Numerical and experimental study of a novel three-fluid membrane dehumidification method applied to spacecraft humidity control

Bo Yang ^a, Weixing Yuan ^{*,a}, Yonghong Shang ^b, Jing Wang ^{b,c}, Bin Wei ^a

^a Laboratory of Ergonomics and Environmental Control, School of Aeronautic Science and Engineering, Beihang University, Beijing 100191, China

^b Beijing Institute of Spacecraft Environment Engineering, Beijing 100094, China

^c Science and Technology on Reliability and Environmental Engineering Laboratory, Beijing 100094, China

*Corresponding Author: Tel/fax:+86-10-82338878; E-mail address: yuanwx@buaa.edu.cn

Abstract

Condensing dehumidification method is widely applied to humidity control in manned spacecraft cabin. Under the micro-gravity condition, the condensate water has to be removed actively by additional water/air separation and water recycling processes. This paper presents a novel three-fluid membrane dehumidification method for spacecraft humidity control, which is expected to replace the condensing method. The cold water provides a low water vapor partial pressure to drive the moisture transfer across the membrane from the humid air, while the hot water maintains the humid air temperature above the dew point to prevent the possible dewing. The diffusivity of water vapor in membrane is first determined by an experiment with the help of the particle swarm optimization (PSO) algorithm. With the experimentally determined diffusivity, the three-fluid membrane dehumidification is studied experimentally and theoretically, and the mathematical model reaches a good agreement with the experimental results. The novel dehumidification method is experimentally validated to be feasible; that is to say the dehumidification can be realized and meanwhile there is no dewing on the outer surface of the membrane. The heat and mass transfer performance is also studied under various operating conditions. The three-fluid membrane dehumidification shows superior thermodynamic performance compared with the condensing method.

Keywords: spacecraft; three-fluid; membrane dehumidification; particle swarm optimization; heat and mass transfer

0. Introduction

In the enclosed manned spacecraft cabin, the humidity will increase continuously caused by the metabolic moisture generation of crew members, if no dehumidification devices are installed. The high humidity restrains the perspiration of human bodies, which affects both thermal comfort and ergonomics of astronauts [1]. Meanwhile, bacteria growth will be prosperous at high humidity [2]. In addition, high humidity also harms onboard electronics.

Thus, the air dehumidification is a very important task in spacecrafts. The current dehumidification devices in spacecrafts are mainly based on condensing method, such as in Space Shuttle [3] and

International Space Station (ISS) [4]. In this dehumidification mode, a condensing heat exchanger (CHX) is used to cool the air below the dew point, and therefore the water vapor is condensed into liquid water. A critical problem is that the water/air separation is very tough to accomplish under micro-gravity. One solution is to apply a rotary water/air separator, which consumes much power and increases the system weight and complexity. Hygroscopic material is another solution, by which the water is absorbed and stored. However, the water content in the hygroscopic material will finally reach saturation, limiting the rate of the further absorption, and the condensed water cannot be reused. Furthermore, it should be noted that, after being treated in the CHX, the product air is too cold for crew members to supply directly, so an additional heating step is necessary. As seen, the whole process consumes too much energy. In many spacecrafts on active service such as ISS, the water/air separator is usually used in cooperation with the hygroscopic coating applied on the fins of the CHX. After long-duration use, the hygroscopic coating may dissolve or peel off and will be carried away by the airflow into the separator, causing the separator to get clogged [5]. Malfunctions of the water/air separator have been reported several times by the ISS crew [6, 7]. Since the separator is hard to maintain due to its complex structure, the malfunctioning separator has to be replaced waiting for the next launch and delivery task.

Recently, membrane dehumidification attracts more and more attentions. It is achieved on the basis of the moisture transfer across the membrane driven by the water vapor partial pressure difference between the feed and permeate sides. Various membrane modules have been fabricated and investigated. Zaw et al. [8] studied a cross-flow plate membrane dehumidification unit for air conditioning in tropical climates, in which the moisture was taken away by the dry exhausting air across membranes. Zhang et al. [9] affixed sinusoidal membrane fins between parallel plate membranes in order to enhance the mechanical strength. Li et al. [10] developed a membrane exchanger with cross-corrugated triangular ducts, which achieved good heat and mass transfer performance at low Reynolds numbers. Ge et al. [11, 12] and Abdel-Salam et al. [13] developed a counter-cross-flow liquid-to-air membrane energy exchanger (LAMEE) and conducted comprehensive experimental and theoretical investigations to find optimal geometries. In addition to the plate-structured membrane modules mentioned above, hollow fiber membrane modules are also common, including counter-flow [14] and cross-flow [15] configurations. Huang and Yang [16] found that the heat and mass transfer performance was enhanced by using elliptical hollow fibers in a membrane contactor. More detailed research progress on membrane modules used for air dehumidification was reviewed in references [17-21].

In common applications, liquid desiccant [11-16, 22] or dry air [8-10] is usually employed as the permeate-side fluid. However, they are not applicable in the enclosed environment of spacecrafts in consideration of the regeneration of the permeate-side fluid. Especially, the corrosive liquid desiccant may do harm to pipelines.

Considering that cold water can also provide low water vapor partial pressure, it is an alternative for the permeate-side fluid. Compared with the liquid desiccant and dry air, the main advantage of the cold water is that there is no need for the concentration recovery of the permeate-side fluid. When the water vapor is absorbed by the pure cold water, the water temperature will rise due to the total heat transfer from the humid air. So only the thermal recovery is required by cooling the temperature-raised cold water before cycling back. In the meanwhile, the moisture in the air is totally recycled for reuse.

However, there is contradiction between the heat and mass transfer in the membrane dehumidification using cold water. On one hand, the water temperature should be low enough, or there

will be not enough dehumidification capacity. For example, the equilibrium humidity ratio of water at 23.9 °C is 18.8 g/kg, equal to the humidity ratio of the air at 30 °C and 70% RH. If the water temperature is above 23.9 °C, the air at 30 °C and 70% RH will not be dehumidified. On the other hand, if the water temperature is too low, the dewing problem will occur, which has been observed in our previous work [23]. The unmatched transfer ability between heat and mass transfer is the key reason. Qualitatively speaking, once the mass transfer ability is lower than the heat transfer ability, the dewing will be unavoidable. It will make no difference from the condensing method if the dewing occurs, meaning the membrane dehumidification fails.

1. Concept of three-fluid membrane dehumidification

To overcome the problem presented above, this paper proposes a novel three-fluid membrane dehumidification method. The membranes tested in our work are asymmetric, whose inner surface is hydrophobic while outer surface is hydrophilic. So the humid air flows in the shell side, while the cold water flows in the tube side. In order to prevent the humid air temperature from falling below the dew point, a casing tube is arranged concentrically between the hollow fiber membrane and the shell, creating two adjacent annular passages for the hot water and the humid air, respectively. The humid air flows in the inner annular passage between the outer wall of the membrane and the inner wall of the casing tube. The schematic of the dehumidification unit is depicted in Fig. 1. By the supplementary heat from the hot water, the air temperature can be maintained above the dew point, and therefore the possible dewing on the outer surface of the membrane is avoided.

Fig. 1 Schematic of the three-fluid membrane dehumidification unit

In fact, there has been a similar concept proposed by Abdel-Salam et al., who recently published a series of academic articles on three-fluid membrane dehumidification [24-28]. In the membrane dehumidification using liquid desiccant, the desiccant temperature will be raised caused by the conjugated sensible and latent heat transfer from the humid air, which leads to the increase of the water vapor partial pressure of the desiccant and therefore results in the deterioration of the dehumidification performance. To overcome this problem, Abdel-Salam et al. designed an artful three-fluid LAMEE on the basis of their previously developed two-fluid LAMEE. Inside the desiccant channels were installed a row of parallel titanium tubes. A stream of cold water flew in these tubes to decrease the desiccant temperature. The innovative three-fluid LAMEE achieved remarkable promotion of both the moisture removal rate and cooling capacity compared with the previous two-fluid LAMEE.

In spite of the same three-fluid membrane dehumidification mode, it is different essentially between this paper and the research of Abdel-Salam et al. Firstly, the membrane materials and module structures are different. This paper investigates a kind of concentric-tube configuration based on hollow fiber

membranes, compared with the parallel plate configuration based on flat-plate membranes by Abdel-Salam et al. Secondly, the working fluids are different. The cold water is employed as the permeate-side fluid to absorb the moisture across the membranes in this paper, while Abdel-Salam et al. select the liquid desiccant. Most of all, the purports of the three-fluid mode are different. The third working fluid here in this paper is a hot water stream used for preventing the dewing. In comparison, the supplementary cold water is employed in the three-fluid LAMEE in order to decrease the desiccant temperature and therefore enhance the dehumidification performance.

2. Route map of the present study

As a key parameter affecting the trans-membrane mass transfer significantly, the diffusivity of water vapor in membrane is determined first. A two-fluid membrane module without the supplementary hot water stream was tested so as to eliminate the effects of the third fluid. The diffusivity was determined accurately from the experimental data in cooperation with the particle swarm optimization (PSO) algorithm. With the experimentally determined diffusivity, the three-fluid membrane dehumidification is modeled then. A three-fluid membrane module is designed and fabricated on the basis of the two-fluid module. A three-fluid membrane dehumidification experiment is conducted to validate the feasibility of the three-fluid concept. With the theoretical model having been proven to be reliable, simulation work is conducted to study the effects of the operating conditions and membrane length on the dehumidification performance of a three-fluid membrane dehumidification unit. A virtual full-scale module comprised of many three-fluid units is then studied numerically aiming at the spacecraft application. The simulation results can provide a guide for the design and fabrication of the full-scale prototype in our future work.

3. Determination of the diffusivity

3.1. Commonly applied methods

The transport mechanism of the moisture in the membrane is related to the membrane properties and microscopic structures. For dense membranes, the moisture transport is commonly explained by solution-diffusion theory [29-32]. According to this theory, the water vapor first dissolves in the membrane and then diffuses through the membrane under concentration (i.e. water content in membrane) gradient. For micro-porous membranes, the pore-flow model is mainly employed [33-35]. It states that the water vapor diffuses in gas phase through the membrane pores. The diffusion mechanisms differ with respect to pore sizes. If the pores are large enough in comparison with the free path of the water vapor molecules, the Poiseuille flow will predominate. On the contrary, the Knudsen diffusion will predominate. Sometimes the molecular diffusion is also considered [33].

Whatever diffusion mechanism predominates in the micro-porous membranes, the structure parameters including pore diameter d_p , porosity ε and tortuosity τ must be known, as presented in Eqs. (1) and (2) [34]

$$D_K = \frac{d_p \varepsilon}{3\tau} \sqrt{\frac{8RT}{\pi M_v}} \quad (1)$$

$$D_p = \frac{d_p^2 \varepsilon p}{32 \tau \mu} \quad (2)$$

where D_K and D_p are the Knudsen and Poiseuille diffusivities, respectively, which are both based on the humidity ratio difference; δ is the membrane thickness; M_v is the molecular mass of water vapor; T and p are the mean temperature and pressure in pores, respectively.

Conventionally, the mean values of the pore diameter and tortuosity are used for the diffusivity calculation. For many membranes, however, the pores are not ideally cylindrical and the pore sizes are not uniform, such as the membrane used in this paper as shown in Fig. 2. The average treatment on the membrane structures may not be accurate enough for the calculation of the diffusivity.

Fig. 2 SEM image of the tested hollow fiber membrane

Zhang has developed a fractal model to calculate the diffusivity more accurately [36]. The distribution of the pore size and amount are both assumed to follow the fractal scaling law (a kind of power relationship) by introducing some fractal dimensions. Through the analysis on the SEM (scanning electron micrograph) pictures of membrane cross section and surface by the box-counting method, the fractal dimensions are obtained, and therefore the fractal model is solved to determine the diffusivity. Recently, Zhang et al. have employed the Lattice-Boltzmann simulation approach to solve the detailed flow and mass transfer in membranes [37]. By these new approaches, some valuable investigation results have been obtained. However, the fractal model is strongly dependent on the SEM image quality and the statistical analysis process, while the Lattice-Boltzmann simulation approach is

quite complex and requires extremely high computation ability.

3.2. Experimental method in this paper

In this paper, we decide to determine the diffusivity via a heat and mass transfer experiment. A simple two-fluid membrane module is fabricated in the shell-and-tube configuration, as shown in Fig. 3. Six hollow fiber membranes are packed inside the SUS304 (a kind of stainless steel) shell. The air flows in the shell side counter to the water in the tube side. The hollow fiber membrane is constructed of a hydrophilic PVA layer and a hydrophobic PVDF layer. The outer PVA layer provides air/water vapor selectivity, while the inner PVDF layer provides mechanical support and prevents the leakage of the water in the tube side. The thermo-physical properties of the hollow fibers and structural parameters of the two-fluid membrane module are listed in Table 1.

Fig. 3 The two-fluid membrane module prepared for the diffusivity determination experiment.

Table 1 Parameters of the hollow fibers and the two-fluid membrane module

Parameter	Value	Unit
Pore size	0.15	μm
Porosity	0.65	
Tortuosity	3	
Conductivity	0.17	W/m K
Membrane amount	6	-
Membrane effective length	423	mm
Membrane outer diameter	1.44	mm
Membrane inner diameter	0.91	mm
Shell inner diameter	7	mm
Shell outer diameter	9.48	mm

The two-fluid membrane module was tested in a micro-flow experimental set-up, as shown in Fig. 4.

1

2

3

Fig. 4 The diffusivity determination experimental set-up

4

Two air streams are pumped into the test rig, one of which is processed in a wet curtain humidification apparatus. By regulating the water temperature with the PID controller (UT55, Yokogawa, Japan), the humidification capacity will be adjusted accurately. The humidified air stream is then mixed with the dry air stream, reaching the required humidity and flow rate. Then the air flows through an electrical heater, which can accurately control the air temperature via another PID controller. With the temperature and humidity reaching the set points, the air is blown into the membrane module.

9

The water loop is driven by a peristaltic pump. A thermoelectric module is mounted to heat or cool the water according to requirements. By regulating the input power of the thermoelectric module via a PID controller, the water temperature will be controlled accurately.

13

The parameters of all the measuring instruments are listed in Table 5. The humidity ratio is calculated from the measured temperature and relative humidity. Under the experimental conditions, the uncertainty of the humidity ratio is ± 0.55 g/kg approximately.

16

Table 2 Parameters of the measuring instruments

Item	Type	Model	Scale	Uncertainty
Air temperature	PT100	STT-C2, Sailing, China	-50~100 °C	± 0.1 °C

Air relative humidity	Humidity transducer	STH-TD, Sailing, China	0~100% RH	±2% RH
Air flow rate	Glass rotameter	LZB-6WB, Shuanghuan, China	1.5~15 l/min	±2.5%
Water temperature	PT100	STT-C2, Sailing, China	-50~100 °C	±0.1 °C
Water flow rate	Glass rotameter	LZB-3WB, Shuanghuan, China	2.5~25 ml/min	±4%

In order to obtain an accurate enough diffusivity value, ten groups of experiment under different inlet conditions are conducted. Under the same inlet conditions as the experiment, the heat and mass transfer process is calculated with an assumed diffusivity. In the calculation, the heat losses including the module shell (R_s), air-side joints (R_{aj}) and water-side joints (R_{wj}) are all considered. Then all the calculated results (V_{Exp}) and experimental results (V_{Cal}) are compared, and the residual squares are summed.

$$\sigma = \sum_{i=1}^{10} \left(\frac{V_{Exp} - V_{Cal}}{V_{Exp}} \right)^2 \quad (3)$$

If the residual sum of squares is too big relative to the expected, a new diffusivity will be selected for an iteration calculation. As seen, it is a multi-objective optimization problem essentially. Besides the diffusivity, the thermal resistances including R_s , R_{aj} and R_{wj} are also the optimization objectives. The comprehensive effects of the four parameters on the heat and mass transfer performance are not simply monotonous, so there may be lots of local optimal solutions.

In response to this problem, this paper applies the PSO algorithm to deal with the experimental results to search the global optimal solution. PSO is a recently-developed heuristic search technique by simulating the social behavior of an insect swarm or a bird flock [38]. Particles are employed to simulate the insects and birds. Each particle compares its current value with its historical optimal value, and meanwhile it exchanges information with other particles and makes comparison, then it will update the location following the iteration formulas as below

$$v_{k+1} = wv_k + c_1\zeta(p_h - x_k) + c_2\eta(p_s - x_k) \quad (4)$$

$$x_{k+1} = x_k + rv_{k+1} \quad (5)$$

where the subscript k denotes the current step, and $k+1$ denotes the next step; v is the velocity; x is the location, denoting the four optimization objectives; p_h is the historical optimal location of a specific particle; p_s is the global optimal location of the society which is formed by the “adjacent” population comprised of several particles or all particles; ζ and η are historical and social random numbers, respectively, following 0-1 uniform distribution; c_1 and c_2 are historical and social factors, respectively; w is the weight factor; r is the restriction factor designed for cushioning the movement, which favors the convergence especially when the particle is close to the global optimal location.

On right side of Eq. (4), the second term represents the self-cognition ability of a single particle, while the third term represents the social learning ability of the particle, reflecting the information exchange level of the society. Through the self-cognition and social learning of all particles, the exploration ability for the optimal location of the swarm reaches a high level. After updating locations many times, all particles will converge at the optimal location finally.

Since the PSO algorithm is originally designed for maximum optimization, the objective function in this paper is

$$f = -\min(\sigma) \quad (6)$$

Considering the exploring ability of the swarm and the computation load altogether, the particle number is set as 20. The society of each particle is formed by the adjacent ten particles in clockwise direction. The historical and social factors are both 2. The restriction factor is 0.729. The weight factor

is decreased linearly from 1 to 0.1 as iteration goes on. The maximal iteration number is 200. The iterating calculation reaches convergence finally, with a minimal residual sum of squares of 0.00657 obtained. Correspondingly, D_{vm} , R_s , R_{aj} , R_{wj} are $6.44 \times 10^{-6} \text{ m}^2/\text{s}$, 734.0 K/W, 5.681 K/W, 232.7 K/W, respectively. It is observed from the results that the heat loss of the module shell and water-side joints can be neglected, while the heat loss of the air-side joints is considerable. The experimental and calculated results are listed in Table 3, from which we can see that the temperature deviations are below 0.7 °C and the humidity ratio deviations are below 0.5 g/kg. Considering the heat loss at the joints and shell, the deviation between the total enthalpy changes in the shell side and the tube side is less than 7.40%.

Table 3 Comparison between the experimental and calculated results

Q_{va}	t_{ai}	RH_{ai}	Q_{vw}	t_{wi}	t_e	t_{ao}		ω_{ao}		t_{wo}	
l/min	°C	%	ml/min	°C	°C	°C		g/kg		°C	
						Exp	Cal	Exp	Cal	Exp	Cal
10.3	30	34.7	15.4	30.0	26.5	26.9	27.47	16.58	17.007	26.0	26.56
10.3	30	40.0	14.5	30.0	26.8	27.0	27.66	17.48	17.727	26.1	26.69
10.3	30	45.2	13.0	30.0	26.9	27.1	27.72	18.41	18.394	26.4	26.72
10.3	30	50.2	13.0	30.0	27.0	27.2	27.81	19.25	19.154	26.7	27.02
10.3	30	55.0	13.0	30.0	27.2	27.4	27.97	20.16	19.887	27.0	27.32
10.6	30	34.8	13.7	35.0	26.8	29.2	28.84	20.06	19.953	29.1	28.83
10.6	30	40.0	14.1	35.0	27.1	29.5	29.07	20.85	20.819	29.4	29.25
10.6	30	45.2	13.7	35.0	27.3	29.7	29.21	21.81	21.570	29.7	29.42
10.6	30	50.1	13.3	35.0	27.4	29.8	29.29	22.50	22.281	29.8	29.59
10.3	30	55.0	14.3	35.0	27.3	29.1	29.26	23.12	23.377	30.1	30.18

4. Three-fluid membrane dehumidification experiment

A three-fluid membrane module is designed and fabricated, as shown in Fig. 5. A single fiber membrane is packed in a SUS304 casing tube which is arranged concentrically in a SUS304 shell, successively creating a cold water passage, an air passage and a hot water passage inside and out. Restricted by the fabrication processes, the effective heating length of the casing tube is shorter than the effective membrane length. The structural parameters of the three-fluid membrane module are listed in Table 4.

Fig. 5 Three-fluid membrane module

Table 4 Structural parameters of the three-fluid membrane module

Parameter	Value	Unit
Effective heating length	380	mm
Effective membrane length	600	mm
Membrane inner diameter	0.91	mm
Membrane outer diameter	1.44	mm
Casing tube inner diameter	4	mm
Casing tube outer diameter	6	mm
Shell inner diameter	10	mm

The three-fluid membrane dehumidification experimental set-up is established similar to the diffusivity determination experimental set-up, with a hot water loop added, as shown in Fig. 6. The measurement and control approaches are the same as those in the diffusivity determination experiment. Under the experimental conditions as listed in Table 5, the uncertainty of the dehumidification rate is $\pm 5.0\text{--}7.1 \text{ g/h m}^2$.

Fig. 6 Three-fluid experimental set-up

Table 5 Operating conditions of three-fluid experiment

t_e (°C)	t_{ai} (°C)	RH_{ai} (%)	t_{ci} (°C)	t_{hi} (°C)
21.8	26.0	64.83	14.0	35.1
21.6	27.0	64.78	14.0	35.2
21.3	28.0	65.33	14.0	35.0
21.4	29.0	65.00	14.0	35.0
21.3	30.0	65.03	14.0	35.0
22.7	28.0	65.13	14.9	39.9
23.1	29.0	65.15	15.1	40.0
23.2	30.0	65.19	15.2	40.0
22.6	26.0	69.94	14.7	34.9
22.4	27.0	70.06	14.9	35.0
22.3	28.0	70.14	15.0	35.0

22.7	29.0	70.24	15.0	35.2
------	------	-------	------	------

5. Three-fluid membrane dehumidification model

5.1. Conservation equations

In the three-fluid membrane dehumidification unit as shown in Fig. 7, the humid air releases both sensible heat and moisture towards the cold water across the membrane wall, and simultaneously absorbs sensible heat from the hot water across the casing tube wall.

Fig. 7 Heat and mass transfer in the three-fluid membrane dehumidification unit

In the three-fluid unit, the radial dimension is far smaller than the axial dimension. The fluids are assumed to occupy their respective passages homogeneously in radial direction, so it is reasonable to assume the three-fluid membrane dehumidification process as a one-dimensional problem [39, 40]. Due to the relatively high heat and mass transfer Peclet numbers which are generally larger than 10 in this paper, the axial thermal conduction and moisture diffusion in fluids are neglected [39, 40]. Based on the above assumptions, the energy and composition conservation equations in the control volume are established in finite difference scheme as follows.

For the humid air, the changes of temperature and humidity ratio are presented in Eqs. (7) and (8)

$$dQ_1 - dQ_2 = -\dot{m}_a c_{pa} dt_a \quad (7)$$

$$dJ = -\dot{m}_a d\omega_a \quad (8)$$

where the subscript a denotes the humid air; dQ_1 and dQ_2 are the heat transfer rates towards the cold water and from the hot water, respectively; dJ is the moisture transfer rate from the humid air to the cold water across the membrane.

The absorption heat is released when the moisture is absorbed by the cold water. In addition to the sensible heat transferred from the humid air, the cold water temperature will increase, as presented in Eq. (9)

$$dQ_1 + r dJ = \dot{m}_c c_{pc} dt_c \quad (9)$$

where the subscript c denotes the cold water; r is the absorption heat, 2501 kJ/kg K. The absorption heat leads to the further increase of the permeate-side temperature in addition to the sensible heat [28].

The hot water is cooled by releasing the sensible heat towards the humid air through the casing tube wall, as presented in Eq. (10)

$$dQ_2 = \dot{m}_h c_{ph} dt_h \quad (10)$$

where the subscript h denotes the hot water.

The heat and mass transfer rates in each control volume can be calculated as follows.

$$dQ_1 = h_1 \Delta t_{m1} dA_1 \quad (11)$$

$$dQ_2 = h_2 \Delta t_{m2} dA_2 \quad (12)$$

$$dJ = \rho_a k \Delta \omega_m dA_1 \quad (13)$$

where Δt_m is the logarithm mean temperature difference; $\Delta \omega_m$ is the logarithm mean humidity ratio difference. The equilibrium humidity ratio of the water is dependent on the water temperature, which can be calculated by Eq. (14)

$$\omega(t) = 10^6 \exp\left(-\frac{5294}{t + 273.15}\right) \quad (14)$$

In Eqs. (11-13), the overall heat and mass transfer coefficients h_1 , h_2 and k in each control volume are calculated based on the inner surface areas of the hollow fiber membrane and the casing tube, as

$$\frac{1}{h_1} = \frac{1}{h_{ai}} \frac{d_{mi}}{d_{mo}} + \frac{1}{h_c} + \frac{d_{mi}}{2\lambda_m} \ln\left(\frac{d_{mo}}{d_{mi}}\right) \quad (15)$$

$$\frac{1}{h_2} = \frac{1}{h_{ao}} + \frac{1}{h_h} \frac{d_{ti}}{d_{to}} + \frac{d_{ti}}{2\lambda_m} \ln\left(\frac{d_{to}}{d_{ti}}\right) \quad (16)$$

$$\frac{1}{k} = \frac{1}{k_a} \frac{d_{mi}}{d_{mo}} + \frac{1}{k_c} + \frac{d_{mi}}{2D_{vm}} \ln\left(\frac{d_{mo}}{d_{mi}}\right) \quad (17)$$

where d_{mi} and d_{mo} are the inner and outer diameters of the membrane, respectively; d_{ti} and d_{to} are the inner and outer diameters of the casing tube, respectively; h_{ai} and h_{ao} are the convective heat transfer coefficients of the inner and outer surface of the air-side annular passage, respectively; D_{vm} is the diffusivity of the water vapor in membrane. The convective mass transfer resistance of the cold water side, $1/k_c$, can be neglected [40, 41]. The calculation of the convective heat and mass transfer coefficients and membrane diffusivity in Eqs. (15-17) will be described later.

Grid independence has been analyzed previously. The numerical results of 50 grids have less than 1% differences compared with 100 grids, indicating that 50 grids are adequate in this paper. By setting the outlet parameter values of the upstream control volume as the inlet conditions of the downstream control volume, all the conservation equations in 50 control volumes are combined. Then the equation set is solved by Newton iteration algorithm.

5.2. Convective heat and mass transfer

For the tube flow of the cold water, the Hausen equation is applicable when the Graetz number ($Gz = RePr_d/L$) is below 100 [42]

$$Nu = 3.66 + \frac{0.0668 RePr(d_h/L)}{1 + 0.04[RePr(d_h/L)]^{0.67}} \quad (18)$$

If the Graetz number exceeds 100, the Sieder-Tate equation is applied [43]

$$Nu = 1.86 \left(RePr \frac{d_h}{L} \right)^{0.33} \quad (19)$$

If the flow is in the turbulent pattern, the Dittus-Boelter equation is applied [43]

$$Nu = 0.023 Re^{0.8} Pr^{0.4} \quad (20)$$

For the flow of the air and hot water in the annular passages, the heat transfer characteristics on the inner and outer surfaces of the annular passages are different depending on the passage geometries and heat flux. The inner and outer Nusselt numbers are calculated as follows [44, 45]

$$Nu_i = \frac{Nu_{ii}}{1 - (q_o/q_i)\theta_i} \quad (21)$$

$$Nu_o = \frac{Nu_{oo}}{1 - (q_i/q_o)\theta_o} \quad (22)$$

where q_i and q_o are the heat flux of the inner and outer surfaces of the annular passages, respectively. Nu_{ii} , Nu_{oo} , θ_i and θ_o , which are determined by the inner/outer radius ratio, r^* , are given in Table 5 [44].

Table 5 Parameters for the Nu calculation of the annular passages

Annular passage	r^*	Nu_{ii}	Nu_{oo}	θ_i	θ_o
Air	0.36	6.808	4.961	0.637	0.165
Hot water	0.55	6.047	5.064	0.490	0.232

Since the mass transfer rate is relatively small, the mass transfer can be analogous to the heat transfer according to the Chilton-Corburn equation [46]

$$Sh = NuLe^{-1/3} \quad (23)$$

where $Le = Pr/Sc$ is the Lewis number.

6. Results and discussions

6.1. Model validation

According to the experiences in the cold water membrane dehumidification experiment, when the dewing occurs, the air flow rate will fluctuate because the condensate water accumulates in the air passage. In the three-fluid membrane dehumidification experiment, the air flow rate keeps stable all through, and the outlet relative humidity is indeed below 100% RH. It is validated that the three-fluid membrane dehumidification can actually eliminate the dewing.

The three-fluid membrane dehumidification experiment is conducted under various inlet conditions. The experimental and calculated dehumidification rates are compared, as shown in Fig. 8. As seen, the model reaches a good agreement with the experiment. Energy balance is also checked that the deviation of enthalpy changes is less than 6.31% in consideration of the heat loss.

Fig. 8 Comparison between the experimental and calculated dehumidification rates

6.2. Basic performance analysis by simulation

With the three-fluid membrane dehumidification model having been validated, a simulation work is conducted to study the performance of the three-fluid membrane dehumidification comprehensively. In the simulation work, we design an ideal three-fluid membrane unit with the effective heating length equal to the effective membrane length. The membrane area is 0.00191 m^2 . The computation domain is discretized as 50 nodes along the longitudinal direction. The parameters of each node are calculated.

The inlet conditions are 1 l/min, 28 °C and 75% RH for the air, 5 ml/min and 10 °C for the cold water, 5 ml/min and 32 °C for the hot water. The distribution of the temperature and humidity ratio is plotted in Fig. 9, where the air flows to right, while both the cold and hot water flow to left. As observed from the curves, the cold water temperature keeps rising along the flow direction, while the hot water temperature keeps falling. It is different for the air that its temperature variation is not simply monotonic. At the beginning section before Node 7, the air temperature falls remarkably because of the small temperature difference between the air and the hot water but large temperature difference between the air and the cold water. Then, the heat transfer reaches balance almost, so the air temperature nearly stays constant, with a tiny increase. The air humidity ratio decreases continuously along the flow direction from 17.90 g/kg to 10.29 g/kg, achieving a dehumidification rate of 279 g/h m^2 . At the beginning, with the air temperature falling greatly, the relative humidity appears to increase sharply though the humidity ratio decreases, up to 85.9% RH at the peak. Then, the air temperature stays unchanged almost, so the relative humidity decreases with the humidity ratio, falling to 52.6% RH at the outlet. As seen, there is no dewing all through.

Fig. 9 Distribution of (a) temperature of three streams and (b) humidity of the air, where the inlet conditions are 1 l/min, 28 °C, 75% RH for the air, 5 ml/min, 10 °C for the cold water and 5 ml/min, 32 °C for the hot water

The hot water maintains the air temperature to prevent it from falling too much so that the dewing is avoided. There should be a lower limit of the hot water temperature when the air temperature cannot be maintained, so the effect of the hot water temperature is necessary to investigate. As shown in Fig. 10, the hot water temperature affects the air outlet temperature greatly and linearly, with the outlet temperature difference between the air and hot water remained at 3-5 °C. On the contrary, the cold water outlet temperature increases little with respect to the hot water temperature, so the dehumidification performance hardly changes. The dehumidification rate varies in a narrow range of 276 to 284 g/h m², which is not plotted here. Although the influence of the hot water temperature on the dehumidification rate is rather slight, the hot water temperature greatly affects the peak value of the local air relative humidity as shown in Fig. 11. If the hot water temperature is decreased down to 26.8°C, the humid air temperature will reach the dew point at a certain location close to the entrance. It implies that increasing the hot water temperature will help to reduce the dewing risk. However, the hot water temperature should not be too high either, or else the thermal loads of both the hot and cold water will be increased. The effects of the hot water flow rate on the dehumidification rate and on the outlet parameters are not remarkable, so the curves have not been plotted here.

Fig. 10 Effect of the hot water temperature on the outlet parameters, where the inlet conditions are 1 l/min, 28 °C, 75% RH for the air, 5 ml/min, 10 °C for the cold water and 5 ml/min for the hot water

Fig. 11 Distributions of the air relative humidity at different hot water inlet temperature, where the other inlet conditions are 1 l/min, 28 °C, 75% RH for the air, 5 ml/min, 10 °C for the cold water and 5 ml/min for the hot water

With the tough dewing problem having been successfully resolved by employing a hot water stream to maintain the air temperature above dew point, it is possible to operate the cold water at a relatively low temperature to advance dehumidification performance, as the cold water provides a low water vapor partial pressure and therefore a high water vapor partial pressure difference for moisture transfer across the membrane. The lower the cold water temperature is, the larger the moisture transfer potential is and the higher the driving force is, which results in the higher dehumidification rate, as shown in Fig. 12. At a cold water temperature of 5 °C, the dehumidification rate reaches as high as 348 g/h m². When the cold water temperature is up to 20 °C, the dehumidification rate is only 87 g/h m². Low cold water temperature benefits the dehumidification naturally, but the cooling load of the cold water will be increased as a result of the more sensible and latent heat transfer amount. As the cold water temperature rises, the dehumidification rate declines remarkably, and therefore the outlet humidity ratio of the air increases, and the outlet relative humidity increases correspondingly, as shown in Fig. 12.

Fig. 12 Effect of the cold water temperature on the dehumidification rate, where the inlet conditions are 1 l/min, 28 °C, 75% RH for the air, 5 ml/min for the cold water and 5 ml/min, 30 °C for the hot water

The effect of the cold water flow rate on the dehumidification performance is also studied, as shown in Fig. 13. The cold water flow rate affects the dehumidification rate indirectly via the cold water temperature. At low flow rate, the cold water temperature increases rapidly along the flow direction, diminishing the mass transfer potential, so the dehumidification rate is low. At high flow rate, the cold water can be maintained at low temperature more easily, achieving a significant dehumidification rate therefore. A further increase in the cold water flow rate raises the dehumidification rate slightly.

Fig. 13 Effect of the cold water flow rate at various cold water temperature, where the inlet conditions are 1 l/min, 28 °C, 75% RH for the air and 5 ml/min, 30 °C for the hot water

The air flow rate exerts considerable impact on the dehumidification performance. On one hand, the air-side convective mass transfer coefficient increases along with the air flow rate as a result of the increased Reynolds number. On the other hand, the air humidity ratio decreases slowly at high flow rate, obtaining a higher mass transfer potential. As a consequence, the dehumidification rate increases along with the air flow rate, as shown in Fig. 14. However, the humidity ratio drop decreases due to the increase of the air flow rate though the dehumidification rate is promoted.

The effect of the air inlet relative humidity is also studied, as shown in Fig. 14. It is apparent that, the higher the inlet relative humidity is, the greater dehumidification rate is. That is because the water vapor partial pressure difference is larger at higher inlet relative humidity. For the same reason, the humidity ratio drop increases along with the inlet relative humidity.

Fig. 14 Effect of the air flow rate at various relative humidity, where the inlet conditions are 30 °C for the air, 5 ml/min, 10 °C for the cold water and 5 ml/min, 30 °C for the hot water

The effectiveness of the three-fluid membrane module is also calculated and analyzed. Since the hot water limits the further decrease of the air temperature, the sensible effectiveness is not high indeed, generally below 30%. The membrane module achieves a relatively good latent effectiveness, however. Compared with other operating conditions, the air flow rate exerts remarkable effects on the latent effectiveness. Though the dehumidification rate increases along with the air flow rate, the humidity ratio drop will decrease. So the latent effectiveness decreases as the air flow rate increases, as shown in Fig. 15.

Fig. 15 Effect of the air flow rate on the latent effectiveness, where the inlet conditions are 30 °C, 75% RH for the air, 5 ml/min, 10 °C for the cold water and 5 ml/min, 30 °C for the hot water

The effect of the membrane length on the dehumidification rate is also studied. It is obvious that, the longer the membrane is, the greater the dehumidification rate of a single membrane is. However, the dehumidification per unit membrane area will decrease, as shown in Fig. 16. It is because that the membrane area increases linearly along with the membrane length. Suppose that the membrane is infinitely long, the dehumidification capacity of the single membrane will approach a maximum value and tend to be steady as the heat and mass transfer reaches sufficient, so the dehumidification rate per unit membrane area will tend to zero. It is concluded theoretically that, for fabricating a module with a given membrane area, it is better to select more short membranes than fewer long membranes. In practice, however, it should be comprehensively considered including the dimensions and the flow

1 uniformity.

Fig. 16 Effect of the membrane length on dehumidification rate

6.3. Simulation study aiming at spacecraft application

In the manned spacecraft cabin, the humidity load differs widely depending on the activity levels of the astronaut. At nominal awake state, one astronaut releases less than 70 g/h moisture [47]. At exercise state, the moisture generation rate increases sharply. For instance, the moisture generation rate can reach as high as nearly 1 kg/h after 30 minutes of exercise [47]. Though, the exercise duration accounts for quite small ratio in the whole day. Especially considering the long duration for sleeping when the moisture generation rate is rather low, the overall moisture generation amount of one astronaut during the whole day is 1.5 kg approximately, i.e. 63 g per hour [48].

According to the dehumidification demand of spacecrafts, a three-fluid membrane dehumidification module comprised of 523 dehumidification units is designed with the membrane area of 1 m². By packing the dehumidification units in square inline arrangement with a space of 12 mm, the membrane module core has dimensions of 276 mm × 276 mm × 423 mm. With the water inlet conditions fixed at 2.62 l/min, 10 °C for the cold water and 2.62 l/min, 30 °C for the hot water, the dehumidification performance at various air inlet conditions is evaluated to study if the designed membrane module is competent for the inner environment of spacecraft cabin.

The dehumidification capacity (g/h) of the membrane module at various air inlet temperature and relative humidity is plotted in Fig. 17, where the air flow rate is fixed at 1.05 m³/min. Under the operating conditions in the calculation, both Cr_{ca} and Cr_{ha} are 8.87, while NTU and NTU_m are 0.79 and 0.51, respectively. As seen, the dehumidification capacity increases along with both the air temperature and relative humidity as a result of the increased humidity ratio which raises the mass transfer potential. The dashed and dotted lines represent the dehumidification capacity demands of two-person crew and three-person crew, respectively, i.e. 126 g/h and 189 g/h. As demonstrated by the figure, in the range of 24 to 28 °C and 65 to 85% RH of the air inlet conditions, the membrane module can perfectly satisfy the dehumidification demand of two-person crew. When the air temperature is above 26°C, the membrane module is also capable for three-person crew.

Fig. 17 Dehumidification capacity at various air inlet temperature and relative humidity

During the dehumidification process, the cold water absorbs both the sensible and latent heat from the humid air, leading to the increase of the temperature and enthalpy of the cold water. At the same time, the hot water releases the sensible heat to the air to prevent the air temperature from falling below the dew point, so the temperature and enthalpy of the hot water are reduced. As discussed above, higher inlet relative humidity of the humid air brings about larger latent heat transfer amount, while higher air temperature gives rise to larger sensible and latent heat transfer amount altogether. Hence, the enthalpy change of the cold water increases along with the air inlet temperature and relative humidity, as shown in Fig. 18.

It has been known that the enthalpy change as well as the temperature change of the cold water is affected by the relative humidity of the humid air because of the latent heat. However, the mutual interact between the cold and the hot water temperature is weak, as revealed by Fig. 10. The air shows an excellent thermal insulation effect between the cold and hot water. So the relative humidity of the humid air affects the enthalpy change of the hot water slightly.

The enthalpy changes of the cold and hot water reflect the energy expense in maintaining the steady run of the dehumidification process. As a result of the enthalpy increase, the cold water needs to be cooled before cycling back, which consumes cooling energy. In contrast, the hot water needs to consume heating energy.

Fig. 18 Enthalpy changes of the cold and hot water, respectively

It is found from the calculation that the air outlet temperature hardly changes as long as the inlet conditions of both water streams are fixed, no matter how the air inlet temperature and relative

humidity vary, displaying a pretty good air temperature control performance. The air outlet temperature can be regulated by adjusting the hot water temperature. For example, the air outlet temperature can be controlled at 25.5 °C nearby at the hot water temperature of 33 °C when other conditions stay unchanged as above (2.62 l/min, 10 °C for the cold water and 2.62 l/min for the hot water).

Sometimes the metabolic moisture generation rate may rise significantly caused by a sudden and considerable promotion of the activity level. The common solution is to increase the air flow rate [49]. The effect of the air flow rate on the dehumidification rate/capacity has been demonstrated by Fig. 14. Here, the thermal loads of recovering the cold and hot water at various air flow rates are calculated, as shown in Fig. 19. It has been known that the dehumidification capacity can be raised by increasing the air flow rate. Nevertheless, the cooling load of the cold water will be increased at higher air flow rate. The balance between the dehumidification capacity and the cooling energy expense should be considered sufficiently. On the contrary, the heating load of the hot water decreases with the increase of the air flow rate. As the air flow rate increases, NTU decreases from 1.37 to 0.42, while NTU_m decreases from 0.93 to 0.24.

Fig. 19 Thermal loads of recovering the cold and hot water, respectively, at various air flow rates

The heat source is sufficient in spacecrafts, e.g. the electrical instruments, so the heating load is not considered here. The cooling load needs to be dealt with by the thermal radiator. In current spacecrafts, the condensing dehumidification is commonly operated at the cold water temperature of 5 °C, and the cooling load is 1.2 kW around for producing 1 kg/h dehumidification capacity. As a result of the heat transfer from the hot water to the cold water indirectly through the air, the cooling loads of the three-fluid membrane dehumidification at various cold water temperature are a little greater than that of the condensing method. However, the three-fluid membrane dehumidification can be operated at higher cold water temperature, which is favorable for the heat dissipation of the radiator. Comprehensively considering the cooling load and the cold water temperature, the three-fluid membrane method will consumes less exergy under most conditions than the condensing method, as shown in Fig. 20. Consequently, the radiator area and weight can be decreased. The exergy is calculated as the following equation.

$$Ex = \frac{t_{ref} - t_{ci}}{t_{ci} + 273.15} Q \quad (23)$$

where Q is the cooling load. The reference temperature t_{ref} is set as the ambient temperature in the cabin.

Fig. 20 Comparison of the exergy between the three-fluid membrane and condensing methods

7. Conclusions

A novel three-fluid membrane dehumidification method is proposed in this paper to apply to the humidity control in manned spacecraft cabin. The cold water provides a low water vapor partial pressure circumstance to create the driving force for the trans-membrane moisture transfer. The water vapor is transported from the air across the membrane and is finally absorbed by the cold water, realizing the dehumidification, water/air separation and water recovery functions simultaneously. With the air temperature maintained at a relatively high value in the help of the hot water, the potential dewing of the humid air is avoided. Without the dewing problem, the cold water temperature can be low enough to promote the dehumidification rate.

With the diffusivity accurately obtained via the diffusivity determination experiment in cooperation with the PSO algorithm, the three-fluid membrane dehumidification is studied experimentally and theoretically. Some conclusions have been drawn as follows:

- (1) Both the humidity ratio and relative humidity of the humid air are reduced, meaning that this dehumidification mode is applicable and safe for spacecraft humidity control;
- (2) The dehumidification rate is affected mainly by the inlet conditions of the air and cold water, while the inlet conditions of the hot water mainly affect the outlet temperature and relative humidity of the air;
- (3) A hypothetical three-fluid membrane module with the membrane area of 1 m^2 is simulated. It is able to handle the metabolic moisture load generated by spacecraft crew of 2-3 astronauts or even more under various ambient conditions, and the dehumidification capacity can be regulated by adjusting the air flow rate. The dehumidification capacity can reach as high as 400-500 g/h.
- (4) Due to the higher operating temperature of the cold water, the three-fluid membrane dehumidification consumes less exergy than the condensing method, indicating superior thermodynamic performance.

On the basis of the present study, we intend to conduct further work in the future, including the fabrication and test of a prototype-scale three-fluid membrane dehumidifier.

Acknowledgements

This research is supported by National Science Foundation of China, No. 51176006.

1

2

3 **Nomenclature**4 A heat and mass transfer area (m^2)5 c_p specific heat capacity (J/kg K)6 Cr heat capacity ratio7 d diameter (m)8 D diffusivity of water vapor (m^2/s)9 h heat transfer coefficient ($\text{W/m}^2 \text{K}$)10 J mass transfer rate (kg/s)11 k mass transfer coefficient (m/s)12 L membrane length (m)13 Le Lewis number14 $\dot{m}$ mass flow rate (kg/s)15 M molar mass (kg/mol)16 NTU number of heat transfer units17 NTU_m number of mass transfer units18 Nu Nusselt number19 p optimal location20 P pressure (Pa)21 Pr Prandtl number22 q heat flux (W/m^2)23 Q heat transfer rate (W)24 r absorption heat (J/kg), restriction factor25 r^* radius ratio of the annular passage26 R ideal gas constant (J/mol K), heat resistance (K/W)27 Re Reynolds number28 ref reference29 Sc Schmidt number30 Sh Sherwood number31 t temperature in Celsius ($^{\circ}\text{C}$)32 v particle velocity33 V parameter value34 w weight factor35 x particle location

36

37 *Greek letters*38 δ membrane thickness (m)39 ε porosity40 η social random number, effectiveness41 θ coefficient of annular passage for Nu calculation42 λ thermal conductivity (W/m K)43 μ dynamic viscosity (Pa s)44 ζ historical random number

1	ρ	density (kg/m ³)
2	σ	residual sum of squares
3	τ	tortuosity
4	ω	humidity ratio (kg/kg),
5		
6	<i>Subscripts</i>	
7	a	air
8	aj	air-side joint
9	c	cold water
10	C_{al}	calculated
11	e	environment
12	Exp	experimental
13	h	hot water, hydrodynamic, historical
14	i	inner
15	k	iteration step
16	K	Knudsen diffusion
17	l	latent
18	m	membrane, mean
19	o	outer
20	p	pore
21	P	Poiseuille flow
22	s	shell, social, sensible heat
23	t	casing tube
24	v	water vapor
25	vm	water vapor in membrane
26	wj	water-side joint

28 **References**

- 29 [1] H. Zhang, H. Yoshino, Analysis of Indoor Humidity Environment in Chinese Residential Buildings,
30 Bldg. Environ., 45 (2010) 2132-2140.
- 31 [2] J. Iverson, B. Butrymowicz, T. Crabb, Advantages of Distributed Humidity and Environmental
32 Control, in: Space 2006, American Institute of Aeronautics and Astronautics, San Jose, California, US,
33 2006.
- 34 [3] B. North, O. Stoll, Space Shuttle Environmental Control and Life Support Systems, in: 7th
35 Annual Meeting and Technical Display, American Institute of Aeronautics and Astronautics, Huston,
36 Texas, US, 1970.
- 37 [4] J. Doyé, A.O. Kohlhase, N. Porth, An Advanced Columbus Thermal and Environmental Control
38 System, in: SpaceOps 2012 Conference, American Institute of Aeronautics and Astronautics,
39 Stockholm, Sweden, 2012.
- 40 [5] D. Westheimer, G. Tuan, Active Thermal Control System Considerations for the Next Generation of
41 Human Rated Space Vehicles, in: 43rd AIAA Aerospace Sciences Meeting and Exhibit, American
42 Institute of Aeronautics and Astronautics, Reno, Nevada, US, 2005.
- 43 [6] W. David, D. Jason, G. Gregory, International Space Station Environmental Control and Life
44 Support System Status: 2009 - 2010, in: 40th International Conference on Environmental Systems,

1 American Institute of Aeronautics and Astronautics, Barcelona, Spain, 2010.

2 [7] W. David, D. Jason, G. Gregory, International Space Station Environmental Control and Life
3 Support System Status for the Prior Year: 2010 - 2011, in: 42nd International Conference on
4 Environmental Systems, American Institute of Aeronautics and Astronautics, San Diego, California, US,
5 2012.

6 [8] K. Zaw, M.R. Safizadeh, J. Luther, K.C. Ng, Analysis of a Membrane Based Air-Dehumidification
7 Unit for Air Conditioning in Tropical Climates, *Appl. Therm. Eng.*, 59 (2013) 370-379.

8 [9] L.-Z. Zhang, Flow Maldistribution and Thermal Performance Deterioration in a Cross-Flow Air to
9 Air Heat Exchanger with Plate-Fin Cores, *Int. J. Heat Mass Transfer*, 52 (2009) 4500-4509.

10 [10] Z.-X. Li, T.-S. Zhong, J.-L. Niu, F. Xiao, L.-Z. Zhang, Conjugate Heat and Mass Transfer in a
11 Total Heat Exchanger with Cross-Corrugated Triangular Ducts and One-Step Made Asymmetric
12 Membranes, *Int. J. Heat Mass Transfer*, 84 (2015) 390-400.

13 [11] G. Ge, M.R.H. Abdel-Salam, R.W. Besant, C.J. Simonson, Research and Applications of
14 Liquid-to-Air Membrane Energy Exchangers in Building Hvac Systems at University of Saskatchewan:
15 A Review, *Renew. Sustain. Energy Rev.*, 26 (2013) 464-479.

16 [12] G. Ge, D. Ghadiri Moghaddam, A.H. Abdel-Salam, R.W. Besant, C.J. Simonson, Comparison of
17 Experimental Data and a Model for Heat and Mass Transfer Performance of a Liquid-to-Air Membrane
18 Energy Exchanger (Lamee) When Used for Air Dehumidification and Salt Solution Regeneration, *Int. J.*
19 *Heat Mass Transfer*, 68 (2014) 119-131.

20 [13] M.R.H. Abdel-Salam, R.W. Besant, C.J. Simonson, Sensitivity of the Performance of a Flat-Plate
21 Liquid-to-Air Membrane Energy Exchanger (Lamee) to the Air and Solution Channel Widths and Flow
22 Maldistribution, *Int. J. Heat Mass Transfer*, 84 (2015) 1082-1100.

23 [14] L.-Z. Zhang, S.-M. Huang, J.-H. Chi, L.-X. Pei, Conjugate Heat and Mass Transfer in a Hollow
24 Fiber Membrane Module for Liquid Desiccant Air Dehumidification: A Free Surface Model Approach,
25 *Int. J. Heat Mass Transfer*, 55 (2012) 3789-3799.

26 [15] Y.-W. Ouyang, L.-Z. Zhang, Conjugate Heat and Mass Transfer in a Skewed Flow Hollow Fiber
27 Membrane Bank Used for Liquid Desiccant Air Dehumidification, *Int. J. Heat Mass Transfer*, 93 (2016)
28 23-40.

29 [16] S.-M. Huang, M. Yang, Heat and Mass Transfer Enhancement in a Cross-Flow Elliptical Hollow
30 Fiber Membrane Contactor Used for Liquid Desiccant Air Dehumidification, *J. Membrane Sci.*, 449
31 (2014) 184-192.

32 [17] L.-Z. Zhang, Progress on Heat and Moisture Recovery with Membranes: From Fundamentals to
33 Engineering Applications, *Energ. Conv. Manage.*, 63 (2012) 173-195.

34 [18] A.K. Pabby, A.M. Sastre, State-of-the-Art Review on Hollow Fibre Contactor Technology and
35 Membrane-Based Extraction Processes, *J. Membrane Sci.*, 430 (2013) 263-303.

36 [19] S.-M. Huang, L.-Z. Zhang, Researches and Trends in Membrane-Based Liquid Desiccant Air
37 Dehumidification, *Renew. Sustain. Energy Rev.*, 28 (2013) 425-440.

38 [20] M.R.H. Abdel-Salam, G. Ge, M. Fauchoux, R.W. Besant, C.J. Simonson, State-of-the-Art in
39 Liquid-to-Air Membrane Energy Exchangers (Lamees): A Comprehensive Review, *Renew. Sustain.*
40 *Energy Rev.*, 39 (2014) 700-728.

41 [21] A.H. Abdel-Salam, C.J. Simonson, State-of-the-Art in Liquid Desiccant Air Conditioning
42 Equipment and Systems, *Renew. Sustain. Energy Rev.*, 58 (2016) 1152-1183.

43 [22] S. Bergero, A. Chiari, Experimental and Theoretical Analysis of Air
44 Humidification/Dehumidification Processes Using Hydrophobic Capillary Contactors, *Appl. Therm.*

1 Eng., 21 (2001) 1119-1135.

2 [23] B. Yang, W. Yuan, X. He, K. Ren, Air Dehumidification by Hollow Fibre Membrane with Chilled
3 Water for Spacecraft Applications, *Indoor Built Environ.*, 25 (2016) 758-771.

4 [24] M.R.H. Abdel-Salam, R.W. Besant, C.J. Simonson, Design and Testing of a Novel 3-Fluid
5 Liquid-to-Air Membrane Energy Exchanger (3-Fluid Lamee), *Int. J. Heat Mass Transfer*, 92 (2016)
6 312-329.

7 [25] M.R.H. Abdel-Salam, R.W. Besant, C.J. Simonson, Performance Definitions for Three-Fluid Heat
8 and Moisture Exchangers, *ASME J. Heat Transfer*, 139 (2016) 022003-022003-8.

9 [26] M.R.H. Abdel-Salam, R.W. Besant, C.J. Simonson, Performance Testing of 2-Fluid and 3-Fluid
10 Liquid-to-Air Membrane Energy Exchangers for Hvac Applications in Cold–Dry Climates, *Int. J. Heat
11 Mass Transfer*, 106 (2017) 558-569.

12 [27] M.R.H. Abdel-Salam, R.W. Besant, C.J. Simonson, Performance Testing of a Novel 3-Fluid
13 Liquid-to-Air Membrane Energy Exchanger (3-Fluid Lamee) under Desiccant Solution Regeneration
14 Operating Conditions, *Int. J. Heat Mass Transfer*, 95 (2016) 773-786.

15 [28] M.R.H. Abdel-Salam, Experimental Study of Effects of Phase Change Energy and Operating
16 Parameters on Performances of 2-Fluid and 3-Fluid Liquid-to-Air Membrane Energy Exchangers,
17 *ASHRAE T.*, 26 (2016) 105-108.

18 [29] J.G. Wijmans, R.W. Baker, The Solution-Diffusion Model: A Review, *J. Membrane Sci.*, 107
19 (1995) 1-21.

20 [30] S. Ulutan, D. Balköse, Diffusivity, Solubility and Permeability of Water Vapor in Flexible
21 Pvc/Silica Composite Membranes, *J. Membrane Sci.*, 115 (1996) 217-224.

22 [31] Y. Li, H. Jia, Q. Cheng, F. Pan, Z. Jiang, Sodium Alginate–Gelatin Polyelectrolyte Complex
23 Membranes with Both High Water Vapor Permeance and High Permselectivity, *J. Membrane Sci.*, 375
24 (2011) 304-312.

25 [32] J. Huang, R.J. Cranford, T. Matsuura, C. Roy, Sorption and Transport Behavior of Water Vapor in
26 Dense and Asymmetric Polyimide Membranes, *J. Membrane Sci.*, 241 (2004) 187-196.

27 [33] L.-Z. Zhang, Fabrication of a Lithium Chloride Solution Based Composite Supported Liquid
28 Membrane and Its Moisture Permeation Analysis, *J. Membrane Sci.*, 276 (2006) 91-100.

29 [34] H. Fan, Y. Peng, Application of PvdF Membranes in Desalination and Comparison of the Vmd and
30 Dcmd Processes, *Chem. Eng. Sci.*, 79 (2012) 94-102.

31 [35] M. Tomaszewska, M. Gryta, A.W. Morawski, Mass Transfer of Hcl and H2o across the
32 Hydrophobic Membrane During Membrane Distillation, *J. Membrane Sci.*, 166 (2000) 149-157.

33 [36] L.-Z. Zhang, A Fractal Model for Gas Permeation through Porous Membranes, *Int. J. Heat Mass
34 Transfer*, 51 (2008) 5288-5295.

35 [37] L.-Z. Zhang, R.-R. Cai, J.-C. Xu, Moisture Transport through Asymmetric Porous Membranes
36 with Finger-Like Holes for Indoor Humidity Control: A Lattice Boltzmann Simulation Approach,
37 *Indoor Built Environ.*, 25 (2015) 151-168.

38 [38] D.-W. Wang, J.-W. Wang, H.-F. Wang, R.-Y. Zhang, Z. Guo, *Intelligent Optimization Methods (in
39 Chinese)*, Higher Education Press, Beijing, 2007.

40 [39] L.-Z. Zhang, An Analytical Solution to Heat and Mass Transfer in Hollow Fiber Membrane
41 Contactors for Liquid Desiccant Air Dehumidification, *ASME J. Heat Transfer*, 133 (2011)
42 092001-092001-8.

43 [40] L.-Z. Zhang, S.-M. Huang, Coupled Heat and Mass Transfer in a Counter Flow Hollow Fiber
44 Membrane Module for Air Humidification, *Int. J. Heat Mass Transfer*, 54 (2011) 1055-1063.

- [41] S.-M. Huang, M. Yang, S. Chen, Effects of the Random Distributions on the Longitudinal Transport Phenomena between an Elliptical Hollow Fiber Membrane Bundle, *J. Membrane Sci.*, 471 (2014) 362-371.
- [42] Ó. Andrjesdóttir, C.L. Ong, M. Nabavi, S. Paredes, A.S.G. Khalil, B. Michel, D. Poulikakos, An Experimentally Optimized Model for Heat and Mass Transfer in Direct Contact Membrane Distillation, *Int. J. Heat Mass Transfer*, 66 (2013) 855-867.
- [43] E.N. Sieder, G.E. Tate, Heat Transfer and Pressure Drop of Liquids in Tubes, *Ind. Eng. Chem. Res.*, 28 (1936) 1429-1435.
- [44] W. Kays, M. Crawford, B. Weigand, *Convective Heat and Mass Transfer*, 4th ed., McGraw Hill Higher Education, New York, 2004.
- [45] A. Ünal, Theoretical Analysis of Triple Concentric-Tube Heat Exchangers Part 2: Case Studies, *Int. Commun. Heat Mass*, 28 (2001) 243-256.
- [46] L.-Z. Zhang, An Analytical Solution for Heat Mass Transfer in a Hollow Fiber Membrane Based Air-to-Air Heat Mass Exchanger, *J. Membrane Sci.*, 360 (2010) 217-225.
- [47] I. Mike, C. Weibo, B. Grant, P. Scott, Nonventing Thermal and Humidity Control for Eva Suits, in: 41st International Conference on Environmental Systems, American Institute of Aeronautics and Astronautics, Portland, Oregon, US, 2011.
- [48] L.S. Bobe, A.A. Kochetkov, A.S. Tsygankov, S.J. Romanov, P.O. Andreychuk, S. Ju, The Performance of the System for Water Recovery from Humidity Condensate (Srv-K) on Russian Segment of the International Space Station, Iss Missions 1 through 30, in: 42nd International Conference on Environmental Systems, American Institute of Aeronautics and Astronautics, San Diego, California, US, 2012.
- [49] Z. Anton, M.R. Christof, P. Peter, C. Markus, Simulation of the Temperature and Humidity Control System of International Space Station in the Virtual Habitat, in: 43rd International Conference on Environmental Systems, American Institute of Aeronautics and Astronautics, Vail, Colorado, US, 2013.