuss further the experimental results, a physics model was constructed (Fig. 3). In which, the flesh airstream and exhaust airstream flow through in a cross-flow pattern. The model was formulated based on the following assumptions:

(1) Both the heat and the mass transfers are steady.
(2) The physical properties of the air fluid and membrane are constant.
(3) Heat conduction and vapor diffusion in the two airstreams in the directions parallel to the channel walls are negligible compared to the bulk convection.

According to Liu [3], with Peclet number (Pe) bigger than 2, the effect of axial air conduction/diffusion can be totally negligible. In our present work, with $Pe = Re \times Pr = c_{pa}\rho_a u d_e/\lambda_a = 7400$, the assumption (3) is valid. The mathematical control equations