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Experimental study of thermal performance in air gap membrane distillation systems, including the direct solar heating of membranes

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Abstract

Membrane distillation (MD), a thermally driven membrane technology which runs at relatively low pressure and withstand high salinity feed streams, has shown potential as a means of desalination and water purification. This paper focuses on the air gap MD (AGMD) process experimentally with the goal of demonstrating and predicting means of improving the energy efficiency of AGMD systems. In particular, a novel configuration which delivers solar radiation directly to the membrane is investigated using a composite solar-absorbing membrane. The use of reduced pressure in the air gap, for lower diffusion resistance, was also explored. A parameter to relate the performance of a bench-scale experiment with similar membrane and gap size to a production system was developed through the application of previously developed models. Small scale experiments were conducted to verify performance for the novel solar powered configuration and the effect of reduced gap pressure. Experiments demonstrated the efficacy of a solar absorbing membrane to improve the thermal performance of the cycle beyond heating an opaque surface in contact with the feed stream. The results also establish a benefit from the deformation of the membrane into the air gap as a result of hydraulic pressure.

Keywords: Air Gap Membrane Distillation, Energy Efficiency, GOR, Heat and Mass Transfer, Experimentation, Scaling Parameter

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Nomenclature

Roman Symbols

\( A \)  
Area [\( \text{m}^2 \)]

\( B \)  
Membrane distillation coefficient or membrane flux coefficient [\( \text{kg/m}^2\text{Pa s} \)]

\( C \)  
Simplified Antoine Equation constant

\( c_p \)  
Specific heat capacity at constant pressure [\( \text{J/kg K} \)]

\( d_{\text{gap}} \)  
Air gap width [\( \text{m} \)]

\( D_{w-a} \)  
Diffusion coefficient of water in air [\( \text{m}^2/\text{s} \)]

\( h \)  
Specific enthalpy [\( \text{J/kg} \)]

\( h_t \)  
Convective heat transfer coefficient [\( \text{W/m}^2\text{K} \)]

\( h_{fg} \)  
Latent heat of evaporation [\( \text{J/kg} \)]

\( I \)  
Solar Irradiation Flux [\( \text{W/m}^2 \)]

\( J \)  
Vapor flux through membrane [\( \text{kg/m}^2\text{s} \)]

\( \dot{m} \)  
Mass flow rate [\( \text{kg/s} \)]

\( M_w \)  
Molecular weight of water [\( \text{kg/mol} \)]

\( P \)  
Total pressure [\( \text{Pa} \)]

\( p \)  
Partial pressure [\( \text{Pa} \)]

\( \dot{Q} \)  
Heat flow [\( \text{W} \)]

\( q \)  
Heat flux [\( \text{W/m}^2 \)]

\( \bar{R} \)  
Universal gas constant [\( \text{K} \)]

\( \bar{T} \)  
Mean temperature [\( \text{K} \)]

\( T \)  
Temperature [\( \text{K} \)]

\( x \)  
Mole fraction

\( z \)  
Lengthwise coordinate [\( \text{m} \)]

Greek Symbols

\( \alpha \)  
Thermal diffusivity [\( \text{m}^2/\text{s} \)]

\( \delta \)  
Thickness [\( \text{m} \)]

\( \Psi \)  
Non-dimensional scaling parameter

\( \rho \)  
Density [\( \text{kg/m}^3 \)]

Subscripts

\( a \)  
Air gap

\( b \)  
Bulk

\( \text{bot} \)  
Bottom temperature

\( c \)  
Condenser stream

\( f \)  
Feed

\( i \)  
Condensate film interface

\( l \)  
Liquid phase

\( m \)  
Membrane

\( p \)  
Permeate

\( \text{sat} \)  
Saturation
<table>
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<tbody>
<tr>
<td>(top)</td>
<td>Top</td>
</tr>
<tr>
<td>(v)</td>
<td>Vapor phase</td>
</tr>
<tr>
<td>(w)</td>
<td>Water (liquid)</td>
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1. Introduction

Membrane distillation is a separation process in which a hot feed stream is passed over a microporous hydrophobic membrane. The temperature difference between the two sides of the membrane leads to a vapor pressure difference that causes water to evaporate from the hot side and, pass through the pores to the cold side. The vapor is pure water which can be condensed. This process has application to desalting water. Compared to reverse osmosis, MD does not require a high pressure feed, and can process very high salinity brines. Compared to other large thermal processes, it can be easily scaled down. Demonstrated pilot plants have been used at a small scale (0.1 m$^3$/day), including stand-alone systems disconnected from municipal power or water networks [1, 2, 3, 4].

MD systems can be used in many configurations; direct contact (DCMD), air gap (AGMD), vacuum (VMD), and sweeping gas (SGMD). All of these configurations can be applied to seawater and brackish water desalination [5, 6]; however, those most commonly used for desalination are DCMD, AGMD, and VMD. In AGMD, an air gap separates the membrane from a cold condensing plate which collects vapor that moves across the gap. Air gap sys-tems have been tested experimentally. AGMD in particular offers promise as a desalination technology with high energy efficiency as it has favorable heat transfer characteristics. The insulation properties of the air gap prevent direct thermal loss between hot and cold sides and the built in condenser surface allows fluid to be condensed at the local saturation temperatu-re instead of being mixed and condensed at the mean saturation temperature as in a VMD system. Creative design improvements and optimization could potentially make AGMD competitive with more established thermal desalination systems.

Most research on MD desalination focuses on maximizing membrane flux, or vapor produced per unit area of membrane. However some studies have examined energy efficiency for experimental plants at the 0.1 m$^3$/day scale [1, 2, 4]. Additionally, more recent MD desalination studies have also examined energy efficiency experimentally [1, 7, 8]. Numerous studies have examined flux in experimental settings [9, 10, 11, 12, 13]. However, using membrane flux as a proxy for thermal performance may not lead to the correct conclusion about overall system performance, as fresh water output and energy consumption can be highly dependent on system configuration, membrane area, system top temperature, and heat recovery from hot brine and condensing vapor. In a complete cycle, the highest flux may not lead the best use of energy, as it often requires high heat inputs and the resulting high vapor flux can increase resistance to heat and mass transfer, driving up energy use.

In this paper, an experiment was devised to test the AGMD system in the context of a complete thermal cycle, with the goal of assessing its energy efficiency. The experiment allows for the assessment of the impact of the AGMD system to improvements in energy efficiency in two ways; by delivering heat directly to the membrane where the water evaporates by means of solar energy as described in previous work [14, 15], and reducing resistance to mass transfer by reducing the total pressure in the gap. The results of the experiment can be scaled up to relate them to the performance of a production-scale system through a scaling parameter developed in this study.

The use of direct heating on the membrane to eliminate temperature polarization was exper-
imentally tested by Hengl et al. [16]. Heating was delivered using an electrically resistive metal membrane which would be impractical to use in a larger scale system. Energy efficiency performance was not measured. Chen and Ho [17] used uniform solar flux to heat the feed stream by placing a solar absorbing surface above the feed stream. This method still retained the temperature polarization effect, but captured the idea of integrating solar collection and desalination into one unit. The feature that strongly distinguishes the system tested in this study from others developed in the past, is a solar absorbing membrane that sits below the water layer. The membrane used in this study is a composite with a hydrophilic polymer such as polycarbonate or cellulose acetate, layered on top of a standard MD membrane material, like Teflon (PTFE). Experimental tests of reducing the pressure inside the membrane gap were attempted previously [18] at very low vacuums (approximately 9/10ths of atmospheric pressure) and only a small enhancement of mass transfer was reported.

2. Experimental Scaling

To understand how the energy efficiency performance of a bench-top experimental system relates to that of a large-scale production system of the air gap membrane distillation (AGMD) configuration shown in Figure 1, it is necessary to know how the system scales with input parameters such as system size, feed mass flow rate, and operating temperature. The model for an AGMD system consists of many equations and is highly nonlinear, depending in large part on exponential functions of temperature and of the permeate flux itself multiplied by the effects of system size. However, it can be simplified using some order of magnitude estimates derived from the numerical solution to the detailed system of equations developed in previous work by the authors [19].

In analyzing the results of the detailed model the following approximations can be made:

- Heat conduction through the membrane, $q_m$, is negligible. $q_m/J_mh_{fg} < 0.1$. Since the air in the gap has good insulating properties heat flux as a result of conductive losses through the membrane are small compared to the energy carried by the latent heat of the vapor.

- The thickness of the liquid condensate layer in the gap, $\delta$, is negligible. $\delta/d_{gap} < 0.15$. This ratio is even lower at the top of the module where there is a small amount of condensed vapor.

- The change in temperature across the gap along the length of the module is small relative to the absolute temperature (in kelvin) of the vapor in the gap. $(T_{f,b} - T_{c,b})/T_{gap} < 0.05$. For the purpose of calculating the vapor concentration in the gap by means of the ideal gas law, the average absolute temperature can be held constant. For this simplified model, it is fixed to $(T_{top} + T_{bot})/2$.

- The diffusion coefficient of water vapor in air, $D_{w-a}$, is assumed to be constant at a given gap pressure, as it is only a weak function of temperature.
To further simplify the model, the vapor pressure/dewpoint temperature equation, also known as the Antoine Equation, can be simplified as an exponential function with only two constants by fitting data obtained from the full Antoine Equation. For the operational range of MD (25-95 °C) the following equation is obtained:

\[ p_w = C_1 \exp(C_2 T) \]  

where \( C_1 = 1134.8 \text{ Pa} \) and \( C_2 = 0.0473 \text{ 1/°C} \) with an \( R^2 \) value of 0.998. This makes this a good approximation of the vapor pressure as a function of temperature for most MD systems, and makes the model easier to simplify.

The flux across the module will be evaluated at the top of the module where the bulk feed temperature is the top temperature of the cycle, \( T_{top} \). Evaluating the flux starts with the equation for the flux permeating through a membrane:

\[ J_m = B(p_{w,f,m} - x_{a,m} P_a) \]  

The vapor pressure on the feed side can be found in terms of feed membrane temperature, \( T_{f,m} \), and the simplified Antoine Equation, Equation 1, evaluated at that temperature. This membrane temperature is lower than the bulk temperature as a result of temperature polarization. This is estimated by the heat flow out of the feed and heat transfer coefficient, \( h_t \). If the location of the evaluation is the top, the membrane temperature is:

\[ T_{f,m} = T_{top} - \frac{J_m h_{fg}}{h_t} \]  

The sensible cooling of the liquid that becomes the permeate as it travels from the bulk feed to the membrane is neglected, as this is typically less than 0.01\( J_m h_{fg} \) in solutions of the detailed model.

The vapor mass fraction on the gap side (denoted by the subscript \( a \)) of the membrane is a function of the vapor fraction gradient. Since the thickness of the condensate layer is neglected and the concentration \( c_a \) is fixed at the constant average temperature, \( T_{gap} \), the mass fraction is described by:

\[ J_m = \frac{P_a}{(R/M_w)T_{gap}} \frac{D_{w-a}}{d_{gap}} \ln \left( 1 + \frac{x_i - x_{a,m}}{x_{a,m} - 1} \right) \]  

which may be rearranged to solve for \( x_{a,m} \):

\[ x_{a,m} = 1 + \exp \left[ \frac{-J_m(R/M_w)T_{gap}d_{gap}}{D_{w-a}P_a} \right] (x_i - 1) \]  

The vapor fraction at the condensate layer interface, \( x_i \), can be determined using Equation 1 by evaluating it at the interface temperature where \( x_i = p_w(T_i)/P_a \). This interface temperature is also the wall temperature of the condenser, since the condensate layer has no thickness. Since the condenser wall is typically made of a highly thermally conductive material, such as copper, it too has negligible resistance. Therefore the interface tempera-
ture can be related to the coolant temperature by the heat transfer coefficient, which for a coolant channel of the same geometry as the feed channel is also \( h_t \):

\[
T_i = T_{c,b} + \frac{J_m h_{fg}}{h_t}
\]

(6)

To evaluate the flux across the top of the module, the temperature of the coolant at the top of the module must be approximated. Since AGMD systems operate in counterflow, as shown in Figure 1, the temperature at the exit of the coolant channel is increased by the energy of condensation that is absorbed along the full length of the membrane. A simple energy balance, shown in Equation 7, can be used to approximate the coolant exit temperature. Since membrane flux varies along the length, with flux at the bottom of the module approximately 1/3 of that at the top, an average value of flux expressed as a fraction of flux at the top is used. This fraction is approximately 2/5 and is determined by solving the detailed numerical model for a variety of top temperatures, mass flow rates, and system sizes, and taking an average of this ratio. The ratio is a strong function of top temperature, owing to the exponential dependence of the driving pressure on temperature, and a very weak function of system size.

\[
\dot{m}_f c_p (T_{c,b} - T_{bot}) = (2/5) J_m h_{fg} A_m
\]

(7a)

\[
T_{c,b} = \frac{(2/5) J_m h_{fg} A_m}{\dot{m}_f c_p} + T_{bot}
\]

(7b)

It is now possible to back substitute Equations 3 - 7 into each other and then into Equation 2 to obtain an implicit expression for flux, as shown in Equation 8.

\[
J_m = B P_a \left[ \exp \left( -J_m \frac{(\bar{R}/M_w) T_{fg} a_{fg}}{D_{w,a} P_a} \right) - 1 \right] +
\]

\[
BC_1 \exp \left( C_2 T_{top} - J_m \frac{C_2 h_{fg}}{h_t} \right) - BC_1 \left[ \exp \left( C_2 T_{bot} + J_m \frac{C_2 h_{fg}}{h_t} - \frac{(\bar{R}/M_w) T_{fg} a_{fg}}{D_{w,a} P_a} + \frac{C_2 (2/5) A_m h_{fg}}{\dot{m}_f c_p} \right) \right]
\]

(8)

Equations for heat input, GOR, and recovery ratio (RR) in terms of flux at the top of the system can also be obtained:

\[
\dot{Q}_{in} = \dot{m}_f c_p (T_{top} - T_{c,b}) = \dot{m}_f c_p (T_{top} - T_{bot}) - (2/5) J_m h_{fg} A_m
\]

(9a)

\[
\text{GOR} = \frac{(2/5) J_m h_{fg} A_m}{\dot{Q}_{in}} = \left( \frac{\dot{m}_f c_p (T_{top} - T_{bot})}{(2/5) J_m h_{fg} A_m} - 1 \right)^{-1}
\]

(9b)

\[
\text{RR} = \frac{J_m (2/5) A_m}{\dot{m}_f} = \frac{c_p (T_{top} - T_{bot})}{h_{fg}} \left( \frac{1}{\text{GOR}} + 1 \right)^{-1}
\]

(9c)

First, a non-dimensional water production rate based on the driving evaporation potential governed by difference between the top and bottom temperatures is found in the formulation
of Equation 8:

\[ R_P = \frac{BC_1 \exp[C_2 \Delta T]}{m_f} A_m \]  

(10)

where \( \Delta T \) is the temperature difference \( T_{\text{top}} - T_{\text{bot}} \). This captures the operating temperature difference and membrane properties, in the form of \( B \). It is non-dimensionalized by the feed mass flow rate.

From Equation 8 several parameter combinations, or ratios, can be observed, which relate to the energy recovered and effective energy used to drive evaporation:

\[ R_D = \frac{(R/M_w)T_{\text{gap}} d_{\text{gap}}}{D_{w-a} P_a} \]  

(11a)

\[ R_T = \frac{C_2 h_{fg}}{h_t} \]  

(11b)

\[ R_R = \frac{C_2 (2/5) A_m h_{fg}}{\dot{m}_f c_p} \]  

(11c)

To obtain the effect on the driving force, these ratios are multiplied by the total flux, and thus the ratios have units of inverse flux, or s m²/kg.

The energy consumed to drive the evaporation goes with the mass transfer across the gap \( (R_D) \), and temperature polarization in the feed and coolant channels \( (R_T) \). \( R_R \) goes with energy recovered by recapturing the latent heat of vaporization in the coolant stream.

In a membrane distillation system, GOR generally goes up with an increase in flux, driving potential, and energy recovery, with energy recovery being the dominant contribution to GOR for systems of low flux and large membrane size. The first two ratios relate to the resistance to vapor mass transfer (in the form of reduced pressure driving force due to temperature polarization, and mass transfer resistance through the gap). GOR would vary with the inverse of these ratios. The heat transfer coefficient that appears in the temperature polarization ratio is also a function of system size, most strongly the mass flow rate and cross sectional area of the flow channel. Since the relation between system size and heat transfer coefficient varies with the flow regime, a simplified relation between heat transfer coefficient and system size cannot be obtained for all systems. Therefore, the inlet parameters will remain expressed as \( h_t \) and explicitly calculated based on the fixed system size and constant fluid properties.

Given a non-dimensional driving force and several additional parameters relating to the energy consumption and recovery, a non-dimensional parameter relating to the system performance can be postulated by arranging each additional parameter \( (R_D, R_T, \text{and } R_R) \) based on their relative impact on GOR into a non-dimensional ratio and multiplying that ratio by the non-dimensional driving force \( (R_P) \):

\[ \Psi = R_P \frac{R_R}{R_T + R_D} = \frac{BC_1 \exp[C_2 \Delta T]}{m_f} A_m \frac{(2/5) A_m h_{fg}}{\dot{m}_f c_p \left( \frac{h_{fg}}{h_t} + \frac{(R/M_w)T_{\text{gap}} d_{\text{gap}}}{C_2 D_{w-a} P_a} \right)} \]  

(12)
The effect of each parameter can be seen on system performance. The system is more energy efficient when the area is large relative to the feed mass flow rate, the gap is small (reducing the length for vapor diffusion), the heat transfer coefficient is higher, the diffusion coefficient is maximized, and the driving potential for evaporation is high. Some parameter changes, such as providing a large membrane area (i.e., large channel width and long length) can be at odds with increasing heat transfer coefficient (i.e., reducing channel width and increasing Reynolds number for a given feed mass flow rate) and require optimizing the system parameters given cost and constructibility constraints. The trend in $\Psi$ holds for systems of any size. For some small systems, such as the experimental system described in this paper, the effect of temperature polarization and diffusion resistance ($R_T + R_D$) can be much larger than the energy recovery, and as a result these systems perform substantially more poorly and $\Psi$ is small, or less than one.

Since $\Psi$ is developed by arranging the system parameters according to their impact on GOR, in either a proportional or inverse manner, validation is required to confirm that $\Psi$ is an appropriate scaling parameter. To validate $\Psi$, it can be evaluated across a range of MD systems for which GOR is determined using the validated detailed model from a previous work by the authors [19], which completely define the heat and mass transfer processes in an MD system. A set of test cases can be created by varying several design parameters one at a time while keeping the remaining ones constant. The baseline parameters are listed in Table 1. These systems represent all sizes and performance levels, and are not necessarily optimized for GOR. The baseline system is representative of a typical production-scale system.

Each of the design parameters in $\Psi$ is varied, and the resultant GOR and $\Psi$ values are plotted on the same graph. If $\Psi$ is an appropriate non-dimensional parameter describing the AGMD process, then all the points should fall essentially on the same curve. There will be some inconsistencies arising from the approximations made to obtain the simplified model and $\Psi$. Figure 2 shows this relationship for 66 different AGMD systems as modeled with the detailed model [19].

It can be seen that, for most of the design parameters varied, the relationship between GOR and $\Psi$ for each collapses onto the same curve. There is some variation from the model simplification, but most parameters behave with a similar polynomial dependence. The relationship predicted by varying the air gap size, however does not fit, and results in a linear relationship between GOR and $\Psi$. Varying $B$ results in a relationship that is of a higher order polynomial than the rest of the data. $\Psi$ is completely insensitive to the effects of gap pressure, because approximations made in the simplified model result in the gap pressure dropping out in the term $D_{w-a}P_a$. Therefore $\Psi$ cannot be used for comparisons across pressure, gap size, and membrane properties.

For the purposes of comparing the experimental system designed in this paper (Table 2) with production scale systems such as one described in Table 1, the poor prediction of GOR using $\Psi$ for systems of varying gap size and membrane properties can be eliminated when assessing the overall relationship between GOR and $\Psi$. This is because production and bench scale systems can be built with similar (small) gap sizes and the same membrane, but a system with hundreds of square meters of membrane area and multiple kilograms per
second of feed mass flow rate that cannot fit on a lab bench. As a result the test cases generated from Table 1 use the same gap size as the experimental system, 1.55 mm, as well as the same membrane, $B = 16 \times 10^{-7}$ kg/m$^2$ Pa s. When systems of varying gap size and membrane properties are eliminated from calculating the relationship between $\Psi$ and GOR, Figure 3 shows the resulting relationship. The relationship is polynomial in nature in the form of $\Psi = AGOR^n$. A very good fit ($R^2 = 0.995$) is obtained for $A = 7.3$ and $n = 2.9$. The bench-scale experiment is shown for comparison, which is very well predicted by the curve.

Therefore, in comparing the performance of a bench scale experiment to a production scale system with the same membrane and gap size, one would multiply the results from the small system by the ratio $(\Psi_{\text{prod}}/\Psi_{\text{expt}})^{1/2.9}$.

Due to the highly non-linear equations governing the AGMD process, the relationship between $\Psi_{\text{large}}$ and $\Psi_{\text{small}}$ can only be an approximate means of scaling a system of two different sizes provided they have the same gap size, membrane properties, and gap pressure, as the coefficients of $\Psi = AGOR^n$ change significantly with the change in those parameters. For all other design parameters, $\Psi$ can provide a robust description of the process across the parameter range, allowing prediction of the performance of a system of larger size from an experiment with the same membrane properties, gap size, and gap pressure.

3. Experimental Setup

A modular experiment was constructed which could easily be reconfigured depending on the mode being tested. Figure 4 shows the layout with all the components necessary to test the various different configurations.

3.1. Objectives

There were three primary goals of the experimental work:

- Validate models for conventional AGMD, reduced pressure gap, and solar direct heated concepts.
- Establish the importance of solar absorbing membrane in the solar direct heated concept.
- Demonstrate the energy efficiency (GOR) of various concepts.

3.2. Experimental Design

Due to lack of adequate sunlight in the northeastern United States, the system was sized to be illuminated by a bench-top solar simulator for testing of the solar direct heating concept. This constrained the illuminated channel to a 9 x 9 in (23 x 23 cm) square. In order to achieve a long aspect ratio a serpentine channel was designed to fit into the illuminated square as shown in Figure 5.
The resulting channel is nearly 1 m long, and 4.5 cm wide, and is completely contained in the square window. This allows the entire channel to be uniformly illuminated by the solar simulator.

The channels were milled into a 1 inch (2.54 cm) thick Delrin (Acetal) block. The plastic provides thermal insulation of the hot feed and thermal mass to even out temperature variations that result from short time scale changes in mass flow rate in the system. The disadvantage of this design is that the system takes a long time to warm up when adjusting the heat input.

The block containing the coolant channel contains features to access the air gap. A copper plate, which serves as the condenser surface closes the serpentine channel containing the coolant. The condensate flows down along the copper plate between the ribs or around the woven mesh that separates the membrane from the copper plate and forms the air gap. The block through which the coolant flows allows the permeate to pass into it and out of the back of the system. The module is oriented vertically with a slight tilt backward allowing gravity to assist the movement of condensate. The membrane spacer, which contains only two-dimensional features, allowing for easy manufacturing, spans the area between the vacuum port and condensate collection channel.

Both channel blocks, the condenser surface, and the gap spacers are sandwiched between two aluminum plates. The plates contain fittings for connecting the feed stream, extracting permeate, and reducing gap pressure. The front plate also contains an opening to allow a window for solar collection. Figure 6 shows each component in the experiment stack for the solar collecting system. For experiments without solar collection, the glazing frame and thin feed channel are replaced by a 1 inch (2.54 cm) thick Delrin block similar to the coolant channel, but without the condensate collection and vacuum port.

The primary membrane used was the a PVDF unsupported membrane made by Millipore (Immobillon PSQ). It was approximately 170 μm thick and had a nominal pore size of 0.2 μm. The membrane distillation coefficient of $16 \times 10^{-7}$ kg/m$^2$Pa s was taken from prior work by Khayet et al. [11]. For the a reduced pressure test at an absolute pressure of 40 kPa, or 0.4 atm, a polyester supported PTFE membrane made by Pall Life Sciences was used. It too had a nominal pore size of 0.2 μm, but a thickness of 300 μm including its support layer. The membrane distillation coefficient of this membrane was scaled from comparison testing between it and the Millipore PVDF membrane in another setup, and was found to be $23 \times 10^{-7}$ kg/m$^2$-Pa-s.

Once the stack is bolted together, an electric heater is added in-line in the feed flow stream between the coolant outlet and feed side inlet. Due to the area of the heater and the resulting surface heat flux, the top temperature of the system was limited to 70 °C to avoid boiling that occurs on the surface of the heater at higher temperatures. Temperature measurement were accomplished by several threaded T-Type thermocouples from Omega (TC-T-NPT-G-72) using the standard calibration. The probes had an error of ±1 K for the temperature range considered.

A tank containing 10-13 ppt sodium chloride solution provided the feedwater for the experiments, and allowed conductivity to be used as a tracer measurement to detect leaks. The
feed was circulated by magnetic drive pump (Little Giant PE-1.5-MDI-SC). A second pump (Little Giant 3-MDX) circulated fluid through a fan coil radiator to maintain a consistent temperature in the feed tank.

Flow rate was controlled by means of a needle valve placed on the outlet before the brine reject returns to the tank. This valve captures most of the pressure drop in the system, keeping hydraulic pressure consistent inside the feed and coolant channels. The resulting positive hydraulic pressure kept the membrane flat in the channel, preventing non-uniformity in the flow and membrane area. This high pressure also made it easier to remove air that may be trapped in the serpentine channel. Feed flow rate was measured by taking a ballistic measurement, which involved collecting the output of the feed for 10 or 20 seconds and weighting the result on a balance with 0.1 g accuracy. Condensate was also collected and weighted periodically on the same balance to measure permeate flowrate. The completely assembled experiment (conventional heating) is shown in Figure 7.

During an experiment the whole setup was surrounded by 1 in (2.5 cm) thick foam insulation, limiting loss heat transfer coefficient to less than 1 W/m²-K. The operating parameters of the experiment are shown in Table 2.

3.3. Solar Direct Heated Experiment

For experiments conducted with a solar absorbing membrane, the large feed channel block was replaced by a spacer comprising the flow channel and a frame holding two polycarbonate glazing panels which was placed over the flow channel spacer. Feed water passes between the inner glazing panel and membrane, bounded by the feed channel spacer. Figure 8 shows the device being heated by solar energy from a solar simulator.

3.3.1. Composite Membrane Design

To enable a conventional MD membrane to absorb solar energy, the highly reflective hydrophobic membranes needed to be treated in such a way to absorb solar energy or layered with another material that would provide the absorption while the MD membrane provides hydrophobicity. Because of the hydrophobic nature of MD membranes, using a liquid dye would be very difficult, so a composite membrane strategy was employed instead.

A variety of layering materials were first selected for their black-colored appearance and ability to pass liquid. A selection of these materials were then tested in a Perkin Elmer Lambda 950 spectrophotometer for their absorptivity in the UV, visible, an infrared. Figure 9 shows absorptivity as it varies with wavelength. A standard solar spectrum is plotted for comparison.

As a result of this analysis, the Millipore HABP, a hydrophilic nitrocellulose based membrane, was selected. It had consistently high absorptivity across the range, and was highly hydrophilic, promoting the passage of water. Materials with higher absorptivity, such as the coated fiberglass or woven carbon fiber cloth, were far less permeable to water, and would add to the mass transfer resistance. Other hydrophobic membranes such as the mixed cellulose esters (MCE) and polycarbonate track etched (PCTE) membrane saw absorptivity drop sharply outside of the visible range and would fail to capture a large portion of the
solar spectrum. The mass transfer characteristics were captured with the lumped membrane distillation coefficient $B$, where addition of the HABP membrane reduces $B$ by 7%, or to $14.8 \times 10^{-7}$ kg/m$^2$Pa s. The additional membrane treated as an additional conduction resistance, reducing heat transfer through the membrane stack.

3.4. Spacer Experimentation

Due to the sensitivity of the AGMD system to the gap size, the spacer design can have a profound effect on performance. Spacers had to support a membrane adequately under the hydraulic pressure of the feed pressure and any reduced gap pressure without tearing the membrane or allowing it come into contact with the condenser surface. Several different spacer types were tested, with the mesh style proving to be the superior choice. The first iteration of spacer utilized open areas to maximize the area for vapor to pass through. Figure 10 shows the spacer in context of the module. The wider solid portions directly correspond to the division between the vertical parts of the serpentine flow channel.

However, these openings were too wide to maintain even a hydraulic pressure difference over the membrane of 14 kPa without the membrane sagging and contacting the condenser surface. This would limit the area available for vapor flow and lead to a non-uniform gap. The thin ribs in the spacer also suffered from buckling as the module heated up creating even wider gaps between ribs.

The next iteration retained the ribs, but narrowed the ribs and narrowed the gap between them to 3 mm. The bottom of the ribs were disconnected from the spacer frame, eliminating buckling due to thermal expansion. The increased frequency of ribs resulted in a spacer that was approximately 60% open. The spacer is shown in Figure 11.

While the ribs decreased the amount of vapor that could be passed through by masking off 40% of the membrane, it was able to stand up to the hydraulic pressure on the membrane; however, when the gap pressure was reduced, the membrane failed from being excessively stretched around the corners of the ribs.

Since mechanical support was highly important for the module to function at all, the final design was similar to what is used in high pressure RO systems, and utilized a woven mesh. A mesh was selected to be the same material and thickness as a standard polypropylene sheet. A frame was cut using a water jet cutter and a mesh was pressed into the frame. The resulting spacer was slightly less than 50% open, but supported the membrane more uniformly and eliminated sharp corners that could damage the membrane when strong differential pressure was applied. Figure 12 shows the spacer installed.

4. Results

4.1. Validation and Reduced Pressure Gap AGMD

The first set of experiments was conducted to validate previously developed models [19] and to test the improvement of GOR as the pressure in the gap was reduced. Measurements were made at successive heat inputs ranging from 100 to 270 W, at a mass flow rate around 0.0024 kg/sec.
To reduce the pressure in the air gap, the gap was connected to a vacuum pump and a two stage water column for pressure stabilization and measurement. Reduced pressures are stated in terms of fractions of an atmosphere, or atm, as means of comparison to a typical air gap system where the gap is at atmospheric pressure. 1 atm is a standard atmosphere which is 101325 Pa in absolute terms. A schematic diagram of the water column is shown in Figure 13. The first stage of the water column is capable of reducing the gap pressure to 0.7 atm, adding a second stage doubles that and enables pressure reduction to 0.4 atm.

As the heat input was increased, top temperature and mass flow rate increased. The top temperature of the system was limited by the heat flux of the electric immersion heater, shown installed in a tube at the top of the experiment in Figure 7, such that the feed fluid would not boil at the surface of the heater. This limited the top temperature to 72 °C.

Figure 14 shows the variation of GOR with top temperature in the experimental system. A comparable GOR for an optimized large scale system is shown on the right hand axis.

4.1.1. Gap Narrowing Through Membrane Deformation

The results shown in Figure 14 demonstrate a consistent under-prediction of GOR by the model. For a system of a given membrane area and heat transfer coefficient, the analysis in the first part of this paper suggests a strong effect of the gap size on overall performance. Because the membrane is subjected to a hydraulic pressure of up to 14 kPa gage, it will tend plastically deform and sag into the open spaces in the mesh. This has the effect of reducing the gap size, which in turn decreases diffusion resistance across the gap. The maximum amount of membrane depression was measured, and approximating the stretching as uniform over an opening in the spacer, an average membrane depression value was calculated. The model was run again with the gap size reduced by this value. Figure 14 compares the model with membrane stretching to the one without.

The model agrees significantly better with experimental data once membrane stretching is taken into account. However since this is may not be uniform for all runs or membrane types, further model comparisons are made assuming the gap is at the full designed size. In production systems, membrane stretching could be a design feature; improving performance in other systems provided the membrane is sufficiently strong to stretch without tearing on the spacer.

4.1.2. Air Gap Pressure Reduction

Since GOR changes only slightly with a change in pressure up to 0.7 atm, reducing pressure in the gap only slightly will not have a large impact. This effect can be seen in Figure 15, which shows how GOR varies with temperature at various gap pressures. Superimposing the experimental results at atmospheric pressure over Figure 15, pressure reductions for 0.9 and 0.7 atm fall within the experimental error (Fig.16). As a result, reduced pressure experiments were conducted at pressures below 0.7 atm. Tests were done at an absolute pressure of 0.4 atm with the supported Pall PTFE membrane. The experiments are plotted in Figure 17.

Evident in these results are the differences between the membranes tested. The supported membrane did not stretch into the open spaces in the mesh, but did exhibit a constant leak rate under mechanical pressure. This leak rate was measured by taking the exact
flow conditions as a typical test, but with the heater turned off. Verification by salinity measurements confirmed the collected fluid in these tests was entirely undiluted feed. The leakage rate was subtracted to obtain the results shown.

The benefits of pressure reduction can be more easily seen by comparing the recovery ratio, or non-dimensional product flow rate, and reduced pressure. These data are plotted in Figure 18. This plot shows a clear benefit for reducing pressure in the gap as a means of enhancing energy recovery and increasing water output.

4.2. Solar Direct Heated System

Tests of the novel directed heated concepts were performed on the bench-scale experiment. In this system, the top temperature has less of an effect on system performance than the solar irradiation flux. As the heating is distributed, the top temperature does not necessarily occur at the inlet or outlet of the feed channel, and does not occur in the bulk fluid stream. As conventionally heated systems of any size operate at similar top temperature, small and large solar heated systems, are operated at comparable solar irradiation flux. The solar flux has a greater impact on temperature, and thus permeate flux, at lower feed flow rates.

Given the coupling of membrane area and solar collector area, membrane flux, \( J_m \), is directly related to GOR:

\[
GOR = \frac{J_m h_f g A_m}{I A_m} = \frac{J_m h_f g}{I} \quad (13)
\]

Systems with comparable membrane flux have comparable energy efficiency, as they are only related by the constant factor \( h_f g \), which is the same for a system of any area. Representing the result in terms of permeate flux is clearer, because it is a directly measurable quantity and the experimental error is confined to the measurement error in flux alone, instead of being combined in the uncertainty in \( I \). This serves the purpose of validating the models for direct heating of the membrane as well.

The solar simulator used (Newport 92190) provided a uniform (within 10%) solar flux, \( I \), up to 750 W/m\(^2\) for a 9 x 9 in square. Tests were also done at 580 W/m\(^2\) using a lower lamp power supply setting.

Figure 19 shows tests conducted at both irradiation fluxes across a range of mass flow rates. In all the subsequent plots, the mass flow rates are normalized to the mass flow rate needed for the highest temperature in the system to be within 1 °C of the seawater inlet temperature, effectively providing no driving potential for evaporation, at the highest solar flux (750 W/m\(^2\)).

The experiments show broad agreement with the model. The narrow range of tests is primarily owing to the small scale of the experiments. Higher temperatures require lower mass flow rates, which are harder to control consistently with manually adjusted valves. Higher mass flow rates cause the feed temperature to drop, approaching the feed water inlet temperature, and eliminate the distinction between different solar fluxes and decreasing permeate flowrate. Using a higher irradiation by bringing the simulator closer to the membrane would result in greater non-uniformity in flux, heating the inner channels significantly more than the outer ones. Calibration of the simulator for an average heat input of 1.8
kW/m² resulted in the outer channels being heated at an average of 25% less than the inner ones. Further reducing solar flux by reducing power to the simulator lamp would also cause the system top temperatures to approach the feed water inlet temperature, resulting in low permeate flowrate.

The range of operating parameters shown in Figure 19 can provide a basis of comparison to demonstrate the importance of heating the feed at the point of evaporation. This model will be compared to several other solar heating configurations, schematically shown in Figure 20. The first shows heating the composite solar absorbing membrane directly, the second shows heating an opaque surface over the fluid, much like a flat plate or tube solar collector would heat water. The last is direct illumination of a plain white MD membrane, where most of the solar energy would be reflected back with some being absorbed into the fluid stream.

Figure 21 compares the performance of different heating schemes. From these experiments it is clear that heating at the membrane surface with a solar absorbing membrane is superior for reasons of both increased flux and increased energy efficiency. When compared to heating above the fluid stream (with a coated absorber plate), the effect of eliminating the heat transfer resistance from the bulk fluid stream to the membrane is clearly evident. There are also additional losses as the coated opaque absorber is closer to the environment. The effect of a solar radiation absorbing surface on the membrane is also apparent in this comparison. Since three times as much energy is absorbed at the surface of the composite membrane than in the bulk fluid stream, a membrane without a solar radiation absorbing layer would reflect most incident radiation resulting in low heating and poor performance.

5. Conclusions

An experimental system was used to successfully test the MD process over a variety of operating temperatures and flow rates. Experiments for both a conventionally heated and solar direct-heated AGMD system showed reasonable agreement with the model.

The performance of systems with different membrane areas, operating temperatures, mass flow rates, and heat transfer coefficients is shown to be related by a single non-dimensional parameter, $\Psi$, derived from a simplified form of the governing equations. A simple polynomial relationship relates performance with $\Psi$, allowing bench-scale experiments to predict the performance for a wide variety of systems, optimized or otherwise, provided that the experiment is performed with the same membrane and the same gap size.

Tests of the solar direct heated system validated the concept, and proved the use of a composite membrane to absorb solar energy where a hydrophilic membrane which absorbs solar radiation can be layered on top of a standard hydrophobic MD membrane. The experimental results generally agreed with the model, and demonstrated the importance of the novel element of absorbing solar energy at the membrane surface as opposed to an opaque absorber placed above the feed flow channel. Despite limitations of energy input and top temperature imposed by the size of the experiment, the experimental results match the model, and by the model can be used to assess the performance of larger scale systems, demonstrating the viability of this configuration for production-scale desalination. Larger
scale systems, which (owing to the need to absorb solar energy directly onto the membrane) cannot use stacked membrane like more conventional MD systems, can be up to three times more efficient than a simple solar still with the same irradiated area.

Reduced gap pressure shows demonstrated superiority in permeate mass flow rate, and thus energy efficiency in the experiment. Only substantial reductions in pressure offer a significant improvement, with performance increasing over 1.5 times as the gap pressure is reduced from 1 to 0.4 atm. While reducing the pressure allows for the construction of larger gaps without sacrificing energy efficiency, it is more suited to improving the performance of a system with a small air gap. The additional cost of pressure reducing equipment has to be weighed against the performance improvement from a gap pressure reduction, depending on the overall design of the system.

The experiment also revealed important real-world limitations and features of the operation of an MD system:

- Despite the selection of a grid spacer with only 50% open area, there was a slight amount of membrane deformation into the gap. Thus, the model under-predicts the performance of the experiment, which, as a result of the deformation, has an effectively smaller gap size. Reducing the gap size in the model to account for this deformation leads to an increase in performance and more accurate prediction of the experimental results.

- Under reduced pressure, membranes with an integrated support layer worked better, as they did not stretch into the openings in the gap spacer, and agreement with the model was generally better.

- In solar experiments, the amount of energy collected by the device is limited due to the small illumination area. As a result, the feed mass flow rate must be lowered to increase the temperature rise and the resultant permeate flow rate. However, at low flow rates, temperature polarization increases sharply, and the potential for high efficiency goes down.

- Further, reduced flow-rates in the solar experiment are hard to maintain at a consistent operating condition. However, as flow rates increase the low heat input quickly drives the temperature rise in the feed too close to zero, eliminating the potential for evaporation.

6. Acknowledgements

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Table 1: Parameters of a baseline AGMD system from which test cases are generated.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
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<tbody>
<tr>
<td>Mass Flow Rate [kg/s]</td>
<td>1</td>
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<tr>
<td>Inlet Temperature, $T_{bot}$ [°C]</td>
<td>20</td>
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<tr>
<td>Top Temperature, $T_{top}$ [°C]</td>
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<td>$\Delta T$ [°C]</td>
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<tr>
<td>Length [m]</td>
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<tr>
<td>Flow Channel Depth [mm]</td>
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<tr>
<td>Gap Size [mm]</td>
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</tr>
<tr>
<td>Membrane Material</td>
<td>PVDF</td>
</tr>
<tr>
<td>Membrane Dist. Coefficient, $B$</td>
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<tr>
<td>Membrane Pore Size [$\mu$m]</td>
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</tr>
<tr>
<td>Membrane Thickness [mm]</td>
<td>0.2</td>
</tr>
<tr>
<td>Air Gap Spacer Open Area [-]</td>
<td>50%</td>
</tr>
<tr>
<td>Air Gap Spacer Material</td>
<td>Polypropylene</td>
</tr>
</tbody>
</table>
Table 2: Experiment attributes and operating parameters.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
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<tbody>
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<td>Top Temperature, $T_{f,in}$ [°C]</td>
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<td>Flow Channel Depth [mm]</td>
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<tr>
<td>Gap Size [mm]</td>
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</tr>
<tr>
<td>Membrane Material</td>
<td>PVDF or PTFE (0.4 atm gap pressure)</td>
</tr>
<tr>
<td>Membrane Dist. Coefficient, $B$ [kg/m²Pa s]</td>
<td>$16 \times 10^{-7}$</td>
</tr>
<tr>
<td>Membrane Pore Size [µm]</td>
<td>0.2</td>
</tr>
<tr>
<td>Membrane Thickness [mm]</td>
<td>0.2</td>
</tr>
<tr>
<td>Spacer Open Area [-]</td>
<td>50%</td>
</tr>
<tr>
<td>Spacer Material</td>
<td>Polypropylene</td>
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