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Analysis of the effect of turbulence promoters in hollow fiber membrane distillation modules by computational fluid dynamic (CFD) simulations

Xing Yang[1,2], Hui Yu[1,2,3], Rong Wang*[,1,2], Anthony G. Fane[1,2]

1. Singapore Membrane Technology Centre, Nanyang Technological University, Singapore 639798
2. School of Civil and Environmental Engineering, Nanyang Technological University, Singapore 639798
3. School of Chemical Engineering, Sichuan University, China 610065

*Corresponding author at: School of Civil and Environmental Engineering, Nanyang Technological University, 639798 Singapore, Singapore. Tel.: +65 6790 5327; fax: +65 6791 0676.

E-mail address: rwang@ntu.edu.sg (R. Wang).

Abstract

As an extended exploration of process enhancing strategies, nine modified hollow fiber modules with various turbulence promoters were designed and modeled using a two dimensional computational fluid dynamic (CFD) heat-transfer model to investigate their potential in improving heat transfer and module performance for a shell-side feed direct contact membrane distillation (DCMD) system.

With the aids of turbulence promoters, the feed heat-transfer coefficient h_f of the modified modules generally showed much slower decreasing trends along the fiber length compared to the original (unmodified) module. A 6-fold h_f enhancement could be achieved by a modified module with annular baffles and floating round spacers. Consistently, the temperature polarization coefficient (*TPC*) and mass flux distribution curves of these modified modules presented increasing trends and gained an optimal improvement of 57% and 74%, respectively. With the local flow fields and temperature profiles visualized in CFD simulations, it was confirmed that an appropriate selection of turbulence promoters could promote intense secondary flows and radial mixing to improve the shell-side hydrodynamics and enhance heat transfer. Moreover, an increase of flow velocity was used and compared as a conventional approach to improve hydrodynamics. It was found that a well-designed module could bring more significant enhancement for a liquid-boundary layer dominant heat-transfer process.

Finally, the hydraulic energy consumption (*HEC*) caused by the insertion of turbulence promoters or the increase of circulating velocity was compared. Configurations with attached quad spacers or floating round spacers achieved a good compromise between enhanced permeation fluxes and modest *HECs*. Overall, the *TPC* decreases with increasing MD coefficient (*C*) values and operating temperatures; while the thermal efficiency increases dramatically with increasing *C* and operating temperatures in a MD system.

Key words: membrane distillation; computational fluid dynamics; turbulence promoters; heat-transfer resistance; temperature polarization; process enhancement.

1. Introduction

As a promising technology for desalination, membrane distillation (MD) has many attractive features such as high salt rejection, modest operating temperature (50-60ºC), low hydrostatic pressure drop and relatively low equipment cost. MD is a thermally-driven process, in which a hydrophobic membrane serves as a barrier to separate the hot feed and cold permeate. In this combined mass- and heat-transfer process, water molecules in the hot stream first evaporate at the mouth of membrane pores, then the vapor flows through the membrane matrix until condensation takes places on the cool permeate surface (in the direct contact MD mode). As a result, high-purity water is produced. Despite many attractive characteristics of the MD process and intense lab-scale studies on MD systems, MD has not been widely implemented in industry [1, 2]. The major challenges impeding its applications include the following: developing appropriate MD membranes to prevent membrane pore wetting, enhancing the permeation flux; reducing the energy consumption, and mitigating flow maldistribution and/or poor hydrodynamics and severe temperature polarization (TP) that compromises module performance [3, 4].

In recent years, intensive research has been done to develop better MD membranes, among which only a few highly permeable membranes with large MD coefficients are available [5-11]. In addition to the development of new membrane materials, many researchers have also investigated strategies to improve the MD process such as optimizing operation parameters [12-18] and designing novel modules [19-21] to alleviate the TP phenomenon and enhance permeation flux. However, to date, most of the investigations of MD module design have focused on flat sheet membrane modules [22-26]. On the other hand hollow fiber-based membrane modules have great potential for industrial applications due to their versatility, more compact structure and reduced vulnerability to TP effects [3]. It is well-recognized that by incorporating proper flow alteration aids (e.g. channel design, flow channel spacers or baffles) in flat sheet modules to create secondary flows or eddies, the MD flux can be enhanced and TP phenomenon can be mitigated [12-14, 26, 27]. However, efforts are still needed for configuration designs and hydrodynamic improvements in hollow fiber MD processes [1, 15, 16, 28-31].

In addition to experimental research, computational fluid dynamics (CFD) modeling has been gradually adopted and has proved to be a useful tool in analyzing fluid dynamic behavior in membrane modules [32, 33]. With the benefits of flow-field visualization (including velocity, pressure, temperature and concentration profiles) at any locations in a defined flow channel, CFD modeling can be used to correlate the fundamental mass- and heat-transfer performance with the hydrodynamic behavior and as a result provide guidance for scale-up and industrial applications. Nevertheless, due to the complex coupling of mass and/or heat transfer across bulk fluids and the membrane matrix, prior CFD models of membrane separation processes have adopted simplified methods [32]. For instance, in a membrane-based ventilator system [34, 35], the mass and heat transfer through a membrane and two fluids was treated as a conjugate problem by ignoring phase changes. In a study of an MD system [36], the feed, permeate and membrane were incorporated into the simulation to obtain velocity and temperature fields, but the concentration transport and latent heat induced by evaporation were ignored. Another CFD study of MD flat sheet membrane module design suggested that spacer orientations should have great impact on the heat and mass transfer [37]. However, the heat-transfer model developed in this study was over-simplified being based on non-porous and rigid shell and tube heat exchangers, which are not coupled with the mass transfer and phase changes. Moreover, these prior simulation studies only focused on mass- and/or heat-transfer improvement by designing better flow channels or incorporating spacers for both non-MD and MD flat sheet or spiral wound membrane modules [37-42]. Thus far, CFD analysis for process modeling in hollow fiber MD modules has been limited to our previous work [33, 43].

A recent review of the development of CFD modeling stated that most MD researchers tended to simplify the transmembrane transfer models by ignoring the permeate flow and focusing on the bulk feed flow [32]. Nevertheless, our recent CFD study has proposed an improved heat-transfer model, which couples the latent heat to the energy conservation equation and combines it with the Navier-Stokes equations, to address the transport correlation between the fluids (feed and permeate) and the membrane in a single fiber MD module [33]. Using the same heat-transfer model, a series of numerical simulations were conducted to analyze the

effectiveness of different process enhancement strategies by identifying the controlling local resistances in mass- and hear-transfer processes under laminar flow [43]. It was found that hydrodynamic means showed a significant effect on improving the heat transfer in a hollow fiber module system when the heat-transfer controlling resistance is in the liquid boundary layers, i.e., where highly permeable membranes (with high *C* values) or high operating temperatures were employed.

As an extension of the previous study, the present work focuses on a MD system, in which the liquid-boundary layers play dominant roles in determining the overall heat-transfer resistance, to investigate the potential of incorporating different turbulence promoters into the shell-side flow to enhance hydrodynamic conditions. This analysis has been motivated by our experimental observations on hollow fiber MD [31]. To simulate the effect of turbulence promoters on enhancing process performance, a series of single fiber MD modules with attached annular quad/round spacers, floating spacers and baffles have been structured and modeled under the following conditions: (1) low operating temperatures and large MD coefficient (*C*) values at constant flow velocity; (2) high operating temperatures and varying *C* values at constant flow velocity; (3) varying feed velocity at low operating temperatures. Conditions (1) and (2) are chosen to investigate the effectiveness of various turbulence promoters employed in the modified modules; while condition (3) is to study the effect of feed flow velocity on enhancing the heat transfer in an 'original' (unmodified) module. In these simulations, various performance metrics, including heat transfer coefficients, TP coefficient (*TPC*), mass flux and thermal efficiency, are examined as functions of membrane properties (*C* values) and/or operating temperatures. Finally, a comparison is provided of hydraulic energy consumption (*HEC*) caused by the introduction of various turbulence aids with the original module design.

2. Theory

2.1 Geometric structures and modeling methods in CFD

Two dimensional double precision models were developed using the commercial software

Fluent 6.3 to study the hydrodynamic behavior and the heat transfer process of an original MD hollow fiber module and a series of modified configurations equipped with turbulence promoters of various specifications.

The novel designs, which were adapted from the original single fiber module, all have the same cylindrical housings with regularly distanced annular spacers attached to the membrane outer surface, and/or baffles attached to shell walls, and/or floating spacers in the shell-side chambers. The assumed dimensions of these single fiber modules are 0.25 m length and 0.0095 m housing diameter. For the convenience of CFD modeling, their geometries were assumed to be ideal axially-symmetric structures. Hence, a series of geometric structures for half of the 2D computing domains were built using Gambit® v2.4.6 for these modules, whose local geometric structures (within a length range of 0−0.06 m with respect to the overall length of 0.25 m) are shown Fig. 1 (a)-(b) and (d)-(e); while Fig. 1 (c) shows a local domain amplification to specify the dimensions of the turbulence promoters inserted in a modified module, in which R_{mi} and R_{mo} are the inner and outer radii of the fiber, Δx and Δy are the cross-sectional dimensions of the regularly shaped internals in x and r directions, Lx is the interval between two internals and Ly is the vertical distance between an internal and the membrane outer surface. In these novel configurations, insertions with regular shapes are periodically distributed on the shell sides, through which the feed stream flows (Fig. 1). In this study ten different designs of the insertions have been investigated and their respective specifications are listed in Table 1.

With the geometric structures built, the simulation process was conducted using Fluent, and incorporated a coupled heat-transfer model combining the latent heat involved in evaporation/condensation on the membrane surfaces in the MD process but ignored the influence of the normal mass flow across the membrane matrix. This is because the MD mass flux has a negligible contribution to either the feed or permeate bulk flow when compared to the operating feed flow rate in a single fiber module [33]. Detailed governing transport equations and boundary conditions in the CFD simulation can be found in our previous work [33]. A brief summary of the mathematical models, related boundary conditions and modeling algorithms is given in Table 2. In this study, a laminar model is used to simulate the unaltered

module operated under laminar operating conditions (Re<2000); and a realizable k-ε method is applied to the unaltered module under turbulent conditions (Re>2000) or the modified configurations with insertions.

2.2 Computational domain and grid scheme

In the geometric structures created in Gambit for the current study, quad elements were adopted for all modules except the configuration with round spacers, whose feed chamber is scaled by triangular meshes due to the irregular domain. In the r direction, a grid scale of 5×10^{-6} m was chosen for the bulk permeate (lumen), the membrane and bulk feed (shell); while in the x direction, a grid scale of 1×10^{-4}m was employed. An example of the quad grid configuration was given in our previous work [33]. Smooth membrane surfaces were assumed in the wall boundary conditions due to its much smaller scale than that of a mesh element. The computational accuracy for convergence is 10^{-5} in Fluent.

2.3 Analysis of MD heat-transfer process

With the geometric structures of membrane modules and heat-transfer models built for CFD simulations, the MD related definitions and equations are required for data post-processing. Having a comprehensive heat-transfer analysis provided in our prior study [43], a brief summary of key heat-transfer equations is given in Table. 3.

Generally, the MD heat transport is described in three steps: i) heat is transferred through the boundary layer on the feed side; ii) heat is carried by vapor which transports through the membrane matrix; iii) heat transports through the boundary layer of the permeate. The overall heat-transfer rate across the membrane Q consists of the latent heat associated with evaporation, Q_{MD}, and heat loss through conduction, Q_{HL}. Based on the resistance-in-series model [44], the overall heat-transfer coefficient, K, for a hollow fiber module can be expressed by the heat-transfer coefficients for the feed and permeate h_f and h_p, and the equivalent heat transfer coefficient for the membrane h_m, which is defined as $\left(h_{MD}+h_{HL}\cdot R_{lm}/R_{mo}\right)$ in our previous study [33]. Based on the temperature field (bulk

temperatures T_f & T_p and membrane wall temperatures T_{fm} & T_{pm}) obtained from CFD simulations, the respective local resistances *1/h* and temperature-polarization coefficient (*TPC*) can be obtained.

Energy is a major concern in MD, and the energy consumption can be assessed via three thermally related metrics such as the thermal efficiency (η_h), temperature-polarization coefficient (*TPC*) and hydraulic energy consumption (*HEC*). The η_h represents the fraction of the evaporation heat with respect to the total heat flux [33], and is mainly determined by the MD coefficient *C* and operating temperatures. The *C* is an intrinsic mass-transfer coefficient of the membrane, which is commonly assumed to be a constant with fixed membrane properties and reasonable ranges of operating conditions [45]. The *C* value for the reference membrane used in this study was calculated from a series of single-fiber module tests as 2.0×10^{-7} kg·m^{-2}·s^{-1}·Pa^{-1} [16]. Based on Eqs. (11) and (12), the *TPC* characterizes the actual driving force of the system [3]; while the *HEC* is a new definition in this present study to describe the hydraulic pressure loss per kg distillate generated when waste heat is available. It is used to assess the advantages of applying different strategies to enhance the mass flux and mitigate the TP effect in terms of the pumping electricity cost.

3. Experimental

This section describes measurements and experiments used to validate the CFD simulation model.

3.1 Materials

In the present study, a hydrophobic polyvinylidene fluoride (PVDF) membrane was characterized experimentally. The properties of the PVDF hollow fiber membrane and testing fluids were presented in our previous work [33].

3.2 DCMD experiment

To confirm the validity of the heat-transfer model for varied operating conditions and modified configurations, both the original (base line) and modified (with attached annular quad spacers 0.2 mm×2 mm of Ly =10 and 30 mm, respectively) single-fiber modules were fabricated and tested. These lab-scale MD modules with an effective fiber length of 0.25 m and a membrane area of 0.0011 m^2 were made by potting the PVDF hollow fiber membranes into Teflon housings.

The experimental data for these modules was obtained via a DCMD setup, which was described in previous work [16]. Briefly, both the feed and permeate solutions were cycled through a hollow fiber module in a countercurrent mode. On the shell side, the feed solution (synthetic seawater: 3.5 wt% sodium chloride (NaCl) with conductivity around 60 $ms \cdot cm^{-1}$) was heated (T_{fi} = 327.2 – 360.2 K) and circulated by a peristaltic pump (u_{fi}= 0.060 – 0.285 $m \cdot s^{-1}$, Reynolds number Re_f=836 – 4000 for the original module). On the lumen side, the permeate (pure water, with conductivity around 0.5 $\mu s \cdot cm^{-1}$) was cooled (T_{pi} = 293.85 – 327.15 K by a cooling circulator and cycled by another peristaltic pump (u_{pi}= 0.417 $m \cdot s^{-1}$, Re_p=460). The distillate was collected in an overflow tank sitting on a balance (±0.1 g); the inlet and outlet pressure for the feed side were monitored by pressure transmitters (±0.01 Pa). Hence, based on the operating conditions, laminar conditions were applied to the conservation equations on the permeate side during the simulations; while either laminar or turbulent models were used for the feed flow.

4. Results and discussion

4.1 CFD heat-transfer model verification

The heat-transfer model for the original MD module without promoters presented in Section 2.1 has been verified in our previous study [43]. To further verify its applicability for altered configurations and varied flow velocities, an original and two modified 0.25 m modules (with annular quad spacers inserted) have been tested in the present study. The comparison between the CFD simulation results and experimental data of mass flux and pressure drop is shown in

Table 4, in which both the inlet temperatures (T_{fi} and T_{pi}) of selected systems (varied feed flow velocities and module configurations) and pressure drop (ΔP_f) along the module on the feed sides of modified modules are listed. It can be seen that the simulation results agree very well with the experimental data. The relative errors are within ±5% for both temperature and pressure drop results, which further verify the applicability of this currently-used heat-transfer model for various experimental settings.

4.2 Shift of dominant resistance in MD heat transfer

For scale up and industrial implementation, a qualitative evaluation of the overall/local heat-transfer resistances in a specific MD system is essential in prioritizing the key design parameters that would most affect the process performance. Fig. 2 shows the distributions of local heat-transfer coefficients for the original (base line) module as functions of the fiber length L for selected MD systems with various membrane properties (MD coefficient C) and operating conditions (temperatures T_{fi} and T_{pi}).

The first system in Fig. 2 (a) (small C=2.0×10^{-7} kg·m^{-2}·s^{-1}·Pa^{-1}, low temperatures T_{fi}=327.15K, T_{pi}=293.85K) shows an absolute dominance from the membrane itself in this heat-transfer process, which was the reference case studied previously [43]. With the same operating temperatures but different membrane properties, the second system [Fig. 2 (b), large C=8.0×10^{-7} kg·m^{-2}·s^{-1}·Pa^{-1}, low temperature T_{fi}=327.15K, T_{pi}=293.85K] indicates that both membrane and feed flow play equally dominant roles; the third system [Fig. 2 (c), small C=2.0×10^{-7} kg·m^{-2}·s^{-1}·Pa^{-1}, high temperatures T_{fi}=360.15K, T_{pi}=326.85K] shows the main heat-transfer resistance partially shifts from the membrane to the liquid-boundary layer at the feed side along the module length; while the fourth case [Fig. 2 (d), large C value, high temperature T] indicates that the heat transfer through the feed liquid-boundary layer is the controlling step in heat transfer. Therefore, as an extended exploration of the effectiveness of hydrodynamic enhancement on the shell-side when the liquid-boundary layer controls the heat-transfer process, cases (b), (c) and (d) were chosen as the simulated conditions in this study. Approaches such as the insertion of turbulence promoters and increase of flow velocity

to alter flow geometries, reduce the TP effect and enhance permeation flux will be discussed in the later sections.

4.3 Effect of turbulence promoters at large C & low temperatures

4.3.1 Improvement on heat transfer coefficients

As discussed previously, in an MD system with a highly permeable membrane of C=8.0×10^{-7} kg·m^{-2}·s^{-1}·Pa^{-1} and low operating temperatures T_f=327.15K, T_p=293.85K [Fig. 2 (b)] the heat transfer coefficient of the liquid-boundary layer on the feed-side, h_f, plays an important role in determining the overall resistance. Fig. 3 shows the simulated distributions of heat transfer coefficients h_f and h_p along the module length, respectively, for the original and modified modules with different turbulence promoters.

In Fig. 3 (a) the h_f distribution curves for all configurations show a decreasing trend along the module length, except the slight changes at the entrances and exits. This is due to the build-up of thermal boundary layers along the flow direction. For the original module, the highest value appears at the entrance of the feed side and then decreases along the flow direction until it reaches a plateau when the flow is fully developed. In contrast, the modified modules generally show a convex decreasing trend along the module length, i.e., starting with a relatively small value at the entrance region before hitting the first barrier (attached/floating spacer or baffle), rising to a higher value when the flow crosses this barrier and starting a slight decrease after reaching the second one.

Overall, the original module has the lowest h_f distribution curve, which indicates an average value of 1495 W·m^{-2}·K^{-1}; while the configuration with baffles 0.2×2×10 shows the highest value of 10057 W·m^{-2}·K^{-1}, which is 6.7 times of the original module, followed by the configurations with attached quad spacers 0.2×2×20 and floating round spacers 0.75. It is of interest that the module with attached quad spacers of a smaller interval Lx=10mm (i.e., more spacers) even shows a lower curve compared to that with Lx=20mm. Generally, the h_f curves for the modified modules decrease more slowly than that of the original. This may be because the flow disturbance has greatly delayed the flow development and build-up of thermal

boundary layers along the flow directions. The significant enhancement of heat-transfer coefficients by incorporating turbulence promoters has confirmed the controlling effect of the liquid-boundary layers in this MD system with a large C & low temperatures. More discussions on the flow-field distribution associated with intensified radial mixing, reduced TP and enhanced permeation flux will be presented in a later section.

In Fig. 3(b), the distribution curves of the heat-transfer coefficients on the permeate side h_p shows a similar trend to the h_f of the original module, i.e., the highest values appear at the entrances of the permeate side (L=0.25) and then decrease along the flow directions until a plateau reached. However, the difference between the original and modified modules is negligible, due to similar hydrodynamics. This observation is consistent with the explanation for Fig. 2(b), which shows that the heat transfer on the permeate side is not a controlling step.

4.3.2 Temperature-polarization mitigation and flow-field visualization

Since the introduction of certain turbulence promoters made a significant improvement in heat transfer coefficient h_f, it is anticipated that the TP effect would be reduced due to the enhanced heat transfer. To explore the ability of TP prevention of various turbulence promoters in the same MD system (Fig. 2 (b)), Fig.4 shows the simulated *TPC* distribution curves along the module length L for both original and ten different modified single fiber modules (i.e., listed in Table 1).

It can be seen from Fig. 4 that the original module presents a downward U shape profile with the maximum value at the entrance and the lowest at the midpoint of the module then a slow increase towards the exit. This is because the transmembrane temperature difference ($T_{fm} - T_{pm}$) first decreases and then increases due to the opposite thermal boundary-layer build-ups on the feed and the permeate sides: the T_{fm} continues to decrease along its flow direction (x) and T_{pm} first increases and then decreases along the x direction. On the other hand, the *TPC* curves of the modified configurations show an U shape with an upward trend—only a slight decrease when the flow hits the first barrier and then a continuous increasing trend along the module length. The maximum values appear at the exits of the modules. A possible reason is that the

insertion of different turbulence promoters has caused secondary flows in between barriers and a degree of radial mixing in the entire flow channel. The formation of vortices has greatly disturbed the thermal boundary layer build-up on the membrane surface and reduced the temperature difference between the bulk and membrane wall. Therefore, the temperature polarization phenomenon in MD is greatly mitigated due to the effective flow alteration that results in an increased effective driving force.

Overall, the original module has the lowest *TPC* curve; while the design with alternate arrangements of quad spacers and baffles (r=0.75 mm, Ly=0.5mm) shows the highest, which is up to a 45% increase compared to the original module, closely followed by the configurations with baffles, floating round spacers and floating quad spacers 0.2×2×10. Interestingly, the results of quad spacers with the same specifications (Δx=0.2 mm, Δy=2 mm) but different intervals Lx (from 10 to 30 mm) indicate that the configuration with the most spacers inserted (smallest interval of 10mm) is not necessarily a better design. For example, a configuration with Lx of 10 mm shows similar results to that with less spacers (Lx=20 mm) in terms of TP mitigation; while its h_f curve was even slightly lower as shown in Fig. 3(a). This is probably because the over-frequent arrangement has instead caused more liquid stagnant zones that compromise the module performance. However, a further decrease in the number of spacers (Lx=30 mm) resulted in insufficient disturbance and mixing, and there may be an optimum spacing.

Moreover, for modules with quad spacers of the same interval Lx=10 mm, the longer spacers (e.g., Δy=2 mm) are less vulnerable to the TP phenomenon than shorter ones (e.g., Δy=1 mm); while wider spacers (e.g., Δx=0.5 mm) are more vulnerable than narrower ones (e.g., Δx=0.2 mm). This indicates that the shorter and wider spacers are less likely to promote secondary flows in the flow channels. It has negligible contributions to disturb the flow or enhance the heat transfer when the spacers have small dimensions ($\Delta y \leq 2$ mm) on the shell side. Interestingly, a design with attached round spacers (r=0.5 mm) shows negligible improvement in terms of TP alleviation, due to its small diameter and the particular cross-sectional shape that possibly causes stagnation of the passing liquid. Therefore, it gives almost the same average *TPC* result as the original module. Nevertheless, it still shows an upward U shape,

which evidently implies its potential in creating stronger secondary flows with an increased diameter (e.g., floating round spacer 0.75).

To relate the enhanced module performance with the hydrodynamic improvement by employing turbulence promoters of various specifications, Fig. 5 shows the local flow fields and temperature distribution in the modified modules. Since all turbulence promoters are inserted with regular intervals, the velocity profiles along the module length can be seen periodically between every two barriers. Therefore, only local flow fields within a certain range of fiber length (0.105−0.125 m) for modified modules are presented in Fig. 5. The velocity profiles (flow fields) are described by the stream traces and temperature distribution by band colors. These results are consistent with the trends of the heat-transfer coefficients curves shown in Fig. 3 and *TPC* distributions in Fig. 4 for these modified modules. Clearly, in Fig. 5 (a) the attached round spacers (r=0.5 mm) do not show effective disturbance in the bulk flow. As those round spacers are raised (Ly=1.5 mm) and have a larger diameter, stronger secondary flows form in between the barriers and vortices appear near the membrane surface to reduce the thickness of liquid-boundary layers. Similarly, there is no visible altering effect from those short and wide quad spacers (e.g., $\Delta x \times \Delta y \times Lx$=0.5mm×0.5mm×10mm). The secondary flows between spacers become more intense with an increasing Δy [from 0.5 mm to 2 mm in Fig. 5 (b)]. As the gap between membrane surface and spacers Ly increases from 0 to 1.5 mm till the spacers reach the shell wall (i.e., baffles), more vortices form along the x direction and more intense radial mixing is observed from the schemes of flow fields. The flow tends to be more homogenous when an alternate arrangement of attached spacers and baffles 0.2×2×10 is employed. Combined with the simulation results shown in Figs. 3 and 4, in a liquid-film controlled heat-transfer system, the more intense secondary flows and radial mixing will result in reduced thermal boundary layers, alleviated TP effect and hence enhanced heat transfer.

4.3.3 Enhancement of permeation flux

Fig. 6 gives the distributions of mass fluxes N_m along the module length for modules with various turbulence promoters. It is noted that the loss of contact area on the membrane surface

occupied by spacers is accounted for the flux calculation. For the original module, the N_m curve has a similar trend to its *TPC* distribution shown in Fig. 4 — first decreasing and then slowly increasing until the exit. This trend can also be explained by the countercurrent build-ups of the thermal boundary layers on the feed and permeate sides, where the thinnest boundary layers occur at the respective entrances. Consistent with the upward trend of *TPC* curves in Fig. 4, the modified modules show dramatically increasing N_m distributions along the module length.

Overall, the original module has the lowest N_m curve; while the modified modules with alternate arrangements of attached spacers & baffles, baffles and floating quad spacers (0.2×2×10, Ly=1.5mm) achieve the best N_m results, closely followed by the designs with floating round spacers (r=0.75mm, Ly=1.5mm) and attached quad spacers (0.2×2×20); the highest average flux improvement is up to 58% when compared to the original configuration. The modules with shorter and wider quad/ smaller round attached spacers show relatively lower fluxes. This may be due to the insufficient disturbance of the fluid from the radial direction, as displayed in Fig. 5, which shows that the intensity of the secondary flows induced by turbulence promoters of different specifications is consistent with their temperature-polarization mitigating performance (Fig. 4) and permeation flux increment (Fig. 6). Additionally, based on the above discussions of Figs. 3−6, the interval of the quad spacers does not necessarily make a significant difference for enhancing module performance. e.g., both modules with intervals of Lx =10 mm and 20 mm show similar results. In this case selection of the most appropriate design should be based on the least complex fabrication and on hydraulic energy consumption (see section 4.6).

4.3.4 Effect of turbulence promoters on thermal efficiency

Fig. 7 depicts the thermal efficiency η_h distribution along the module length for the original and modified modules with a membrane of large C (i.e., C=8.0×10^{-7} kg·m^{-2}·s^{-1}·Pa^{-1}) and lower operating temperatures (i.e., T_f=327.15K, T_p=293.85K). It is observed that the insertion of turbulence promoters (e.g., baffles or floating round spacers) can only achieve up to 5% improvement on the thermal efficiency compared to the original configuration. These results

show that the magnitude of thermal efficiency is not sensitive to the introduction of turbulence aids; this is because the permeability of the membrane (C value) is a determinant factor for the fraction of effective heat in an MD system.

4.4 Effect of turbulence promoters at high operating temperatures

Based on the discussions of the prior selected MD system [Fig. 2 (b)], an appropriate insertion of turbulence promoters in the feed flow could greatly enhance the module performance when the heat-transfer process is controlled by the liquid boundary layer. The other two selected MD systems [Fig. 2 (c) & (d)], which are operated at high feed/permeate temperatures; also indicate a dominant effect of the heat transferred through the feed-side flow. To further explore the effectiveness of turbulence promoters under different operating conditions, a series of simulations were conducted for these two systems [Fig. 2 (c) & (d). The results are shown in Figs. 8−10, which depict the effects of turbulence promoters (floating round and quad spacers) on the distributions of h_f, TPC and N_m along the module length at high operating temperatures (i.e., T_f=360.15 K, T_p=326.85 K), respectively. These two membrane systems with different C values of 2.0 and 8.0×10^{-7} $kg\cdot m^{-2}\cdot s^{-1}\cdot Pa^{-1}$ are simulated and compared.

In Fig. 8 (a), when C is small (i.e., 2.0×10^{-7} $kg\cdot m^{-2}\cdot s^{-1}\cdot Pa^{-1}$) the heat-transfer coefficient h_f of the modified module with regularly-distanced baffles on the feed side shows a 5.8-fold improvement over the original configuration, closely followed by the module with floating round spacers; while the one with attached quad spacers (0.2×2×10) has a 2.3-fold increase. Similarly, when C increases to 8.0×10^{-7} $kg\cdot m^{-2}\cdot s^{-1}\cdot Pa^{-1}$ [(Fig. 8 (b)] the enhancement of the modified modules is as significant as 6- and 3-fold with the same designs of baffles (or floating round spacers) and attached-quad spacers, respectively. In Fig. 9 (a), the most significant increase of TPC is achieved by the design with floating-round spacers— 30% enhancement compared to the original module, closely followed by a design with baffles; while the same configuration (floating-round spacers) shows a much higher increment of 57% when C increases to 8.0×10^{-7} $kg\cdot m^{-2}\cdot s^{-1}\cdot Pa^{-1}$ [Fig. 9 (b)]. This is consistent with the results of

mass fluxes N_m shown in Fig. 10— the percentages of flux enhancement by modified modules with floating-round spacers are 42% and 74% for the membranes with small and large C values, respectively.

These simulation results for an MD system operated at high operating temperatures shows an absolute control of heat transferred through the liquid boundary layers, regardless of the membrane permeability. Therefore, the module design plays an essential role in achieving a higher water production and better performance. Nevertheless, with a more highly permeable membrane, the heat transfer resistance in the liquid boundary layer is more dominant and hence the enhancement of hydrodynamics with the aid of turbulence promoters would be more effective. Thus, it is important to identify the dominant factors when designing novel module configurations for an MD system.

Based on the above discussions, two three-dimensional charts are given in Fig. 11 to present the relationships between the MD coefficient C and operating temperatures T vs. *TPC* and thermal efficiency. In Fig. 11 (a) the *TPC* is shown decreasing as C and the operating temperatures increase; while in Fig. 11 (b) the thermal efficiency increases dramatically with increasing C and operating temperatures. Obviously, to predict the module performance and assess the process efficiency, considerations should be taken for selecting process parameters based on the membrane properties and available heat sources. For example, a potentially selected MD system (as shown in Fig. 11), which has a medium MD coefficient C=3 − 6 $kg \cdot m^{-2} \cdot s^{-1} \cdot Pa^{-1}$ and relatively mild operating feed temperature of 340 K, can achieved an overall *TPC* of 0.55 and thermal efficiency up to 70%.

4.5 Effect of feed flow velocity in original module without promoters

As a conventional strategy to improve hydrodynamic conditions, an increase of flow velocity to reach turbulence is found to be effective. However, similar to other approaches, its effectiveness may differ from system to system. Fig. 12 shows the effects of feed-flow velocity on the *TPC* and mass flux N_m distributions for the original module with membranes of different C values under operating temperatures of T_{fi}=327.15 K and T_{pi}=293.85 K. Similar

to the distributing trends of modified modules with turbulence promoters, in Fig. 12 (a) the *TPC* distributions for both membranes under laminar condition (i.e., u_{fi}=0.006 m·s^{-1}, Re_f=836 for the original module) show a typical U shape; while the curves for the turbulence conditions [i.e., u_{fi}=0.178 m·s^{-1} (Re_f=2500) and 0.285 m·s^{-1} (Re_f=4000)] present an increasing trend. It is also noted that a further increase of turbulent intensity (i.e., Re_f) does not contribute to better module performance.

In general, the MD system with a smaller *C* (2.0×10^{-7} kg·m^{-2}·s^{-1}·Pa^{-1}, upper curves) has a higher *TPC* but is less sensitive to the velocity change compared to that with a larger *C* (8.0×10^{-7} kg·m^{-2}·s^{-1}·Pa^{-1}, lower curves) as shown in Fig. 12 (a). In the former system the turbulence condition brings 15% *TPC* increase while the latter 25%. This is because the controlling heat-transfer resistance shifts from the membrane itself to the liquid-boundary layers on the feed-side with an increased *C* value under the same operating conditions. Interestingly, at a higher flow velocity u_{fi}=0.178 m·s^{-1}, the original module with a membrane of a smaller *C* (i.e., 2.0×10^{-7} kg·m^{-2}·s^{-1}·Pa^{-1} at T_f=327.15K and T_p=293.85K) gains similar enhancement (i.e., 15%) to that of the modified module with annular baffles investigated at a lower velocity (u_{fi}=0.006 m·s^{-1}) in our previous work [43]. However, compared to the 25% increase by employing a high velocity (u_{fi}=0.178 m·s^{-1}), the *TPC* of the same modified design (i.e., 8.0×10^{-7} kg·m^{-2}·s^{-1}·Pa^{-1} at T_f=327.15K and T_p=293.85K) increased by 42% at the same temperature conditions but a low velocity (u_{fi} =0.006 m·s^{-1}) (Fig. 4). This may be due to the more intense radial mixing and surface renewal effect induced by the turbulence promoters than merely increasing the flow velocity. Thus, the membrane wall temperatures tend to be closer to that of the bulk fluids in a properly modified module.

Similar to the N_m curves shown in Fig. 6, the mass flux distributions in Fig. 12 (b) for the same systems under laminar condition initially decrease and then slightly increase towards the exit of the feed flow; while under turbulent conditions, it shows an increasing trend due to better local mixing and surface renewal effect that led to an increase of driving force at a higher flow velocity. However, the system with a larger *C* (upper curves) has a more dramatic flux increment 47% compared to that with a smaller one (lower curves), which achieves 30% enhancement with the same velocity increase. Yet, it was 32% and 53% for a modified

module with baffles for respective *C* values (Fig. 6). Hence, given the more dominant role that liquid boundary layers play in the heat-transfer system, the appropriate selection of turbulence promoters could bring more significant enhancement in improving the module performance. However, for a more comprehensive evaluation of the different enhancement strategies, the hydraulic energy consumption (*HEC*) is another metric that will be compared.

4.6 Analysis of hydraulic energy consumption (HEC)

As discussed in the previous sections, an appropriate selection and arrangement of turbulence aids (e.g., floating spacers, baffles, high velocity, etc.) would greatly reduce the heat-transfer resistance and enhance the module performance, when the heat transferred through the liquid boundary layer is dominant. Nevertheless, with available waste heat sources, the hydraulic loss caused by the insertion of turbulence promoters or increase of circulating velocity becomes a major concern of energy consumption in MD. Fig. 13 shows the average permeation flux and hydraulic loss as a function of turbulence aids, including all turbulence promoters listed in Table. 1 and varied feed-flow velocity. The original module has a low *HEC* of 6.5 $J\cdot kg^{-1}$ when u_{fi}=0.06$m\cdot s^{-1}$ (Re_f=836, laminar flow) and it dramatically rises to 100.7 $J\cdot kg^{-1}$ when entering the turbulence regime with u_{fi}=0.178$m\cdot s^{-1}$ (Re_f=2500). Among these modified modules operated under the same low feed flow velocity u_{fi}=0.06 $m\cdot s^{-1}$, the alternate arrangement of attached quad spacers and baffles (i.e., Q+B) show the highest *HEC* of 191.5 $J\cdot kg^{-1}$, followed by the baffles (B0.2×2×10) and floating quad spacers (FQ0.2×2×10), which cause drastic pressure rises compared to the attached quad (Q0.2×2×10) and floating round spacers (FR0.2×2×10) with fairly insignificant hydraulic pressure losses (<40 $J\cdot kg^{-1}$).

Although the design with alternate spacers and baffles (Q+B) shows the highest flux increase of 58% over the original module, it has the highest *HEC* result, followed by those with floating quad spacers (FQ0.2×2×10) and baffles (B0.2×2×10). Clearly, the configurations with attached quad (Q0.2×2×10) and floating round spacers (FR0.75) show a good compromise for achieving enhanced vapor fluxes with relatively low *HECs* of 28.9 and 38.4 $J\cdot kg^{-1}$, respectively. Presumably, they have reached the same turbulent conditions by achieving the same flux as the original module with a higher velocity (Re_f=2500), which causes an

approximately 3-fold higher hydraulic loss of 100.7 $J \cdot kg^{-1}$. Therefore, a compromise must be made for an enhanced permeation flux as well as a relatively lower *HEC* in evaluating the module performance.

5. Conclusions

In this study a series of CFD simulations were conducted to investigate the effectiveness of nine different turbulence aids in single fiber DCMD modules using a two dimensional heat-transfer model. Three scenarios were studied — MD systems with highly permeable membranes (large MD coefficient *C*), high operating temperatures and varying flow velocity.

It was found that the enhancement of overall heat-transfer coefficients, h_f, was up to 6-fold with annular baffles and floating round spacers. Consistent with the h_f results, their *TPC* and mass flux N_m distribution curves presented increasing trends and gained respective improvement of 57% and 74%. The improved performance was attributed to the secondary flow and radial mixing of different intensities in the entire flow channel, visualized by the local flow fields and temperature profiles in CFD simulations. In addition, an increase of flow velocity was used as a conventional strategy to compare its effectiveness in improving hydrodynamics. The results showed that a well-designed configuration with an appropriate selection of turbulence promoters could bring more significant enhancement for a liquid-boundary layer dominant heat-transfer system.

Moreover, the hydraulic energy consumption (*HEC*) caused by the insertion of turbulence promoters or increase of circulating velocity was compared. The configurations with appropriate quad spacers or floating round spacers show a good compromise for achieving an enhanced permeation flux with rather low *HEC*, compared to other novel designs or the original module with a high velocity. Overall, in an MD system the *TPC* decreases with increasing *C* values and operating temperatures; while the thermal efficiency increases dramatically with increasing *C* and operating temperatures. To predict module performance and assess the process energy efficiency, considerations should be taken for selecting enhancement strategies based on the membrane properties, process parameters and

availability of waste heat sources.

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Nomenclatures

A	membrane area (m^2)
b	membrane thickness (μm)
C	membrane distillation coefficient of the membrane ($kg \cdot m^{-2} \cdot s^{-1} \cdot Pa^{-1}$)
c_p	specific heat capacity of material ($J \cdot kg^{-1} \cdot K^{-1}$)
h	local heat-transfer coefficient of fluids and membrane ($W \cdot m^{-2} \cdot K^{-1}$)
ΔH_T	latent heat of vaporization of water at temperature T ($J \cdot kg^{-1}$)
K	overall heat-transfer coefficient ($W \cdot m^{-2} \cdot K^{-1}$)
k	thermal conductivity ($W \cdot m^{-1} \cdot K^{-1}$)
Lx	interval between two insertions (m)
Ly	vertical distance between an internal and the membrane outer surface
N_m	transmembrane mass flux ($kg \cdot m^{-2} \cdot s^{-1}$)
P	water vapor pressure (Pa)
ΔP_{fluid}	pressure drop along the module length in the shell side
Q	heat-transfer rate through the liquid film (W)
q	heat flux ($W \cdot m^{-2}$)
q_{MD}	transmembrane latent heat flux ($W \cdot m^{-2}$)
q_{HL}	conductive heat loss ($W \cdot m^{-2}$)
Re	Reynolds number
R_{mi}, R_{mo}	inner, outer radii of hollow fiber (m)
S_h	source term of energy transport equation ($J \cdot m^{-3} \cdot s^{-1}$),

$$S_h = \begin{cases} \dfrac{q_{MD}}{\delta r} \cdot \dfrac{R_{mo}}{R_{mi}} & \text{for } r = R_{mi} \\ -\dfrac{q_{MD}}{\delta r} & \text{for } r = R_{mo} \\ 0 & \text{otherwise} \end{cases}$$

T	temperature (K)
v	velocity of feed or permeate ($m \cdot s^{-1}$)
V	volumetric flow rate of the fluid ($m^3 \cdot s^{-1}$)
u	normalized velocity of feed or permeate ($m \cdot s^{-1}$)
x, r	axial, radial directions in cylindrical coordinate (m)

Δx	cross-sectional dimension of the regularly shaped internals in *x* direction (mm)
Δy	cross-sectional dimension of the regularly shaped internals in *r* direction (mm)

Greek letters

η_h	energy efficiency
$\bar{\bar{\tau}}$	stress tensor (kg·m^{-1}·s^{-1})
	$\bar{\bar{\tau}} = \mu \left[\left(\nabla \vec{v} + \nabla \vec{v}^T \right) - \frac{2}{3} \nabla \cdot \vec{v} I \right]$
μ	viscosity (Pa·s)
ρ	density (kg·m^{-3})
N	grid scale in the *r* direction

Suffix

b	bulk average
f	feed
fm	feed-side membrane surface
m	membrane, or membrane surface
i, *o*	inlet and outlet of fluids
p	permeate
pm	permeate-side membrane surface
HL	heat loss

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