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Experimental investigation and performance evaluation of a mixed-flow air to air membrane enthalpy exchanger with different configurations

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© 2019 Elsevier Ltd This paper presents an experimental investigation and performance evaluation of a mixed-flow air to air membrane enthalpy exchanger with different flow configurations. The moisture diffusivity, water contact angle and elastic modulus of five porous membranes were first experimentally determined using a wet cup test method, a contact angle meter and the tensile test, respectively. The five porous membranes investigated consisted of two Polyvinylidene difluoride membranes with the mean pore diameters of 0.22 μm (PVDF22) and 0.45 μm (PVDF45) respectively, two Nylon membranes with 0.1 μm and 0.45 μm pore sizes respectively and one Polyethersulfone (PES) membrane with a 0.1 μm pore size. The optimal membrane was then identified and used to fabricate an enthalpy exchanger. Lastly, the performance of the enthalpy exchanger with different mixed-flow configurations was experimentally investigated. The results showed that the PVDF45 membrane offered the highest moisture diffusivity ($1.91 \times 10^{-6} \pm 2.42 \times 10^{-8} \text{ m}^2/\text{s}$) with hydrophilic surface and a relatively high elastic modulus ($4.97 \times 10^8 \pm 3.86 \times 10^7 \text{ Pa}$). It was shown that the enthalpy exchanger with Z-shape flow of the supply air stream and U-shape flow of the exhaust air stream at an entrance ratio of 0.25 offered the best thermal performance as the sensible, latent and total effectiveness increased by 12.3%, 15.1% and 14.6% respectively, when the flow rate was 0.3 l/s, as compared to that of the pure cross flow. The same flow configuration was able to recover the highest energy among all configurations investigated.

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Experimental investigation and performance evaluation of a mixed-flow air to air membrane enthalpy exchanger with different configurations

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Abstract: This paper presents an experimental investigation and performance evaluation of a mixed-flow air to air membrane enthalpy exchanger with different flow configurations. The moisture diffusivity, water contact angle and elastic modulus of five porous membranes were first experimentally determined using a wet cup test method, a contact angle meter and the tensile test, respectively. The five porous membranes investigated consisted of two Polyvinylidene difluoride membranes with the mean pore diameters of 0.22 μm (PVDF22) and 0.45 μm (PVDF45) respectively, two Nylon membranes with 0.1 μm and 0.45 μm pore sizes respectively and one Polyethersulfone (PES) membrane with a 0.1 μm pore size. The optimal membrane was then identified and used to fabricate an enthalpy exchanger. Lastly, the performance of the enthalpy exchanger with different mixed-flow configurations was experimentally investigated. The results showed that the PVDF45 membrane offered the highest moisture diffusivity ($1.91 \times 10^{-6} \pm 2.42 \times 10^{-8} \text{ m}^2/\text{s}$) with hydrophilic surface and a relatively high elastic modulus ($4.97 \times 10^8 \pm 3.86 \times 10^7 \text{ MPa}$). It was shown that the enthalpy exchanger with Z-shape flow of the supply air stream and II-shape flow of the exhaust air stream at an entrance ratio of 0.25 offered the best thermal performance as the sensible, latent and total effectiveness increased by 12.3%, 15.1% and 14.6% respectively, when the flow rate

25 was 0.3 l/s, as compared to that of the pure cross flow. The same flow configuration was able
26 to recover the highest energy among all configurations investigated.

27 **Keywords:** Enthalpy exchanger; moisture diffusivity; mixed-flow configuration;
28 experiments; performance evaluation

29 **Nomenclature**

J	mass flux (kg/m ² .s)
G	weight change (kg)
t	time during which weight change occurred (s)
A_c	test cup area (m ²)
w	absolute humidity (kg/kg)
D_m	moisture diffusivity (m ² /s)
R_m	moisture diffusion resistance (m ² .s/kg)
m	mass flow rate (kg/s)
P_c	power consumption (W)
T	temperature (°C)
h	specific enthalpy (J/kg)
h_{heat}	convective heat transfer coefficient (kW/m ² .K)
Q	volume flow rate (m ³ /s)
cp	specific heat (kJ/kg.K)
A_t	membrane area (m ²)
D_h	hydraulic diameter (m)
A_{cs}	channel cross section area (m ²)
P_d	channel perimeter (m)
Nu	Nusselt number
h_{mass}	convective mass transfer coefficient (m/s)

Sh Sherwood number 30

D_a moisture diffusivity of air (m²/s)

Δp pressure drop (pa)

Greek letters

ε effectiveness

ρ_a air density (kg/m³)

λ air thermal conductivity (kW/m.K)

Subscripts

in inside the cup

out outside the cup

min minimum

m mean

lm logarithmic

s supply/sensible

e exhaust

i inlet

o outlet

l latent

t total

31

32 **Abbreviations**

MDR moisture diffusion resistance (m².s/kg)

TER total energy recovered (W)

RH relative humidity (%)

33 **1. Introduction**

The undesirable increase in energy demand of the building sector could result from the unsustainable design and intensive usage of Heating, Ventilation and Air Conditioning (HVAC) systems [1]. The primary energy demand in the building sector is around 40% of the total world energy, among which about 50% was consumed by HVAC systems [2, 3]. Therefore, improving energy efficiency of HVAC systems is becoming a priority for building designers and practitioners to promote building energy efficiency and sustainability [4-6].

Air-to-air energy recovery ventilators (ERVs) are among the environmentally friendly devices that can be used to save energy in buildings and reduce greenhouse gas emissions [7]. ERVs can recover sensible and latent energy in the exhausted air stream to pre-condition the incoming outdoor air stream [8], providing opportunities for energy savings. Membrane enthalpy exchanger (MEE) is one type of ERVs using semi-permeable membranes to separate the air streams to exchange heat and moisture simultaneously from one air stream to the other air stream [9, 10]. MEEs have been widely used in building applications as they can offer less embodied energy, simplicity, low contamination crossover and high performance [11].

The membrane properties and the air flow configurations inside the enthalpy exchanger are among the critical parameters that affect the performance of the MEE [12]. Membrane properties such as moisture diffusivity and module of elasticity are important for membrane selection [13]. Both latent effectiveness and total effectiveness are influenced by the moisture diffusivity of the membrane used to fabricate MEEs [14, 15]. On the other hand, the pressure difference across the semi-permeable membrane may cause membrane deflection, which will affect the flow distribution and decrease the exchanger performance. The study from Larson et al. [16] showed that significant membrane deflection can occur by applying a moderate pressure difference on both sides of the membrane. It was shown that membrane deflection was inversely proportional to the modulus of elasticity. The deflection increased with the decrease of the modulus of elasticity, resulting in a decrease in the performance of the MEE. In general,

the maldistribution due to membrane deflection can be mitigated by using membranes with high modulus of elasticity [17].

Another design parameter that can influence the performance of the MEE is the flow configuration. It is well known that the best performance can be generally achieved by using the counter flow configuration [18-21]. However, it is usually difficult to achieve such configuration in practice due to the fact that the inlet and outlet ducts will be overlapped with each other at the same side, and sealing will be problematic [19-21]. Therefore, a new flow configuration named mixed-flow configuration (i.e. hybrid flow) has been developed to reduce the manufacturing challenges of the counter flow configuration and to maintain the high effectiveness of the MEE.

The mixed-flow configuration could consist of two or more basic flow configurations (i.e. parallel, cross and counter). Various flow combinations can be used to obtain the hybrid flow. For instance, Nassif et al. [22-24] experimentally and numerically investigated the performance of a MEE with a Z-shape flow configuration. It was found that the annual energy consumption of an air conditioning system integrated with this Z-shape MEE was decreased by 8% as compared to a conventional HVAC system. Zhang [25] developed a quasi-counter flow membrane-based total heat exchanger. The results showed that the effectiveness of this quasi-counter flow configuration was between the cross-flow configuration and counter-flow configuration. It was further revealed that the flow would be divided into three zones, including two cross flow zones near the inlets and the outlets respectively, and a counter flow zone at the middle part. Al-Waked et al. [26] numerically investigated the thermal performance of a MEE with various flow configurations including Z-shape, L-shape and quasi-counter flow under turbulent flow conditions. It was found that the quasi-counter flow could be the best option among different flow configurations investigated.

Based on the above review, it can be seen that there are a few hybrid flow configurations for air to air MEEs that have been proposed. However, a detailed experimental investigation and performance comparison among different flow configurations has not been thoroughly conducted. This paper presents an experimental investigation of the performance of a mixed-flow air to air membrane enthalpy exchanger with six different hybrid flow configurations and the pure cross flow configuration. The moisture diffusion resistance, the water contact angle and the modulus of elasticity of five porous membranes were first evaluated. An experimental setup was then developed and used to investigate the performance of the MEE with various mixed-flow configurations and different air flow rates. The results obtained from this work could be used to guide the optimal design of MEEs.

2. Description of the mixed-flow membrane enthalpy exchanger

The performance of the MEE is highly dependent on the flow configuration and air velocity [26]. Fig. 1 presents the mixed flow configuration of the MEE that was investigated in this study. It mainly consists of a pure cross flow core and four adjustable headers. The core was constructed with semi-permeable membranes, sealing brackets and upper/lower bases. The inlet and outlet spans of each air stream were identical. The ratio between the inlet span to the span of the membrane, named entrance ratio, can be varied as 1, 0.75, 0.5 and 0.25 respectively, as illustrated in Fig. 2. The sealing brackets and bases were made of acrylic with a thickness of 2 mm and 20 mm, respectively. The membranes and sealing brackets were assembled together using a special glue, and then sandwiched by the upper and lower bases to form six channels. The headers were connected to the core using flanges. The MEE was insulated by polyester with a thickness of 15 mm applied to all the external surfaces.

The air leakage of the exchanger was tested under positive and negative pressures as specified in the test procedure of BS EN 308:1997 standard [27]. The moisture leakage to the surrounding was neglected since the acrylic is strongly impermeable to water vapour.

Fig. 1 Schematic of the mixed-flow membrane enthalpy exchanger.

Fig. 2 Illustration of the exchanger headers with four different entrance ratios.

3. Experimental methodology

The overall method used in this study is illustrated in Fig. 3. It mainly consists of four steps including i) membrane selection and characterisation; ii) development of the test facility;

iii) experimental design and execution; and iv) experimental data analysis and performance evaluation. They are briefly described as follows.

Fig. 3 Outline of the research method.

3.1 Membrane selection and characterisation

Membranes used in MEEs should be extremely permeable to water vapour as the vapour pressure difference is relatively small between the exhaust and supply air streams. In addition, the membrane must limit the diffusion of unwanted gases [28]. For this reason, five candidate porous membranes namely (i) Polyvinylidene fluoride (PVDF) membrane with the pore sizes of 0.22 and 0.45 μm , respectively; (ii) Nylon membrane with the pore sizes of 0.1 and 0.45 μm , respectively; and (iii) Polyethersulfone (PES) membrane with a 0.1 μm pore size, were initially selected based on the data of the permeability and selectivity of water vapour over the non-desirable gases (i.e. CO_2 and N_2) reported in the previous studies as listed in Table 1. The selected membranes were then characterised by measuring the moisture diffusivity, water contact angle and elastic modulus to determine the membrane that will be used to fabricate the exchanger core.

Table 1 Permeability and selectivity data for the selected membranes

Membrane	Permeability (Barrer ^a)						Selectivity	
	H_2O	Ref.	CO_2	Ref	N_2	Ref	$\text{H}_2\text{O}/\text{CO}_2$	$\text{H}_2\text{O}/\text{N}_2$
Nylon	275	[29]	0.088	[30]	0.025	[29]	3125	11000
PES	2620	[29]	3.38	[31]	0.25	[29]	775	10480
PVDF22	8542813	[32]	3846	[33]	160	[33]	2221	53393
PVDF45	11329341	[32]	3846	[33]	160	[33]	2946	70808

^a 1 Barrer = $1 \times 10^{-10} \text{ cm}^3 (\text{STP}) \text{ cm}/\text{cm}^2 \text{ s cm}_{\text{Hg}} = 7.5005 \times 10^{-18} \text{ m}^2 \text{ s}^{-1} \text{ Pa}^{-1}$

3.1.1 Moisture diffusivity measurement

The wet cup test method introduced in the standard of ASTM E96 [34] was used as the experimental approach to measuring moisture diffusion resistance, and then calculating the moisture diffusivity of the selected membranes. The principle of the wet cup test method is to maintain a certain vapour pressure difference on both sides of the test specimen and monitor

the weight loss of the cup assembly. Fig. 4 illustrates the apparatus of the wet cup test method used in this study. The tests were operated using a PVC cup with a diameter of 90 mm filled with distilled water. The membrane sample was mounted over the mouth of the cup using flanges. A gasket was applied on the rim to avoid any leakage. The cup assembly was placed in an environmental chamber which can well control the required temperature and humidity. An electronic weighing scale with a resolution of 0.01 g was used to monitor the weight of the cup assembly during the experiments. The environmental chamber was functioned to maintain the conditions outside of the cup based on the test cases defined in Table 2. The relative humidity inside the cup can be considered as 100% while the temperature inside the cup was assumed to be the same as the temperature of the environment chamber. The uncertainties of the temperature and relative humidity sensors used were 0.1 °C and 1%, respectively.

Table 2 Test conditions for the moisture diffusivity measurements

Test No.	T1	T2	T3	T4	T5	T6	T7	T8	T9
Temperature (°C)	27.5	30.0	32.5	27.5	30.0	32.5	27.5	30.0	32.5
Relative humidity (%)	80	80	80	50	50	50	30	30	30

Fig. 4 Test apparatus of the wet cup method.

3.1.2 Water contact angle measurement

The water contact angle (CA) of each tested membrane was measured using a contact angle meter (CAST3, KINO, USA) with an accuracy of $\pm 3^\circ$. CAST®3.0 software for drop shape analysis was implemented to analyse the wetting behaviour of the five porous membranes.

3.1.3 Elastic modulus measurement

The tensile tests were conducted for the five selected membranes using a Shimadzu EZ-S universal test device. The tensile specimens were prepared by cutting the membranes into the rectangular strips with approximate dimensions of 120 mm $\times$ 10 mm. The specimens were attached to the test device with the screw loaded clamps. The height of the specimen between the top and bottom clamps was maintained at 100 mm for all tests. It is worth noting that the nominal thickness of all the tested membranes was 100 μ m. The tests were then undertaken using a 500 N load cell and the samples were strained at a rate of 1 mm/min until the failure occurred. The tensile stress was calculated based on the applied load and the average cross-sectional area. The strain was determined using the original length of the unstrained samples. Based on the recorded data, the corresponding stress-strain curves can be generated and used to determine the elastic modulus of the tested membranes. For each type of the membranes, the tests were repeated for three times and the standard errors were estimated by using one standard deviation method.

3.2 Development of the experimental test facility

Based on the optimal membrane identified in the first step, the enthalpy exchanger core as presented in Section 2 was constructed. An experimental setup, as illustrated in Fig. 5, was then developed to evaluate the hydrodynamic and thermal performance of the mixed-flow MEE. The simplified schematic of this experimental test setup is presented in Fig. 6. This

experimental setup mainly consists of a MEE core, two environmental chambers, a vacuum pump, connecting pipes and measurement instruments.

The experimental setup was designed as a closed loop system. The air in the two air streams was conditioned in the two environmental chambers at the desired conditions and then supplied to the test MEE core. The air vacuum pump was used to recirculate the outlet air to the environmental chamber. The connecting pipes between the test section and the environmental chamber were insulated to minimise any potential heat loss. Four T-type thermocouples were calibrated and mounted at the inlets and outlets of the exchanger to measure the air temperatures. Two variable area flow meters (Dwyer, RMC-103-SSV) equipped with a control valve were installed at the exits of each air stream of the exchanger respectively to measure the air flow rates. The scale range of the air flow meters used was 10-100 l/min. Four Vaisala humidity transmitters (HM110) were installed at the inlets and outlets of the MEE to monitor the relative humidity at each air stream. The pressure drop across the exchanger was measured by a Testo850 differential pressure gauge. During the experiments, the temperature, relative humidity, pressure drop and flow rate of each air stream at the inlet and outlet of the test section were measured every 5 minutes.

Fig. 5 Experimental test setup (1- differential pressure gauge; 2- air flow meter; 3- thermocouples; 4- humidity sensors; 5- data logger in the box; 6- power supply; 7- vacuum pump; 8- control valve; 9- environmental chamber; 10- connecting pipes; 11- exchanger core; 12- computer).

Fig. 6 Simplified schematic of the experimental test setup for the mixed-flow MEE.

3.3 Experimental design and execution

A total of 28 experimental cases were designed and carried out based on the test module of the mixed-flow MEE introduced in Section 2 to examine the performance of the MEE with various mixed-flow configurations. The tests were conducted under four different air flow rates (i.e. 0.3, 0.6, 0.9, 1.2 l/s). As summarized in Table 3, seven air flow patterns were investigated in order to determine the best flow configuration. Fig. 7 illustrates the flow configurations of the seven cases listed in Table 3. In each experiment, the flow rates of the two air streams were maintained the same in order to reduce the pressure difference on the membrane sides. In the flow configurations of Case 1-Case 3 and Case 7, the same entrance ratios for both air streams were used. However, different entrance ratios for both air streams were used for the flow configurations of Case 4-Case 6. All the test cases were conducted based on the same inlet air conditions, that is, the inlet temperature and relative humidity of the exhaust stream were 25 °C and RH 45%, while those of the supply stream were 35 °C and 70% RH, respectively.

Table 3 Summary of the flow configurations investigated

Configurations	Supply channel		Exhaust channel		Flow pattern
	Entrance ratio	Flow shape	Entrance ratio	Flow shape	
Case 1	25%	Z- shape	25%	И- shape	Hybrid flow
Case 2	50%	Z- shape	50%	И- shape	Hybrid flow
Case 3	75%	Z- shape	75%	И- shape	Hybrid flow
Case 4	25%	Z- shape	100%	Straight	Hybrid flow
Case 5	50%	Z- shape	100%	Straight	Hybrid flow
Case 6	75%	Z- shape	100%	Straight	Hybrid flow
Case 7	100%	Straight	100%	Straight	Pure cross flow

Fig.7 Flow configurations of the seven investigated cases.

3.4 Data analysis

The vapour transmission rate (J) across the membrane was calculated by Eq. (1). The corrections for the air gap and the surface resistance were performed using the methods detailed

in the standard [34]. The vapour transmission rate can also be represented as Eq. (2) [35]. The moisture diffusivity (D_m) was calculated using Eq. (3). The sensible, latent and total effectiveness can be calculated by using Eqs. (4), (5) and (6), respectively [36, 37]. The convective heat and mass transfer coefficients were calculated by Eqs. (7)-(12) [38]. The hydraulic diameter, Nusselt number and Sherwood number were calculated by Eqs. (13), (14) and (15), respectively [39]. The total energy recovered (TER) was calculated by Eq. (16) [26] and the fan power consumption (P_c) is determined by Eq. (17) [37], based on the pressure drop measured by the differential pressure gauge.

$$J = \frac{G}{t A_c} \quad (1)$$

$$J = \frac{w_{in}-w_{out}}{R_m} \quad (2)$$

$$D_m = \frac{1}{R_m \rho_a} \quad (3)$$

$$\varepsilon_s = \frac{m_s(T_{si}-T_{so}) + m_e(T_{eo}-T_{ei})}{2m_{min}(T_{si}-T_{ei})} \quad (4)$$

$$\varepsilon_l = \frac{m_s(w_{si}-w_{so}) + m_e(w_{eo}-w_{ei})}{2m_{min}(w_{si}-w_{ei})} \quad (5)$$

$$\varepsilon_t = \frac{m_s(h_{si}-h_{so}) + m_e(h_{eo}-h_{ei})}{2m_{min}(h_{si}-h_{ei})} \quad (6)$$

$$h_{heat} = \frac{\rho_a Q_s c_p \Delta T_m}{A_t \Delta T_{lm}} \quad (7)$$

$$\Delta T_m = \frac{(T_{si}-T_{so}) + (T_{eo}-T_{ei})}{2} \quad (8)$$

$$\Delta T_{lm} = \frac{(T_{si}-T_{eo}) - (T_{so}-T_{ei})}{\ln \left[\frac{(T_{si}-T_{eo})}{(T_{so}-T_{ei})} \right]} \quad (9)$$

$$h_{mass} = \frac{\Delta w_m}{\rho_a A_t w_{lm}} \quad (10)$$

$$\Delta w_m = \frac{(w_{si}-w_{so}) + (w_{eo}-w_{ei})}{2} \rho_a Q_s \quad (11)$$

$$\Delta w_{lm} = \frac{(w_{si}-w_{eo}) - (w_{so}-w_{ei})}{\ln \left[\frac{(w_{si}-w_{eo})}{(w_{so}-w_{ei})} \right]} \quad (12)$$

$$D_h = \frac{4A_{CS}}{P_d} \quad (13)$$

$$Nu = \frac{h_{heat} D_h}{\lambda} \quad (14)$$

$$Sh = \frac{h_{mass} D_h}{D_a} \quad (15)$$

$$TER = m_s (h_{si} - h_{so}) \quad (16)$$

$$P_c = n Q_s \Delta p + n Q_e \Delta p \quad (17)$$

where J is the mass flux across the membrane, G is the weight change of the cup assembly, t is the time during which weight change occurred, A_c is the area of the cup, w_{in} and w_{out} are the humidity inside and outside of the cup respectively, R_m is the moisture diffusion resistance of the membrane, D_m is the moisture diffusivity of the membrane, ρ_a is the air density, ε_s , ε_l and ε_t are the sensible, latent and total effectiveness respectively, m is the mass flow rate, T is the air temperature, w is the air humidity, h is the specific enthalpy, h_{heat} is the convective heat transfer coefficient, Q is the volume flow rate, cp is the specific heat, A_t is the membrane area, h_{mass} is the convective mass transfer coefficient, D_h is the hydraulic diameter, A_{cs} is the cross section area of the channel, P_d is the perimeter of the channel, Nu is Nusselt number, λ is the air conductivity, Sh is Sherwood number, D_a is the moisture diffusivity in air, TER is the total energy recovered, P_c is the power consumption, Δp is the pressure drop across the channel and n is the number of the channels for each air stream.

The uncertainties in the experimental tests can result from the instrument calibration and data acquisition [35]. Uncertainty analysis is essential to evaluating the accuracy of the experiments. The uncertainties of the measured parameters included temperature, relative humidity, flow rate and pressure drop were ± 0.1 °C, $\pm 2\%$, $\pm 2\%$ and ± 0.1 Pa, respectively. The uncertainty (δRy) of each calculated parameter was estimated using the basic root-sum-square as given by Eq. (18) [40].

$$\delta Ry = \frac{\sqrt{\sum_{i=1}^m \left(\frac{\partial y}{\partial x_i} \delta x_i \right)^2}}{y} \quad (18)$$

where m is the number of the independent parameters, and y is the main parameter dependent on the series of the independent parameters x_i . The estimated uncertainties were found to be highly dependent on the operating conditions. However, the highest values of the uncertainties for the calculated sensible, latent and total effectiveness were 3.2% and 5.1%, 5.9%, respectively.

4. Results and discussions

4.1 Results of the moisture diffusivity measurements

During the practical operation of MEEs, the weather conditions such as temperature and humidity may change continuously. It is therefore important to quantify the influence of those parameters on the moisture transfer across the membrane. Fig. 8 shows the variations in the moisture diffusion resistance of the tested membranes with the test conditions specified in Table 2. It can be seen that all the tested membranes showed similar behaviour against the test conditions. The lowest values of the moisture diffusion resistance were offered by PVDF45 membrane while Nylon10 membrane showed the highest values of the moisture diffusion resistance. The moisture diffusion resistance was insensitive to the test temperature. For instance, for the PVDF45 membrane at RH of 80% (i.e. test cases of T1-T3 in Table 2), the moisture diffusion resistance changed from 42.3 to 44.8 $\text{m}^2\cdot\text{s}/\text{kg}$ when the test temperature increased from 27.5 to 32.5 $^{\circ}\text{C}$. On the other hand, the relative humidity difference across the membrane slightly influenced the moisture diffusion resistance. For instance, for the PVDF45 membrane at the temperature of 30 $^{\circ}\text{C}$ (i.e. test cases of T2, T5 and T8 in Table 2), the moisture diffusion resistance increased from 43.6 to 46.3 $\text{m}^2\cdot\text{s}/\text{kg}$ when the relative humidity outside the cup (i.e. environmental chamber) decreased from 80% to 30%. The moisture diffusion resistance of the PVDF22 membrane at the test conditions of T7 was similar to that reported in [41].

Fig. 8 Variations in the moisture diffusion resistance under different test conditions for the five membranes tested.

The moisture diffusivity is another parameter that can directly indicate the moisture transfer capacity through the membrane. Fig. 9 illustrates the variation in the moisture diffusivity under different test conditions for the five membranes tested. It can be clearly seen that the test conditions slightly influenced the diffusivity of each membrane. The mean diffusivity values for the tested membranes together with the uncertainty due to the test conditions are provided in Table 4. The maximum value of the mean moisture diffusivity was $1.91 \times 10^{-6} \pm 2.42 \times 10^{-8} \text{ m}^2/\text{s}$ for the PVDF45 membrane, while the lowest value was $1.41 \times 10^{-6} \pm 2.71 \times 10^{-8} \text{ m}^2/\text{s}$ for the Nylon10 membrane.

Fig. 9 Variations in the moisture diffusivity under different test conditions for the five membranes tested.

Table 4 Mean moisture diffusivity and calculated uncertainties for the five membranes tested.

Membrane	PVDF45	Nylon45	PVDF22	Nylon10	PES10
Moisture diffusivity (m ² /s)	1.91×10^{-6}	1.50×10^{-6}	1.59×10^{-6}	1.41×10^{-6}	1.57×10^{-6}
Uncertainties (m ² /s)	2.42×10^{-8}	1.23×10^{-8}	4.14×10^{-8}	2.71×10^{-8}	3.71×10^{-8}

4.2 Results of the contact angles

Fig. 10 shows the water contact angles of the five membranes. The contact angle below 90° indicated that the membrane surface was hydrophilic. It can be seen that all the tested membranes were hydrophilic. However, they varied in the degree of hydrophilicity based on the contact angle. The PVDF22 membrane yielded the highest average contact angle of 76.2° while Nylon45 membrane exhibited the lowest average contact angle of 40.5°.

Fig. 10 Water contact angles for the tested membranes.

4.3 Results of elastic modulus measurements

The stress-strain curve is usually used to identify the mechanical properties of engineering materials. In this study, the tensile stress-strain curves of the five membranes tested are illustrated in Fig. 11. These stress-strain curves were used to determine the tensile properties including tensile strength, failure strain and modulus of elasticity (Young's modulus). The membrane PVDF22 showed the highest tensile strength of $1.81 \times 10^7 \text{ MPa}$, while the membrane PES10 showed the lowest value ($1.35 \times 10^6 \text{ MPa}$). The failure strain of the membranes PVDF22, Nylon10 and Nylon45 were around 0.08 mm/mm and they are very close to each other. The membrane PVDF45 had the maximum failure strain at around 0.12 mm/mm.

Table 5 summarises the mechanical properties of the five membranes tested. The properties including tensile strength, failure strain and the average values of the elastic modulus with the errors determined by the standard deviation for all the measurements. The elastic

modulus for the PES10 was significantly lower than those for the other membranes. In addition, small differences were observed amongst the values of the elastic modulus of the PVDF22, Nylon10 and PVDF45 which were around 500 MPa.

As a summary, among the membranes tested, the PVDF45 membrane yielded the highest value of the mean moisture diffusivity. PVDF22, Nylon10 and PVDF45 membranes showed relatively high values of elastic modulus with small differences amongst them. Therefore, the PVDF45 membrane was selected to fabricate the membrane enthalpy exchanger core.

Fig. 11 Stress-strain curves of the five tested membranes.

Table 5 Mechanical properties of the five tested membranes

Membrane	PVDF45	Nylon45	PVDF22	Nylon10	PES10
Tensile strength (MPa)	1.67×10^7	4.20×10^6	1.81×10^7	1.58×10^7	1.35×10^6
Failure strain (mm/mm)	11.94%	7.92%	8.16%	8.06%	2.32%
Elastic Modulus (MPa)	4.97×10^8	1.89×10^8	5.20×10^8	5.16×10^8	7.67×10^7
Uncertainties (MPa)	3.86×10^7	1.94×10^7	1.86×10^7	1.63×10^7	2.03×10^6

4.4 Results from the performance evaluation of the mixed-flow MEE

The performance of the mixed-flow MEE was evaluated in terms of thermal performance, hydrodynamic performance and energy performance.

4.4.1 Thermal performance

The temperature and humidity were presented in order to examine the influence of the hybrid configurations on the outlet air properties of each air stream. The temperature and humidity of the outlet air in both supply and exhaust channels are provided in Table 6.

Table 6 Air conditions at the outlet of the exhaust and supply channels under various flow rates and flow configurations

Case		Flow rate (<i>l/s</i>)					Flow rate (<i>l/s</i>)			
		0.3	0.6	0.9	1.2		0.3	0.6	0.9	1.2
1	T_{eo} (°C)	34.1	33.5	33.0	32.4	w_{eo} (g/kg)	25.88	24.68	23.57	22.51
	T_{so} (°C)	25.8	26.4	26.9	27.4	w_{so} (g/kg)	11.87	13.00	14.08	15.09
2	T_{eo} (°C)	33.9	33.3	32.9	32.4	w_{eo} (g/kg)	25.55	24.31	23.35	22.48
	T_{so} (°C)	26.0	26.5	27.0	27.5	w_{so} (g/kg)	12.13	13.29	14.36	15.18
3	T_{eo} (°C)	33.7	33.1	32.6	32.2	w_{eo} (g/kg)	25.17	23.88	23.07	22.24
	T_{so} (°C)	26.2	26.7	27.4	27.7	w_{so} (g/kg)	12.46	13.62	14.77	15.52
4	T_{eo} (°C)	33.5	33.0	32.5	32.0	w_{eo} (g/kg)	24.91	23.80	22.79	21.92
	T_{so} (°C)	26.2	26.8	27.5	28.0	w_{so} (g/kg)	12.53	13.79	14.96	15.98
5	T_{eo} (°C)	33.4	32.7	32.3	31.8	w_{eo} (g/kg)	24.62	23.17	22.27	21.51
	T_{so} (°C)	26.3	26.9	27.5	27.9	w_{so} (g/kg)	12.59	13.81	14.85	15.75
6	T_{eo} (°C)	33.3	32.8	32.4	31.9	w_{eo} (g/kg)	24.49	23.34	22.35	21.79
	T_{so} (°C)	26.4	26.9	27.7	28.1	w_{so} (g/kg)	12.72	13.97	15.14	16.25
7	T_{eo} (°C)	33.2	32.7	32.0	31.5	w_{eo} (g/kg)	23.50	22.30	21.31	20.70
	T_{so} (°C)	26.9	27.2	27.8	28.2	w_{so} (g/kg)	13.70	14.61	15.30	16.00

As expected, the increase in the flow rate resulted in the increase in the outlet air temperature and humidity of the supply air stream and the decrease in the outlet air temperature and humidity for the exhaust air stream. The temperature and humidity followed the same general trends for all seven flow arrangements. The temperatures varied from 34.1 to 31.5 °C for the exhaust air side and from 25.8 to 28.2 °C for the supply air side, while the outlet air humidity changed from 25.88 to 20.70 g/kg for the exhaust channel and from 11.87 to 16.00 g/kg for the supply channel. For the exhaust air stream, the highest temperature and humidity

were found to be 34.1 °C and 25.88 g/kg respectively in Case 1 at the flow rate of 0.3 l/s, while the lowest values of the temperature and humidity were 31.5 °C and 20.70 g/kg in Case 7 at the flow rate of 1.2 l/s. In the supply channels, the trends were opposite as the lowest temperature (25.8 °C) and humidity (11.87 g/kg) were observed in Case 1 at the flow rate of 0.3 l/s, and the highest values (28.2 °C, 16.00 g/kg) were found in Case 7 at the flow rate of 1.2 l/s.

The variations of the sensible, latent and total effectiveness with the change of the flow rate for the investigated configurations are presented in Fig. 12. It can be seen that the flow rate had a significant impact on the sensible, latent and total effectiveness, which all were inversely proportional to the increase of the flow rate. This is because the flow resident time inside the exchanger decreased with the increasing flow rate. The variations of the effectiveness of the hybrid configurations were relatively more noticeable than that for the pure cross flow configuration. For example, the total effectiveness of Case 1 decreased from 89.3% to 71.5% as the flow rate increased from 0.3 l/s to 1.2 l/s. In the meanwhile, the total effectiveness of Case 7 (i.e. pure cross flow) varied from 77.9% to 63.7% (Fig. 12c).

For the seven cases studied, the variation trends of the sensible, latent and total effectiveness were similar. However, the sensible effectiveness was the highest among the three effectiveness under the same flow configuration and flow rate, whilst the latent effectiveness was the lowest one as the mass diffusivity was higher than the thermal diffusivity. The maximum values of the sensible, latent and total effectiveness were 91.2%, 88.9% and 89.3% respectively at Case 1 when the flow rate was 0.3 l/s. However, the lowest values of the three effectiveness were 66.45%, 63.1% and 63.67% respectively at Case 7 (i.e. pure cross flow) when the flow rate was 1.2 l/s.

It can be also seen that the hybrid cases showed a better effectiveness as compared to the pure cross flow. However, the performance enhancement was more obvious in the identical entrance ratio cases (i.e. Cases 1, 2, 3) and the effectiveness increased as the entrance ratio

decreased. For instance, the sensible, latent and total effectiveness of Case 1 increased by 12.3%, 15.1% and 14.6% respectively when the flow rate was 0.3 l/s while those values for Case 4 were 6.8%, 9.3% and 8.8% respectively under the same flow rate, as compared to that of the pure cross flow.

a) Sensible effectiveness

b) Latent effectiveness

c) Total effectiveness

Fig. 12 Variations in the three effectiveness under different flow rates and flow configurations.

The values of the sensible, latent and total effectiveness obtained for Case7 (i.e. pure cross flow) at the flow rate of 1.2 l/s were compared to the results reported in literature [42]. The operating conditions and the exchanger specifications of the current study and the previous study are summarised in Table 7. The sensible, latent and total effectiveness of Case7 in this study were 66.5%, 63% and 63.7% respectively, while those in [42] were 67%, 54.5% and 57.5% respectively. The comparison showed a good agreement for the sensible effectiveness between the two sets of data as the conductive heat transfer through the membrane can be neglected due to the small membrane thickness [35]. However, both the latent and total effectiveness obtained in this study was different from that in the literature as they highly affected by the membrane properties [14, 15].

Table 7 Summary of the operating conditions and the exchanger specifications of the current study and the previous study.

Conditions and specification		
	This study	[42]
Test conditions	T_{si} : 35 °C; RH_{si} : 70%; T_{ei} : 25 °C; RH_{ei} : 45%; Flow rate for single channel: 0.4 l/s	T_{si} : 35 °C; RH_{si} : 70%; T_{ei} : 26 °C; RH_{ei} : 50%; Flow rate for single channel: $\approx$ 0.4 l/s
Flow configuration	Cross flow	Cross flow
Channel length (m)	0.2	0.25
Channel Width (m)	0.2	0.25
Channel height (mm)	2	2
Membrane type	Porous	dense
Moisture diffusivity (m ² /s)	1.91×10^{-6}	$\approx 2.17 \times 10^{-7}$
Membrane adsorption constant	-	2.5
Membrane thickness (mm)	0.1	0.1

The other key parameters, including Nusselt number and Sherwood number were presented in Table 8. It can be seen that the Nusselt number was higher than the Sherwood number for all the cases as the mass transfer resistance was much higher than the heat transfer resistance. On the other hand, both the Nusselt and Sherwood numbers were nearly proportional to the flow rate. For instance, the Nusselt number increased from 4.76 to 5.76 and the Sherwood number increased from 4.3 to 4.79 at Case 1 when the flow rate increased from 0.3 l/s to 1.2 l/s. The highest values of the Nusselt and Sherwood numbers were 5.76 and 4.79 at Case 1 when the flow rate was 1.2 l/s. However, the lowest values of both numbers were 1.99 and 1.66 respectively at Case 7 when the flow rate was 0.3 l/s.

From the above analysis, it can be concluded that Case 1 with a Z-shape hybrid flow of the supply stream and II- shape hybrid flow of the exhaust stream offered the highest thermal performance.

Table 8 Nusselt and Sherwood numbers for the seven flow configurations under various flow rate conditions.

Case	Flow rate (l/s)					Flow rate (l/s)			
	0.3	0.6	0.9	1.2		0.3	0.6	0.9	1.2
1	4.76	5.62	5.74	5.76		4.30	4.71	4.73	4.79
2	3.93	4.85	5.22	5.45		3.60	4.12	4.34	4.51
3	3.24	4.14	4.49	4.83		2.97	3.58	3.91	4.17
4	<i>Nu</i> 3.02	3.93	4.16	4.33	<i>Sh</i>	2.76	3.43	3.65	3.76
5	2.76	3.52	3.92	4.21		2.56	3.09	3.45	3.67
6	2.57	3.50	3.85	4.10		2.41	3.09	3.35	3.57
7	1.99	3.20	3.42	3.66		1.66	2.40	2.86	3.22

4.4.2 Hydrodynamic performance

The hydrodynamic performance of the exchanger was investigated by determining the pressure drop along the flow channel for different flow configurations and various flow rates (Fig.13). In general, the pressure drop increased with the increase of the flow rate as the shear stress on the membrane surface increased. It can be also observed that the entrance ratio influenced the pressure drop negatively. For a given flow rate, decreasing the entrance ratio resulted in an increase in the flow velocity and generating the vortex zones. The recirculation zones disturbed the flow stream and might turn the flow from laminar into turbulent. The turbulent flow featured with a high boundary layer skin friction can increase the drag, and as a result increased the pressure drop.

Fig.13 Variations of the pressure drop of mixed-flow channels under various entrance ratios and flow rates.

4.4.3 Energy performance evaluation

The energy analysis for the studied MEE was conducted in order to understand the benefit of the proposed hybrid configurations. This analysis was carried out based on the recovered energy and consumed energy. The consumed energy was mainly due to the fan power consumption. Fig. 14 presents the variations in the total energy recovered with the variation of the flow rates for all investigated flow configurations. The total energy recovered increased with the increase in the flow rate, and the total energy recovered was in the range of 16.2-59.1 W. The amount of energy recovered by the MEE in Case 1 was 12.8%, 11.6%, 10.1% and 8.4% higher under the flow rates of 0.3, 0.6, 0.9 and 1.2 l/s respectively than that of the test case 7 with the pure cross flow configuration.

Fig. 14 Variations of total energy recovered under different flow rates and various flow configurations.

The power consumed by the fans of the MEE is shown in Fig. 15. The variation trends of the energy consumption were similar to those of the recovered energy as the pressure drop increased with the increase of the flow rate. However, the fan power consumption was significantly less than the total energy recovered. In addition, the entrance ratio and the channel shape did not show a significant impact on energy consumption at low flow rate conditions. To sum up, both the recovered energy and consumed energy were a function of the flow rate and flow configuration. A lower flow rate offered a lower pressure drop while a higher flow rate resulted in a high capacity for energy recovery.

Fig. 15 Variations of the fan power consumption under different flow rates and flow configurations.

As a summary, the thermal performance of the investigated hybrid flow configurations outperformed the pure cross flow configuration. The Z-shape hybrid flow of the supply air stream and II-shape hybrid flow of the exhaust side at the entrance ratio 0.25 offered the maximum values of the thermal performance and the recovered energy.

5. Conclusion

This paper presented the performance evaluation of a mixed-flow enthalpy exchanger through experimental investigation. The measurements of the properties for five porous membranes were first conducted under various test conditions. The values of the moisture diffusivity and the elastic modulus with the uncertainties for the five membranes were presented and evaluated. The results of the measurements showed that the test conditions slightly affected the moisture diffusivity of all the tested membranes. The PVDF45 membrane offered the highest value of the mean moisture diffusivity (i.e. $1.91 \times 10^{-6} \pm 2.42 \times 10^{-8} \text{ m}^2/\text{s}$). The PVDF22 membrane yielded the highest contact angle of 76.2° while Nylon45 membrane exhibited the lowest contact angle of 40.5° . Three membranes (i.e. PVDF22, Nylon10 and

PVDF45) showed the similar elastic modulus of around 500 *MPa* with small differences amongst them. Based on those results, the PVDF45 membrane was employed and used to fabricate a multi-flow configuration membrane enthalpy exchanger core. The developed core allowed to investigate the performance of the enthalpy exchanger under various flow configurations and entrance ratios.

A total of seven different flow configurations were studied including one pure cross flow and six mixed-flow configurations. The results indicated that the thermal performance was improved as the flow rate and entrance ratio decreased. The Z-shape hybrid flow (Case 1) of the supply air stream and H-shape hybrid flow of the exhaust side at the entrance ratio 0.25 yielded the maximum thermal performance as the total effectiveness increased by 14.6% when the flow rate was 0.3 l/s, as compared to that of the pure cross flow. The hydrodynamic performance investigation showed that the pressure drop was inversely proportional to the entrance ratio while the flow rate influenced the pressure drop positively. The energy analysis showed that the total energy recovered increased as the flow rate increased. The fan power consumption followed similar behaviours as that of the recovered energy however the energy consumption was much lower than the recovered energy for the same flow configuration and flow rate. A high flow rate was generally preferable for high energy saving applications. The results obtained from this study can be used to develop effective enthalpy exchangers for energy recovery ventilation systems.

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