Experimental Characterisation and Modelling of a Membrane Distillation Module Coupled to a Flat Plate Solar Collector Field
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## Abbreviations

<table>
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<th>Abbreviation</th>
<th>Description</th>
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<tbody>
<tr>
<td>DCMD</td>
<td>Direct Contact Membrane Distillation</td>
</tr>
<tr>
<td>MD</td>
<td>Membrane Distillation</td>
</tr>
<tr>
<td>PGMD</td>
<td>Permeate Gap Membrane Distillation</td>
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# Nomenclature

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
<th>Unit</th>
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<tbody>
<tr>
<td>$T_{\text{evap}_{\text{in}}}$</td>
<td>Evaporator inlet temperature</td>
<td>°C</td>
</tr>
<tr>
<td>$T_{\text{cond}_{\text{in}}}$</td>
<td>Condenser inlet temperature</td>
<td>°C</td>
</tr>
<tr>
<td>$V_F$</td>
<td>Seawater feed flow rate</td>
<td>l/h</td>
</tr>
<tr>
<td>$\text{STEC}$</td>
<td>Specific thermal energy consumption</td>
<td>kWh/m$^3$</td>
</tr>
<tr>
<td>$\text{GOR}$</td>
<td>Gained output ratio</td>
<td>-</td>
</tr>
<tr>
<td>$P_{\text{flux}}$</td>
<td>Permeate/distillate flux</td>
<td>l/(h·m$^3$)</td>
</tr>
<tr>
<td>$\varepsilon$</td>
<td>Effectiveness of internal heat exchange</td>
<td>-</td>
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1. Introduction

1.1 Theory and Background

Water scarcity is an increasing issue for a growing global population which is expected to surpass 8.3 billion people by 2030 [1]. In a 2013 study [2], it was estimated that approximately 700 million people were suffering from water stress and scarcity and this number was expected to rise up to 2.8 billion by 2025. The global demand for freshwater is predicted to rise over 50% by 2050 [1]. Seawater desalination is widely seen as a solution to the depleting freshwater resources, especially in arid and semi-arid regions, and its online capacity has surged from 7 million m³/day in 2000 [3] to today’s installed capacity of 78.4 million m³/day [4]. However, these desalination plants, about 17% and 8% using thermally powered multi-stage flash (MSF) and multi-effect distillation (MED) technology and 70% using electrically driven reverse osmosis (RO) technology, are mostly high capacity, capital and energy intensive plants [3]. Only 3.5% of the global desalination capacity comprises of small scale plants (less than 1000 m³/day) [3] which is what is required in rural areas which, in turn, represent 80% of the population without access to fresh drinking water. Moreover, the three desalination technologies mentioned above require significant amounts of chemicals for smooth operation which poses an additional cost and logistical hassle in rural areas. Hence there is a growing demand for the development of a small scale, chemical free and operationally non-complex desalination technology that can be implemented in rural areas preferably coupled to a renewable energy system.

Membrane distillation (MD) as a desalination method can largely meet the above criteria and its scalability makes it suitable for a wide range of application vis-à-vis operational capacity. While the distillation process itself does not rely on any chemical treatment, depending on the conditions of the inlet feedwater, some pre-treatment might be necessary for a longer operational life. The easy coupling with renewable energy (notably solar thermal energy) make MD especially suitable for stand alone and off grid applications [5]–[7].

1.1.1 Basic theory of Membrane Distillation

Membrane distillation (MD) is a water purification method which uses thermal energy to produce freshwater from otherwise impure water such as seawater, industrial wastewater etc. In the MD process, heat is applied to a flowing feedwater stream to generate water vapour. This water vapour then passes through a hydrophobic membrane which doesn’t permit liquid water molecules to pass through but only volatiles, chiefly water vapour, to penetrate the membrane. On the other side of the membrane the vapour emerges and eventually forms the permeate/distillate.

There is always a liquid-vapour interface on the feedwater side of the membrane (at the membrane pores) as the stream of feedwater is kept in direct contact with the membrane surface. With the inner membrane pore volume having to remain dry, the driving force behind the mass transfer of water vapour is the difference in vapour pressures on either side of the membrane.
Besides the mass transfer through the membrane volume, there is also an active transmembrane heat transfer. Ideally the heat transfer would be solely the latent heat of evaporation as the vapour is generated on the feed side and then released on the permeate side. However, in practical application, besides the latent heat demand, there is also a sensible heat transfer that occurs from the hot feedwater side to the cooler permeate side via conduction through the membrane. As this additional heat transfer doesn’t contribute to the mass transfer of water vapour, it is considered as an undesirable heat loss.

1.1.2 Basis of classification and configuration

While the vapour pressure at the feed side liquid-vapour interface is defined, owing to the direct contact between the feedwater and the surface of the membrane, the vapour pressure at the permeate side liquid-vapour interface (and hence the effective vapour pressure gradient across the membrane) is set by one of the following general approaches [3]:

- Application of a lower temperature on the permeate side liquid-vapour interface
- Application of a sweeping gas on the permeate side of the membrane
- Application of a vacuum on the permeate side of the membrane

The implementation and application of these approaches to create a driving force have contributed in the development of different MD configurations which in turn provides a simple means of classification among MD technologies. Variations in geometric and spatial designs/layouts of the channels and overall module provide an additional method of differentiation and classification. The most common configurations, including the one used in this study, utilise a temperature gradient to establish the vapour pressure gradient and hence the driving force.
1.1.3 Direct Contact Membrane Distillation and Permeate Gap Membrane Distillation

The simplest MD configuration is the direct contact membrane distillation (DCMD) type which can be seen in Figure 1 [5]. In DCMD the feed water is heated before passing through the ‘evaporator channel’ which is so called as it is from this channel that the water vapour is generated through evaporation. The water vapour, generated at the liquid-vapour interface in the evaporator channel, then passes through the membrane before emerging on the permeate side also referred to as the condenser channel as it is in this channel that condensation occurs. The driving force in DCMD owes itself to the temperature difference between the hot feedwater in the evaporator channel and the colder permeate/distillate in the condenser channel. Alternatively, the condenser channel may be defined as the channel which carries the coolant that causes condensation. In the case of DCMD, the coolant is the permeate itself. While DCMD has the advantage of a higher driving force compared to other MD configurations, due to a lower heat transfer resistance (only the membrane itself) between the feedwater and the coolant (permeate), there is a high degree of sensible heat loss across the membrane. In other words, the permeate produced in the condenser channel is unproductively heated up due to conduction of heat through the membrane from the hotter evaporator channel which, furthermore, leads to a decrease in the driving force due to a lowering in the temperature difference in the two channels [3], [8].

An improvement on the DCMD configuration is what is referred to as permeate gap membrane distillation (PGMD) or liquid gap membrane distillation (LGMD) which can be seen in Figure 2 below [9], [10].

In terms of construction, the main difference between PGMD and DCMD is the addition of an impermeable foil/film (referred to as condenser foil in Figure 2) after the membrane on the permeate side. As can be observed from Figure 2, there are three channels in PGMD as against two in the case of DCMD. The liquid permeate is formed between the membrane and the impermeable film in what is referred to as the distillate channel. The coolant, which could be any suitable cooling fluid, flows in the condenser channel. Thus the temperature difference, which creates the driving force, is created by the hot feedwater on one side of the membrane (evaporator channel) and the coolant in the condenser channel. It should be noted that the addition of the extra barrier creates an additional heat transfer resistance, in the case of PGMD, reduces the effective temperature gradient and hence it can never achieve the same driving force and flux as compared to a comparable DCMD system [3].

However the main advantage of PGMD lies in that it separates the permeate from the coolant and hence allows for any fluid to be used as the cooling fluid. This provides the opportunity of using the cold inlet feedwater (before it is heated externally) to be used as a coolant and hence
enables sensible heat recovery. Hence the heat transferred across the membrane, which would have been lost as sensible heat to the permeate in the case of DCMD, can be partly recovered as sensible heat to the inlet feedwater. While this sensible heat recovery in the condenser channel reduces the external heat required to reach a desired temperature in the evaporator channel, the driving force is reduced (due to a reduction in the temperature gradient) as the temperature in the condenser channel increases.

Thus, with respect to the desired extent of internal sensible heat recovery from an overall systems perspective, a compromise must be made between the reduction of heat required to operate the MD system and the rate of distillate production.

1.1.4 Module construction and design

The membrane used in this study was of the spiral wound design which allows for a compact arrangement besides effective internal heat recovery [9]. Internal heat recovery is the heat gained by the cool feedwater in the condenser channel. The construction and layout of a spiral wound membrane with PGMD is as shown in Figure 3 [9].

![Figure 3: Schematic of the spiral wound module concept: (1) condenser inlet, (2) condenser outlet, (3) evaporator inlet, (4) evaporator outlet, (5) distillate outlet, (6) condenser channel, (7) evaporator channel, (8) condenser foil, (9) distillate channel and (10) hydrophobic membrane [9]](image)

1.1.5 Solar thermal application in MD

As mentioned earlier, an external heat supply is needed to drive the MD process. The application of solar thermal collectors to this end is an especially promising area of interest and several experimental studies have been done in solar thermal membrane distillation [5]–[8], [11], [12] for the following reasons. Firstly, arid and semi-arid regions with water scarcity tend to also have good solar irradiation i.e are largely in the ‘solar belt’ and hence there is a positive correlation between water necessity and solar energy available. Normal operating temperatures for MD (in the evaporator channel) are between 60°C and 85°C [8] which is well within the working range of flat plate thermal collectors. Moreover, intermittency in heat supplied, which is highly probable in solar thermal generation, can be tolerated in the MD process as it can sustain fluctuations in external heat flux without damage or necessitating complex control systems.
1.2 Aim

- Undertake a characterisation and performance evaluation of a pre-commercial spiral wound permeate gap membrane distillation unit at various operating conditions
- Develop a simplified mathematical model of the solar thermal membrane distillation system
- Develop general guidelines on how best to operate a solar thermal membrane distillation system with an inertial tank and a storage tank

1.3 Method

The characterisation of the PGMD module was carried out using the experimental setup described in detail below. The experiment work and analysis, to the end of the above-mentioned aims, was carried out at the Plataforma Solar de Almeria (PSA) in Spain. For the design of experiments (DoE), the one-variable at a time method was employed where successive experiments are run with one operating variable (may also be referred to as operating parameter or condition) changed leaving the others constant. In this way, the experiment matrix consists of all combinations of values of operating variables, within their respective ranges. The operating variables to be varied, the range within which they’re varied, their levels within the selected range and the performance parameters to be evaluated were first chosen before the experiment runs were initiated.

In parallel with the characterisation and performance evaluation of the PGMD module, it was also desired to investigate methods of optimum integration and operation of the same with a solar thermal collector system. To this end, a simple mathematical model was developed and, after addition of the technical data from the components of the experiment setup to the model, it was validated using field data. After validation, the model can be used to predict and guide toward the best means of solar integration and potential methods to achieve a steady state operation of the overall system.

Thermophysical properties of the fluids used in this work were determined using validated models and equations from earlier studies. Specifically, propylene glycol (used in the solar loop) from [13], pressurised water (used in the tank loop) from [14] and seawater at 3.5 % salinity (in the MD loop) from [15]. These loops will be investigated more closely in the following section. The thermophysical properties relating to water-water vapour phase change (specifically the latent heat of vaporisation) were calculated using equations from [16].
1.3.1 Experimental Setup

The experimental setup that was utilised for this study is located at the Plataforma Solar de Almeria (PSA) in Spain. The overall system can be considered as consisting of three interacting circuits/loops. The solar loop (Figure 4) consists of the solar collector field whose output is fed to a heat exchanger which transfers the solar heat from the solar loop to the tank loop (Figure 5). In other words, this heat exchanger (henceforth referred to as HX1) is used to charge the tank using solar heat. The solar collector field consists of 4 fixed flat plate collector modules, named LBM 10 collectors and manufactured by Wagner and Co, connected in series with a total aperture area of 40.4 m². The working fluid in the solar loop is propylene glycol with a concentration of approximately 5 %. The air cooler in the solar loop serves to regulate the temperature of the fluid entering HX1. To achieve steady state operating conditions, the air cooler plays an important role in preventing the fluid from rising to a higher temperature than required which may occur in case of excessive incident solar irradiation.

![Figure 4: Hydraulic layout of Solar loop of experimental setup](image1)

![Figure 5: Hydraulic layout of Tank loop of experimental setup](image2)
The tank, in turn, discharges itself to the heat load of the MD unit via the heat exchanger (henceforth referred to as HX2) that connects the tank loop to the MD loop (Figure 6). The tank loop uses water (at 2 bar) as its working fluid.

From these three figures it can be observed how the solar heat is utilised to drive the MD process. While carrying out the experiments it was imperative to maintain the operating parameters at a nearly constant level else the results might have been affected. Chief among these operating parameters were the temperatures at the inlet of the heat exchanger HX2 (from the tank side) and at the inlet of the condenser channel. The tank side heat exchanger inlet temperature was maintained by a combination of two effects. Firstly, a large volume, 1.5 m$^3$, of the tank (which acted as an inertial tank during the experiments) absorbed the fluctuations in solar heat output to provide a near steady outlet temperature from its upper node. Secondly the flow diverter (or tempering valve) provided a faster means of regulation of the temperature to HX2 by appropriate mixture with the return flow from the same heat exchanger.

During the course of the MD operation, as can be seen from Figure 6, the brine from the outlet of the evaporator channel returned back to the seawater tank. This led to an increase in the seawater tank's temperature as the evaporator outlet temperature is always higher than the condenser inlet’s (due to the non-ideal effectiveness of the internal heat recovery process). To offset this temperature gain and ensure that the condenser inlet temperature remained constant, the seawater tank needed to be actively cooled.

Technical details about the solar collector field and other important components, along with some on-site pictures of the same, can be found in Appendix A.
1.3.2 Uncertainties in measurement data

Most of the uncertainties that could affect the final evaluated performance/result lie in the MD loop (see Figure 6). The weighing scale, that measures the final and initial weight of the distillate produced, relied on the uniform weight of the force being applied on it. However, in windy ambient conditions, the tube from the distillate channel to the tank shifted slightly and the final distillate container itself rocked slightly when the wind was violent enough and this was a source of random error in the measurement. Moreover, as was discussed above, the evaporator outlet i.e the concentrated feedwater was returned to the seawater tank while the distillate was collected separately and only returned to the tank after the completion of the experiment. Hence the salinity of the seawater increased continuously over the course of the experiment. However, this is not expected to be a major source of error as the seawater tank was of a large capacity and the separation of 40-45 kg of freshwater (approximately the maximum amount of distillate produced during an experiment run from system start up through steady state conditions) from it should not drastically affect the salinity of the tank. Moreover the MD process is not severely affected by minor changes in the salinity of the inlet seawater [7], [9] which is estimated to increase by approximately 15%.

1.4 Previous Work

In this chapter, the current study will be contextualised with an overview of the previous work that has been done in the field of membrane distillation, particularly spiral wound and permeate gap types, and solar thermal energy systems. As experimental work was a major part of this study, many previous experimental analyses on spiral wound PGMD systems have been reviewed [5], [8]–[10]. In [5] eight completely solar powered spiral wound PGMD systems are developed with daily distillate capacities from 60-150 litre are designed and implemented in five different countries. Solar thermal energy was used to drive the MD process while auxiliary electrical loads (such as from pumps and valve operation) were powered by solar photovoltaics. A major aim of [5] was the implementation of long term, low maintenance and independently operational distillation systems in remote areas.

A theoretical analysis of the performance enhancements achieved after the deaeration of membrane pores using feedwater deaerating was carried out in [10] to confirm the understanding that this leads to a decrease in the molecular diffusion (mass) resistance and hence an increase in the distillate flux. A comprehensive experimental campaign was carried out to test this in practice using both aerated and deaerated feedwater on a spiral wound permeate gap membrane distillation unit. Moreover, several operational parameters such as pressures, feed flow rates, feed water salinities and temperature levels were varied in [10] to quantify their effects on the deaerated system. Production and characterisation of spiral wound permeate gap membrane distillation units (with total membrane areas of 5 m², 10 m² and 14 m²) were carried out in [9]. Module operating points were controlled by the automated experimental test rig which utilised an electrical heater as a heat source. The operating variables in [9] included the condenser inlet temperature, evaporator inlet temperature, feed flow rate and feed water salinity. Mathematical equations and models were used to analyse and discuss in depth the results and system behaviour.
Response surface methodology was used in [8] to mathematically (statistically) model and describe the permeate flux and the specific thermal energy consumption and subsequently perform a dual optimisation of the same two parameters. Detailed, statistically determined design of experiment and optimisation study of a spiral wound permeate gap membrane distillation system was performed besides a validation of the RSM model using further experimentation.

These previous experimental works were especially relevant to the current work as they all utilised spiral wound PGMD modules and hence their experiment results can be validly compared with the results from this study vis-à-vis values of performance indicators, behaviour at different operating conditions etc. Additionally the details and discussions regarding the design of experiments, experiment setup, methodology, methods used and their respective limitations, practical experiences etc are all applicable to the current work and hence were reviewed carefully. In fact the experimental test rig used in [8] was the same as the one used in this study.

For the analysis of the flat plate solar collector field, [17] was used which can be summarised as an in-depth look at solar thermal systems and designs including detailed individual component description, reviews and models, system integration and related considerations. While [17] provides a comprehensive study of different types of solar thermal systems (including concentrating types), it was used exclusively for its analysis of flat plate collector systems and tanks besides some basic formulae for solar angles, incidence angle modifiers etc. The analysis of the solar field with regards to system evaluation and strategies to achieve steady state operation were largely guided by this work.

Besides the experiment work with the MD module, for a more descriptive and complete analysis [3] was referred to. It is a dissertation providing a comprehensive review of membrane distillation technology and its physical and mathematical modelling besides detailing methods and areas of optimisation and experimentation. An uncertainty analysis in process (between different module types) and module experimentation is also carried out. The study was aimed towards a broader thermodynamic analysis of membrane distillation to provide a model-based method for analysis and considers otherwise neglected aspects such as electrical load and economic analysis. This work was used to analyse in more depth the behaviour of the membrane distillation unit as well as provide direction when it came to design of experiments and determination of errors (error analysis).

With regards to the mathematical model used, a simplified method of describing the PGMD module was desired. [18] provided a method to model balanced single pass/stage MD systems (including DCMD and PGMD systems) which was used, in part, in this work. The model in [18] centred around treating the MD module as a counterflow/counter current heat exchanger. For PGMD modules, a relation was derived for the gross output ratio (GOR) and the MD thermal efficiency in terms of the effectiveness of heat exchange and the number of transfer units (NTU). The model was validated against a more detailed discretised computational model and found deviations in GOR and flux below 11%. In the development of the simplified mathematical model in this study, the same concept of heat exchanger effectiveness was used and some of the formulation associated with the same was also utilised in this work.
2. Experiment and Calculations

This chapter will subsequently be divided into two sections, one describing the characterisation and performance evaluation of the spiral wound PGMD module and the second part will look into the development of a mathematical model for the solar loop and extend the same principles to the other loops to produce a simplified model of the overall system.

2.1 Characterisation and Performance Evaluation of PGMD module

Before conducting any characterisation and/or performance evaluation, it is imperative to first and foremost identify the experimental inputs (or operating variables) to adjust, the ranges within which they should be adjusted and the specific levels or values that they should be held at during the experiment run. After this is done the final process outputs/results must be selected as performance indicators for analysis of the operational behaviour and performance of the PGMD module. The design of experiments (DoE) is explained in the following sections.

2.1.1 Selection of operating variables

Based on previous experimental work on spiral wound PGMD modules [8]–[10] as well as physical limitations of the module based on the manufacturer’s specifications, the chosen operating variables to vary were the evaporator inlet temperature ($T_{evap\,in}$), the condenser inlet temperature ($T_{cond\,in}$) and the seawater feed flowrate ($V_F$). The ranges and levels of these variables can be seen in Table 1 below.

<table>
<thead>
<tr>
<th>Operating variable</th>
<th>Range</th>
<th>Levels</th>
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<tbody>
<tr>
<td>$T_{evap,in}$ (°C)</td>
<td>60–80</td>
<td>60, 70, 80</td>
</tr>
<tr>
<td>$T_{cond,in}$ (°C)</td>
<td>20–30</td>
<td>20, 25, 30</td>
</tr>
<tr>
<td>$V_F$ (l/h)</td>
<td>200–400</td>
<td>200, 300, 400</td>
</tr>
</tbody>
</table>

During an experiment run it is essential to keep the operating variables stable however it is near impossible to maintain them at exactly constant values. A tolerance of 2 °C was set for the $T_{evap\,in}$ and $T_{cond\,in}$ (the actual achieved experimental dispersions were much lower) and while no tolerance was formally defined for $V_F$ it was aimed to reduce its dispersion to the maximum extent. The experimentally achieved mean values of the operating variables, along with their maximum and minimum values and standard deviations (for all the experimental runs) can be found in Appendix B.
It was initially desired to use a higher feed flowrate (500 l/h) and while it was possible to conduct some experiment runs at this higher flowrate, within the bounds of the above mentioned maximum allowed dispersion in operating variables (especially $T_{\text{cond.in}}$), it was impossible to carry out the entire set of experiment runs at $\dot{V}_F = 500 \frac{\text{l}}{\text{h}}$ simply because of limitations in the experimental setup. Namely that the cooling capacity of the compression chillers for the seawater tank were insufficient to maintain a near-constant $T_{\text{cond.in}}$; instead there was a gradual rise in its value until it exceeded the experimental tolerances defined earlier.

### 2.1.2 Selection of performance indicators

Performance indicators in thermal distillation processes are well defined, generally unchanged across distillation technologies and ubiquitous in experimental works [6], [9], [10], [19] and the following have been selected in this study;

1. **Permeate flux or $P_{\text{flux}}$ [l/(h·m²)]** : The rate of permeate production per unit area of the membrane. As it is area-specific, the $P_{\text{flux}}$ provides a simple means of comparison between MD modules of different sizes and construction. Note that it is the total membrane area which is used in these calculations. It is given by [3];

$$P_{\text{flux}} = \frac{\dot{V}_p}{A_M}$$  \hspace{1cm} \text{Equation 1}

Where $\dot{V}_p$ is the volumetric rate of permeate production [l/h],

$A_M$ is the total membrane area [m²]

This can be alternatively written, in terms of actual measured variables, as;

$$P_{\text{flux}} = \frac{\dot{m}_p}{\rho_p \cdot A_M}$$  \hspace{1cm} \text{Equation 2}

Where $\dot{m}_p$ is the mass flow rate of permeate production [kg/h],

$\rho_p$ is the density of the permeate [14] at mean condenser temperature [kg/l]

2. **Specific thermal energy consumption or $\text{STEC}$ [kWh/m³]** : The heat energy required to produce a unit volume of permeate. This performance indicator is common among all thermal distillation technologies and is the one of the most widely used benchmarks of thermal performance.

$$\text{STEC} = \frac{Q_{HX2}}{\dot{V}_p}$$  \hspace{1cm} \text{Equation 3}
Where $Q_{HX2}$ is the total energy input required for the distillation process [kWh],

$V_p$ is the total volume of permeate production [m$^3$]

The STEC may also be represented in per unit time quantities (for example power instead of energy). As the experiments are run in steady state conditions, by definition, the quantities are stable over time and hence there is no difference in the STEC when integrating over time. Hence the STEC can be expanded and rewritten, in terms of actual measured variables, as;

$$STE_{C} = \frac{\dot{V}_F \cdot \rho_F \cdot C_{pf} \cdot \rho_p \cdot (T_{evap_{in}} - T_{cond_{out}})}{m_p}$$  \hspace{1cm} Equation 4

Where $\dot{V}_F$ is the volumetric flow rate of the feedwater [l/s]

$\rho_F$ is the density of the feedwater [15] at mean cold-side HX2 temperature [kg/l]

$C_{pf}$ is the isobaric specific heat capacity [15] of the feedwater at mean cold-side HX2 temperature [kJ/(kg·°C)]

$\rho_p$ is the density of the permeate [14] at mean condenser temperature [kg/m$^3$]

$T_{evap_{in}}$ is the evaporator inlet temperature [°C]

$T_{cond_{out}}$ is the condenser outlet temperature [°C]

3. Gained Output Ratio or GOR [ - ] : Analogous to STEC, the GOR can be defined as the ratio of the heat required to vaporise the permeate and the actual heat supplied. It is also a widely used performance indicator and can be expressed as;

$$GOR = \frac{m_p \cdot \Delta h_v}{Q_{HX2}}$$  \hspace{1cm} Equation 5

Where $m_p$ is the total mass of permeate production [kg]

$\Delta h_v$ is the normalised latent heat of vaporisation of water [16] at mean evaporator temperature [kJ/kg]

Similar to the STEC, the GOR may be rewritten in terms of per unit time quantities and may hence be alternately expressed as;

$$GOR = \frac{\dot{m}_p \cdot \Delta h_v}{\dot{V}_F \cdot \rho_F \cdot C_{pf} \cdot (T_{evap_{in}} - T_{cond_{out}})}$$  \hspace{1cm} Equation 6
In addition to these three performance indicators, a fourth parameter is defined to determine the extent of sensible heat exchange, within the MD module, at different operating conditions. While this is not directly a measure of the performance of the MD process, it provides insight into the relative operating dynamics at various operational points. This parameter is the effectiveness of the internal heat exchange (between the evaporator and the condenser channels) and is the ratio of the actual heat exchange to the maximum theoretically possible and is given by [17], [18], [20]:

\[ \varepsilon = \frac{T_{\text{cond}_{\text{out}}} - T_{\text{cond}_{\text{in}}}}{T_{\text{evap}_{\text{in}}} - T_{\text{cond}_{\text{in}}}} \]

Equation 7

Where \( T_{\text{cond}_{\text{in}}} \) is the condenser inlet temperature [°C]

2.1.3 Experiment runs

From the Table 1, it can be inferred that, given that there are three experimental inputs (operating variables) and each having three experimental levels, the total number of experiment runs required for a full characterisation is 27. Experiment runs were of 45 minutes barring the first three runs which were of 60 minutes. Two repetitions of a randomly selected experimental run were also performed, on different days, to observe the variability in the performance indicators over different experiment runs.

In a normal operation, after a stationary state is achieved (i.e., the variables remain stable in value over a period of time) the system is allowed to run for at least an hour before measurements are recorded and the experiment run formally begins. This is to ensure that the MD unit has achieved steady state operation. Over one working day there are no more than 2 experiment runs conducted and between each run at least 2.5 hours are provided to ensure that the second run is not affected by the first.

There are several factors/influences, which are largely uncontrolled, that require time to settle into a steady state of operation. The most significant of these being the thermal and hydrodynamic boundary layers that form on either exposed side of the membrane surface i.e. on the membrane surface in the evaporator channel and distillate channel. Establishing and stabilisation of these boundary layers requires time. A similar consideration must be made for the heat transfer surfaces of the heat exchanger between the tank loop and MD loop. Lastly the thermal capacitance of the MD module, besides the heat exchanger, must also be considered as a contributor to potential unsteadiness in experiment operation. It is for these reasons that the MD module was first operated under stationary state conditions for some period of time before any measurements are made.

The date and time of the experiment runs can be found in Appendix C.
2.1.4 Uncertainty Analysis

An uncertainty analysis was carried out to determine the uncertainty error in the final results/parameters. Such an analysis is crucial, especially in experimental work, to quantify the inaccuracies and validity of experimentally determined outputs.

It is first necessary to calculate the error in measured variables such as temperatures from temperature sensors, volumetric flow rates from flow meters etc. In this study, measured variable errors were divided into experimental errors, instrumental/calibration errors and random errors. The total error of a measured variable ($\Delta z$) was calculated as the root mean square of its experimental, instrumental and random error as [3];

$$\Delta z = \sqrt{\sigma_{\text{exp}}^2 + \sigma_{\text{cal}}^2 + \sigma_{\text{random}}^2}$$  \hspace{1cm} \text{Equation 8}

Where $\Delta z$ is the total error of the measured variable

- $\sigma_{\text{exp}}$ is the experimental error of the measured variable
- $\sigma_{\text{cal}}$ is the instrumental/calibration error of the measured variable
- $\sigma_{\text{random}}$ is the random error of the measured variable

Experimental error is the error that arises due to the dispersion/variation of the measured variable during the experiment run. One standard deviation from the mean value was taken as the experimental error. Of course this error can be minimised by careful operation of the system during the experiment to reduce dispersion and operate within experimental boundary conditions earlier defined (see Chapter 2.1.1). The mean, maximum and minimum values, as well as the standard deviation, of the three main operating variables ($T_{\text{cond}_{\text{in}}}$, $T_{\text{evap}_{\text{in}}}$ and $V_F$) for each experiment run, can be found in Appendix B.

Instrumentation/calibration error arises due to inaccuracies in the measuring device itself and is a physical limitation of the device which cannot be altered/minimised.

Random error is one which quantifies the disruptive effect of a largely uncontrollable environmental factor, such as wind, humidity, cloud cover etc on the measured variable. The magnitude of this error is a rough approximate based on real-time observed random variability in the measured values. This has been briefly discussed in an earlier section (see Uncertainties in measurement data).

The final output parameter, whose uncertainty is to be determined, is a function of some measured and derived variables. As it is a multivariate function, it’s error must account for the individual errors of each independent variable. To this effect, the Gaussian law of error propagation [3] is applied to give a probable error for a complex function $f$ with $n$ independent variables as;
\[ \Delta f(\sum_{i=1}^{n} z_i) = \sqrt{\sum_{i=1}^{n} \left( \frac{\partial f(z_i)}{\partial z_i} \Delta z_i \right)^2} \]

Equation 9

The evaluation of the measured and final parameter errors was carried out for each experiment run using the above equations (Equation 8 and Equation 9). The derivation of the final uncertainty/error equation for each output variable \(P_{\text{flux}}, \text{STEBC}, \text{GOR} \) and \(\varepsilon\) using Equation 9 and substituting the functions from Equation 2, Equation 4, Equation 6 and Equation 7 respectively is shown in Appendix D.

2.1.5 Limitations of current method and comparison with other studies

This study utilised the conventional experimental method or one-variable at a time system where one operating variable is changed while the others are held fixed. Although this results in a simplified design of experiments (DoE), the conventional method necessitates a large number of experiment runs and hence more time for experimentation. Moreover, interactions between the input operating variables, and their implications on the result/performance, are ignored. An alternative to the conventional method is the statistically driven experimental model called Response Surface Methodology (RSM) which was used in [8] to model a spiral wound PGMD module. Using this method, the number of experiment runs required reduced from 27 using conventional methods (as is required in this study) to 16. The main drawback of this method is the complexity related to statistical analysis, such as choice of model order (first-order, second-order etc), mathematical model used, design of experiments (DoE), multivariate regression analyses etc.

Electrical consumption as a performance parameter is often overlooked in studies of MD modules despite it being essential for MD operation [3]. In PGMD modules, the only electrical load is the feed water pump and since it is far lower in magnitude than the thermal load it has been neglected in most studies on PGMD modules including in this study. The electrical consumption rises sharply when feedwater deaeration is used [10] due to the additional use of a vacuum pump. However without an operating feedwater pump, MD operation would not be possible and hence the non-inclusion of electrical consumption, irrespective of its magnitude, should be considered as a limitation. Studies on stand-alone MD systems [5]–[7], for example, have had to consider this electrical load when sizing the photovoltaic (PV) arrays to meet the system’s electrical demand.

The variation of performance with salinity was not carried out in this study. Additionally, the PGMD module could not be tested over its entire working range of operating conditions, as defined by the manufacturer (Appendix A). The maximum permissible feed flow rate \(\dot{V}_F\) of the module was 700 l/h but, as explained earlier (see Chapter 2.1.1), the experimental setup was inadequate to test the module at flow rates above 500 l/h.

Lastly, due to time constraints, the experiment runs could not be repeated sufficiently to cross-check the stability and validity of experiment results over different runs. Repetition would have reduced the experimental error significantly and hence provided more accurate results.
2.2 Mathematical Modelling of overall system

The creation of a simple mathematical model using standard and empirical formulae has been performed in a similar setting [19] i.e with a solar flat plate collector field coupled to a desalination process system. Starting with the solar field (see Figure 4 and Appendix A), the thermal efficiency of the solar flat plate collector ($\eta_{\text{solar}}$) is given by [17], [21];

$$\eta_{\text{solar}} = \eta_0 \cdot K_{\tau \alpha} - \frac{k_1}{G_T} (T_{\text{col}} - T_{\text{amb}}) - \frac{k_2}{G_T} (T_{\text{col}} - T_{\text{amb}})^2$$ \hspace{1cm} \text{Equation 10}

Where $\eta_0$ is the zero-loss efficiency or optical efficiency [%]

- $K_{\tau \alpha}$ is the incidence angle modifier (IAM) [-]
- $k_1$ is the heat loss coefficient [W/(m²·K)]
- $k_2$ is the temperature coefficient of the heat loss coefficient [W/(m²·K²)]
- $G_T$ is the global/total irradiance on a titled plane [W/m²]
- $T_{\text{col}}$ is the average temperature of the flat plate solar collector field [°C]
- $T_{\text{amb}}$ is the ambient/environment temperature [°C]

It should be noted that $k_1$ and $k_2$ are solar collector parameters and are constant for a given flat plate collector (see Appendix A). The incidence angle modifier ($K_{\tau \alpha}$) is a multiplier that addresses the effect of the incidence angle (as a deviation from normal incidence) on the collector plane. It is a function of the incidence angle and is given by [17];

$$K_{\tau \alpha} = 1 - b_0 \left( \frac{1}{\cos \theta} - 1 \right)$$ \hspace{1cm} \text{Equation 11}

Where $b_0$ is the incidence angle modifier coefficient which is a constant for a particular collector

- $\theta$ is the incidence angle (the angle between the beam radiation and the surface normal)

A description of the calculation of angle of incidence is not shown in this work as it requires many intermediary calculation steps and, more importantly, is a small and less relevant part of the overall model. However it has been calculated using standard formulae [17, Ch. 1] in a worksheet and MATLAB environment.
The thermal efficiency may also be written as [17];

\[
\eta_{solar} = \frac{\dot{m}_{solar} \cdot C_{p, col} \cdot (T_{solar_{out}} - T_{solar_{in}})}{A_{solar} \cdot G_T}
\]

Equation 12

Where \(\dot{m}_{solar}\) is the mass flow rate in the solar collector field [kg/s]

\(C_{p, col}\) is the isobaric specific heat capacity of propylene glycol [13] at average collector temperature \(T_{col}\) [J/(kg⋅°C)]

\(T_{solar_{out}}\) is the solar collector field outlet temperature [°C]

\(T_{solar_{in}}\) is the solar collector field inlet temperature [°C]

\(A_{solar}\) is the total aperture area of the solar collector field [m\(^2\)]

To describe the heat transfer across the heat exchanger between the solar and tank loops (HX1) the following equation is used [20];

\[
Q_{HX1} = UA_{HX1} \left( \frac{T_{HX1_{in,solar}} - T_{HX1_{out,tank}}}{\ln \left( \frac{T_{HX1_{in,solar}} - T_{HX1_{out,tank}}}{T_{HX1_{out,solar}} - T_{HX1_{in,tank}}} \right)} \right)
\]

Equation 13

Where \(UA_{HX1}\) is the overall heat transfer coefficient of HX1 [W/°C]

\(T_{HX1_{in,solar}}\) is the temperature at the inlet of HX1 on the solar loop side [°C]

\(T_{HX1_{out,solar}}\) is the temperature at the outlet of HX1 on the solar loop side [°C]

\(T_{HX1_{in,tank}}\) is the temperature at the inlet of HX1 on the tank loop side [°C]

\(T_{HX1_{out,tank}}\) is the temperature at the outlet of HX1 on the tank loop side [°C]

The heat transfer may also be written in terms of the sensible heat gained/lost by the fluids on either side of the heat exchanger as [17], [20];

\[
Q_{HX1} = \dot{m}_{solar} \cdot C_{PHX1solar} \cdot (T_{HX1_{in,solar}} - T_{HX1_{out,solar}}) = \dot{m}_{tank_{HX1}} \cdot C_{PHX1tank} \cdot (T_{HX1_{out,tank}} - T_{HX1_{in,tank}})
\]

Equation 14

Where \(\dot{m}_{tank_{HX1}}\) is the mass flow rate in the HX1 on the cold/tank side [kg/s]

\(C_{PHX1solar}\) is the isobaric specific heat capacity of propylene glycol [13] at mean hot-side HX1 temperature [J/(kg⋅°C)]
\( C_{pHX1\text{tank}} \) is the isobaric specific heat capacity of water (at 2 bar) [14] at mean cold-side HX1 temperature [J/(kg\(\cdot\)\(^\circ\)C)]

The second heat exchanger HX2, between the tank loop and the MD loop (see Figure 5 and Figure 6), can also be modelled in a similar way with the heat transfer across HX2 given by [20];

\[
Q_{HX2} = U_{AHX2} \frac{(T_{HX2\text{in}\text{tank}} - T_{\text{evap}\text{in})} - (T_{HX2\text{out}\text{tank}} - T_{\text{cond}\text{out}})}{\ln\left(\frac{T_{HX2\text{in}\text{tank}} - T_{\text{evap}\text{in)}}}{T_{HX2\text{out}\text{tank}} - T_{\text{cond}\text{out}}\right)} \tag{Equation 15}
\]

Where \( U_{AHX2} \) is the overall heat transfer coefficient of HX2 [W/\(^\circ\)C]

\( T_{HX2\text{in}\text{tank}} \) is the temperature at the inlet of HX2 on the tank loop side [\(^\circ\)C]

\( T_{HX2\text{out}\text{tank}} \) is the temperature at the outlet of HX2 on the tank loop side [\(^\circ\)C]

In terms of the sensible heat gained/lost by the fluids on either side of the heat exchanger [17], [20];

\[
Q_{HX2} = \dot{m}_F \cdot C_{pF} \cdot (T_{\text{evap}\text{in}} - T_{\text{cond}\text{out}}) \\
= \dot{m}_{\text{tank}\text{HX2}} \cdot C_{pHX2\text{tank}} \cdot (T_{HX2\text{out}\text{tank}} - T_{HX2\text{in}\text{tank}}) \tag{Equation 16}
\]

Where \( \dot{m}_F \) is the mass flow rate in the HX2 on the cold/MD side [kg/s]

\( \dot{m}_{\text{tank}\text{HX2}} \) is the mass flow rate in the HX2 on the hot/tank side [kg/s]

\( C_{pHX2\text{tank}} \) is the isobaric specific heat capacity of water (at 2 bar) [14] at mean hot-side HX2 temperature [J/(kg\(\cdot\)\(^\circ\)C)]

From Equation 7 and Equation 16, \( Q_{HX2} \) may also be expressed in terms of effectiveness of internal heat recovery \( \varepsilon \) [18] as;

\[
Q_{HX2} = \dot{m}_F \cdot C_{pF} \cdot (1 - \varepsilon) \cdot (T_{\text{evap}\text{in}} - T_{\text{cond}\text{in})} \tag{Equation 17}
\]

The overall heat transfer coefficient of a heat exchanger \((U_{AHX})\), as given in Equation 13 and Equation 15, is normally determined by the heat exchanger’s physical design/construction which includes the surface area, thickness, length, surface conditions etc of the heat exchange material(s) as well as the flow regime, bulk temperature and thermophysical properties of both fluids besides the unavoidable heat losses that occur during the heat exchange process. A comprehensive modelling of HX1 and HX2, to reflect the variations in the \( U_{AHX} \) value that occur at different operating conditions was beyond the scope of this study and instead Equation
14 and Equation 16 will be relied on more heavily to evaluate the heat exchange in HX1 and HX2 respectively.

To enumerate the variations in the $U_{A_{HX}}$ value at different operating conditions, the same was evaluated for HX2 during the entire experiment campaign. Over the course of all the experimental runs, it was found that;

$$U_{A_{HX2}} = 179.3 \, \frac{W}{°C}$$

$$\sigma_{U_{A_{HX2}}} = 70.4 \, \frac{W}{°C}$$

Where $U_{A_{HX2}}$ is the mean overall heat transfer coefficient in HX2 [W/°C]

$\sigma_{U_{A_{HX2}}}$ is the standard deviation of $U_{A_{HX2}}$ [W/°C]

As can be observed above, a single standard deviation makes up almost 40 % of the mean value which therefore makes the same not a reliable parameter to use in a mathematical model as a constant value. The exact $U_{A_{HX2}}$ values for each experiment run can be found in Appendix E.

Heat losses must also be considered and are most significant in the solar field as it holds the longest piping (at over 40m). To model this heat loss to the ambient the $U_{A}$ value of the pipes is estimated by first determining the heat losses by the fluid, from the point of exiting the solar collectors and the point of entry into the heat exchanger HX1, and equating that to the heat loss through the pipes. The heat lost by the fluid is given by [17], [20];

$$Q_{loss} = \dot{m}_{solar} \cdot C_{p_{SF}} \cdot (T_{solar_{out}} - T_{HX1_{in_{solar}}})$$

Equation 18

The same heat is lost through the pipes and can be expressed as [20];

$$Q_{loss} = U_{A_{pipe}} \cdot \left( \frac{T_{solar_{out}} + T_{HX1_{in_{solar}}}}{2} - T_{amb} \right)$$

Equation 19

Where $C_{p_{SF}}$ is the isobaric specific heat capacity of propylene glycol [13] at mean field pipe temperature [W/°C]

$T_{amb}$ is the ambient temperature [°C]

Of course, as was the case with the $U_{A}$ value estimations of the heat exchangers, using this method to approximately determine a single $U_{A}$ value neglects environmental effects (especially wind speed) and system effects (such as flow velocity, mean fluid temperature etc). However, a detailed analysis of the solar field heat losses was beyond the scope of this study, hence the same method was utilised to experimentally determine a $U_{A}$ value of the pipes in the solar field. The $U_{A}$ value of the pipes was experimentally determined to be;
\[ \overline{UA}_{\text{pipe}} = 18.26 \frac{W}{°C} \]
\[ \sigma_{UA_{\text{pipe}}} = 2.079 \frac{W}{°C} \]

Where \( \overline{UA}_{\text{pipe}} \) is the mean overall heat transfer coefficient in the solar field pipe [W/°C]

\( \sigma_{UA_{\text{pipe}}} \) is the standard deviation of \( UA_{\text{pipe}} \) [W/°C]

From the outlet of HX1 to the inlet of the solar collector field there is a small length of piping with some losses as well. However these are ignored as in most cases the heat from compression which is added by the pump is sufficient to render the net heat lost, to the ambient, negligible.
2.2.1 Modelling the MD loop

The MD loop itself cannot be modelled without experimentation on it first to observe and analyse its performance and behaviour at different operating conditions. While doing the same, it was observed that the performance indicators, namely the $P_{\text{flux}}, STEC, GOR$ and $\epsilon$, had a behaviour predictable with changing operating conditions. A regression analysis was performed, with the results from the experiment runs (see Table 4 and Table 5), to model the performance indicators as functions of the varied operating conditions i.e $T_{\text{cond, in}}, T_{\text{evap, in}}$ and $\hat{V}_F$.

Experimental data was fitted to a quadratic function in [8], although only the $P_{\text{flux}}$ and $STEC$ were modelled in that study, as it was found to have the best fit for the given data. In this study it was similarly observed that a quadratic function best fit the experimental data for the $P_{\text{flux}}$ and $STEC$ as well as the $GOR$ while in the case of the $\epsilon$ a linear regression model provided the best fit.

The regression analysis was done using MATLAB and for the linear regression equation;

$$y = y_0 + y_1 \cdot T_{\text{cond, in}} + y_2 \cdot T_{\text{evap, in}} + y_3 \cdot \hat{V}_F$$

Equation 20

the coefficients for $\epsilon$ were calculated and are presented in Table 2;

<table>
<thead>
<tr>
<th>Response Coefficient</th>
<th>$\epsilon$</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Estimate</td>
</tr>
<tr>
<td>$y_0$</td>
<td>0.79723</td>
</tr>
<tr>
<td>$y_1$</td>
<td>0.00034719</td>
</tr>
<tr>
<td>$y_2$</td>
<td>0.0010569</td>
</tr>
<tr>
<td>$y_3$</td>
<td>$-0.00021779$</td>
</tr>
</tbody>
</table>

| $R^2$ value          | 0.957      |
| Root mean square error | 0.0046    |
| Overall $p$-value    | $8.38 \times 10^{-16}$ |

The coefficient of determination i.e $R^2$ value suggests that approximately 95.7% of the variability in the response variable ($\epsilon$) is explained by the model. Values of $R^2$ close to 1 are ideal. The p-value is an indicator of the significance of the term at the 5 % significance level given to other terms. In other words, terms with p-values above 0.05 are not significant to the model and may be ignored. For example, in the case of $\epsilon$, as the p-value of $y_1$ is greater than 0.05, this term is not significant and is ignored. The root mean square error is the root mean
square of the differences between the experimental data values and the values predicted by the model. The lower the root mean square error, the better is the statistical model.

With an overall $p - value = 8.38 \times 10^{-16}$ (should be below 0.05 to be significant) and a root mean square error of 0.0046, it suggests that the above model is sufficient to evaluate $\varepsilon$. The final coefficients are then determined by fitting the experimental data with the significant terms.

Hence, in the case of $\varepsilon$, Equation 20 becomes;

$$\varepsilon = 0.80576 + 0.0010572 \cdot T_{\text{evap}_{\text{in}}} - 0.0002173 \cdot \dot{V}_F$$  \hspace{1cm} Equation 21

Root mean square error of $\varepsilon = 0.00474$

$P_{\text{flux}}, \text{STEC}$ and $\text{GOR}$ were modelled by fitting their experimental data to a quadratic regression equation given by;

$$y = y_0 + y_1 \cdot T_{\text{cond}_{\text{in}}} + y_2 \cdot T_{\text{evap}_{\text{in}}} + y_3 \cdot \dot{V}_F + y_4 \cdot T_{\text{cond}_{\text{in}}} \cdot T_{\text{evap}_{\text{in}}}$$
$$+ y_5 \cdot T_{\text{cond}_{\text{in}}} \cdot \dot{V}_F + y_6 \cdot T_{\text{evap}_{\text{in}}} \cdot \dot{V}_F + y_7 \cdot T_{\text{cond}_{\text{in}}}^2$$
$$+ y_8 \cdot T_{\text{evap}_{\text{in}}}^2 + y_9 \cdot \dot{V}_F^2$$  \hspace{1cm} Equation 22

The results of the regression analysis can be seen in Table 3;

<table>
<thead>
<tr>
<th>Response Coefficient</th>
<th>$P_{\text{flux}}$</th>
<th>STEC</th>
<th>GOR</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Estimate</td>
<td>p-value</td>
<td>Estimate</td>
</tr>
<tr>
<td>$y_0$</td>
<td>1.3556</td>
<td>0.49434</td>
<td>881.22</td>
</tr>
<tr>
<td>$y_1$</td>
<td>-0.091816</td>
<td>0.20262</td>
<td>0.78161</td>
</tr>
<tr>
<td>$y_2$</td>
<td>0.012884</td>
<td>0.7793</td>
<td>-20.105</td>
</tr>
<tr>
<td>$y_3$</td>
<td>-0.012862</td>
<td>0.00012838</td>
<td>0.90971</td>
</tr>
<tr>
<td>$y_4$</td>
<td>0.00094343</td>
<td>0.048688</td>
<td>-0.079829</td>
</tr>
<tr>
<td>$y_5$</td>
<td>-6.6761</td>
<td>$10^{-5}$</td>
<td>0.15066</td>
</tr>
<tr>
<td>$y_6$</td>
<td>0.00031488</td>
<td>5.7305 $10^{-13}$</td>
<td>-0.010932</td>
</tr>
<tr>
<td>$y_7$</td>
<td>0.00046585</td>
<td>0.71224</td>
<td>0.024264</td>
</tr>
<tr>
<td>$y_8$</td>
<td>-0.00039999</td>
<td>0.21543</td>
<td>0.15954</td>
</tr>
<tr>
<td>$y_9$</td>
<td>1.0005 $10^{-6}$</td>
<td>0.74618</td>
<td>2.5926 $10^{-5}$</td>
</tr>
</tbody>
</table>

$R^2$ value: 0.996  0.96  0.961
Root mean square error: 0.0754  8.88  0.106
Overall $p - value$: $2.14 \times 10^{-18}$  $4.19 \times 10^{-10}$  $2.76 \times 10^{-10}$
From Table 3 and Equation 22, the $P_{flux}$, STEC and GOR can be modelled as;

$$P_{flux} = -0.010182 \cdot \dot{V}_F - 0.00015098 \cdot T_{condin} \cdot T_{evapin} + 0.00026597 \cdot T_{evapin} \cdot \dot{V}_F$$  \hspace{1cm} \text{Equation 23}

Root mean square error of $P_{flux} = 0.106 \frac{l}{h \cdot m^2}$

$$STEC = 905.18 - 21.813 \cdot T_{evapin} + 0.99132 \cdot \dot{V}_F - 0.010969 \cdot T_{evapin} \cdot \dot{V}_F + 0.1575 \cdot T_{evapin}^2$$  \hspace{1cm} \text{Equation 24}

Root mean square error of STEC = 15.1 $\frac{kWh}{m^3}$

$$GOR = 0.08827 \cdot T_{evapin} - 0.0092547 \cdot \dot{V}_F + 0.0005928 \cdot T_{condin} \cdot T_{evapin} + 9.1 \times 10^{-5} \cdot T_{evapin} \cdot \dot{V}_F - 0.0006754 \cdot T_{evapin}^2$$  \hspace{1cm} \text{Equation 25}

Root mean square error of GOR = 0.115

For a fixed set of input conditions ($T_{condin}$, $T_{evapin}$ and $\dot{V}_F$) $\varepsilon$ can be calculated using Equation 21 and then the heat transfer through HX2 ($Q_{HX2}$) using Equation 17. In this way the MD loop can be modelled from an energetic scope. To determine the evaporator channel outlet temperature, given only the $T_{condin}$, $T_{evapin}$ and $\dot{V}_F$, it is necessary to redefine $\varepsilon$ as [20];

$$\varepsilon = \frac{T_{evapin} - T_{evapout}}{T_{evapin} - T_{condin}}$$  \hspace{1cm} \text{Equation 26}

Where $T_{evapout}$ is the outlet of the evaporator channel [$^\circ$C]

It is preferred to define $\varepsilon$ in terms of the heat transfer/enthalpy in the condenser channel/cold stream side, as in Equation 7, rather than in terms of the evaporator channel/hot stream side, as in Equation 26 above. This is because the mass flow rate and salinity of the fluid stream remain constant along the length of the condenser channel but, due to evaporation, the same does not hold true in the evaporator channel. However, only for the purposes of determining the $T_{evapout}$ Equation 26 was utilised which when rearranged in terms of $T_{evapout}$ gives;

$$T_{evapout} = T_{evapin} - \varepsilon \cdot (T_{evapin} - T_{condin})$$  \hspace{1cm} \text{Equation 27}
2.2.2 Limitations of mathematical model

The tank loop (see Figure 5) has not been mathematically modelled and hence the developed model only describes the interaction between the solar loop and the MD loop without an intermediary buffer/storage system. This was done for the following reasons;

- The tank loop, as explained earlier, acts chiefly as a buffer between the energy source (solar collector field) and the energy sink (MD module). Its main role is to absorb the fluctuations in incoming solar irradiation and ensure that the MD operation is not disturbed. Thus for the sake of simplicity in a steady state analysis, it has been overlooked and the solar loop is assumed to be directly coupled to the MD loop.

- The time constraints of this study did not allow for a detailed analysis and characterisation of the tank and the tank loop.

Furthermore, the model was developed for a system operating in a steady state and hence cannot be directly applied to an unsteady/transient system. In practical application, especially considering a MD system coupled to a solar collector field, variations in environmental parameters (such as solar irradiation, ambient temperature, wind velocity etc) may cause unsteady operating conditions which may hamper the performance of the system. This topic is briefly discussed in the following chapter with the operation of the auxiliary cooling system in the solar field, in order to achieve steady state operation, when there is variable solar irradiation.

An optimisation analysis was not carried out in this study. A dual optimisation of the $P_{flux}$ and $STEC/GOR$, as carried out in [8], is of particular interest for MD operation as the highest distillate production operation point and lowest energy consumption operation point do not coincide and in most cases there is a trade-off between distillate production and energy consumption [3], [8], [9], [18].

Finally, as the modelling of the PGMD module was carried out based on experimental results from operation of the module within some set of operating conditions, hence the model itself is only valid within this experiment matrix. The manufacturer's specifications of the module allow for operation well beyond the conditions specified in Chapter 2.1.1 especially with regards to the $\dot{V}_F$ and $T_{evap_{in}}$. This was the main reason why a multi-response optimisation was not performed, as such an optimisation would not be reflected of the entire operational range of the PGMD module.
3. Results and Discussion

3.1 Characterisation and performance evaluation of PGMD module

The performance indicators ($\text{STEC}$, $\text{GOR}$, $P_{\text{flux}}$ and $\epsilon$) at various operation points or alternatively the results of the experiment runs are presented in Table 4 below. Secondary results, from experiment runs conducted at $\dot{V}_F = 500 \ l/h$, are shown in Table 5.

Table 4: Primary Experimental Results

<table>
<thead>
<tr>
<th>$T_{\text{cond}}$</th>
<th>$T_{\text{evap}}$</th>
<th>$\dot{V}_F$</th>
<th>$\text{STEC}$</th>
<th>$\Delta\text{STEC}$</th>
<th>$\text{GOR}$</th>
<th>$\Delta\text{GOR}$</th>
<th>$P_{\text{flux}}$</th>
<th>$\Delta P_{\text{flux}}$</th>
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<td>°C</td>
<td>l/h</td>
<td>kWh/m³</td>
<td>kWh/m³</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
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Table 5: Secondary Experimental Results

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<th>$T_{\text{evap}}$ $^\circ\text{C}$</th>
<th>$\dot{V}_F$ l/h</th>
<th>$\text{STEC}$ kWh/m$^3$</th>
<th>$\Delta\text{STEC}$ kWh/m$^3$</th>
<th>$\text{GOR}$</th>
<th>$\Delta\text{GOR}$</th>
<th>$P_{\text{flux}}$ l/(m$^2$ h)</th>
<th>$\Delta P_{\text{flux}}$ l/(m$^2$ h)</th>
<th>$\varepsilon$</th>
<th>$\Delta\varepsilon$</th>
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<td>0.760</td>
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3.1.1 Variation with feed flow rate ($\dot{V}_F$)

The variation of the three main performance indicators $P_{\text{flux}}, \text{STEC}$ and $\text{GOR}$ with $\dot{V}_F$, at different $T_{\text{evap}}$ keeping $T_{\text{cond}} = 25$ $^\circ\text{C}$ constant, is shown in Figure 7, Figure 8 and Figure 9 respectively.

![Graph showing variation of distillate flux with feed flow rate at different evaporation temperatures](image)

*Figure 7: Distillate flux vs feed flow rate at condenser inlet temperature 25 $^\circ\text{C}$*
As can be observed from Figure 7, as the \( \dot{V}_F \) increases \( P_{flux} \) increases as well. This can be attributed to an increase in the driving force (vapour pressure difference across the membrane) brought on by a decreased thermal boundary layer on the membrane surface in the evaporator channel as well as a better heat transfer within this limiting boundary layer due to the higher flow rate. In other words the temperature polarisation (the temperature difference between the membrane surface and the bulk fluid temperature) is reduced due to the higher \( \dot{V}_F \). The reduction in the temperature polarisation leads to an increase in the transmembrane temperature difference, which in turn increases the vapour pressure difference/driving force. The effectiveness of internal heat exchange (\( \varepsilon \)) decreases with an increase in \( \dot{V}_F \) (see Table 4 and Equation 21) and hence the permeate side of the membrane remains at a lower temperature.
which further increases the temperature differential and driving force. Thus the $P_{\text{flux}}$ is positively related to $\dot{V}_F$.

It is inherently apparent that the heat required to lift a fluid’s temperature between two fixed temperature limits must increase with an increase in its thermal capacitance. The thermal capacitance, in this case, increasing chiefly due to an increase in $\dot{V}_F$. Referring to Equation 16, with other variables constant, the $Q_{HX2}$ must increase to match an increase in the $\dot{m}_F$. Besides this, a falling $\varepsilon$ with rising $\dot{V}_F$ implies that the condenser outlet temperature ($T_{\text{cond,out}}$) decreases and as can be inferred from Equation 16, this increases the $Q_{HX2}$. As can be seen from Figure 8 and Figure 9, this increase is large enough to offset the increase in the distillate production and hence with higher $\dot{V}_F$, the $\text{STEC}$ rises and $\text{GOR}$ falls.

Alternatively it can be stated that with a larger $\dot{V}_F$, the distillate production rate increases but each unit of distillate produced requires more thermal energy than at a lower $\dot{V}_F$.

Employing the mathematical model, the increase in driving force (represented by an increase in the average temperature difference between the evaporator and condenser channels) leads to an increase in distillate production with rising $\dot{V}_F$ as can be seen in Figure 10. However the rise in the heat required $Q_{HX2}$ (see Figure 11) far outpaces this increase as can be observed by comparing the magnitude of the difference between the final and initial y-axis values in Figure 10 and Figure 11. Hence an overall rise in the $\text{STEC}$, and a fall in the $\text{GOR}$, can be predicted with increasing $\dot{V}_F$.

Figure 10: Temperature differential driving force vs feed flow rate
The model here utilised a $T_{\text{evap in}} = 70 \, ^{\circ}\text{C}$ and $T_{\text{cond in}} = 25 \, ^{\circ}\text{C}$ to generate Figure 10 and Figure 11.

Additionally it can be observed that at higher $T_{\text{evap in}}$ there is a better performance both in terms of distillate production rate ($P_{\text{flux}}$) as well as energy efficiency ($STEC$ and $GOR$). This effect of the evaporator inlet temperature will be dealt with in more depth in the following section.
### 3.1.2 Variation with evaporator inlet temperature ($T_{\text{evap}_{\text{in}}}$)

The variation of the three main performance indicators $P_{\text{flux}}$, STEC and GOR with $T_{\text{evap}_{\text{in}}}$, at different $T_{\text{cond}_{\text{in}}}$ keeping $\dot{V}_F = 200 \frac{l}{h}$ constant, is shown in Figure 12, Figure 13 and Figure 14 respectively.

![Figure 12: Distillate flux vs evaporator inlet temperature at feed flow rate of 200 l/h](image1.png)

![Figure 13: STEC vs evaporator inlet temperature at feed flow rate of 200 l/h](image2.png)
A rise in the \( T_{\text{evap,in}} \), keeping the \( T_{\text{cond,in}} \) constant, increases the transmembrane temperature difference which directly increases the vapour pressure difference and hence the driving force behind the distillation. This behaviour is reflected in Figure 12 where the \( P_{\text{flux}} \) continuously increases with increasing \( T_{\text{evap,in}} \).

However to attain these higher \( T_{\text{evap,in}} \), a larger supply of heat is required (other variables remaining the same) and hence the \( Q_{\text{HX2}} \) also rises with a rise in \( T_{\text{evap,in}} \). This can be easily understood by referring to Equation 16.

Unlike in the case with \( V_F \), the increase in the distillate flow rate is large enough to offset the increase in the thermal energy consumption of the system and hence the \( STEC \) falls and the \( GOR \) rises continuously with an increasing \( T_{\text{evap,in}} \) as can been observed in Figure 13 and Figure 14.

Thus operating at a higher \( T_{\text{evap,in}} \) implies both a higher distillate production rate as well as a lower thermal energy consumption per unit of distillate produced. This double benefit can be viewed in terms of the variation in the \( \epsilon \). From the Table 4 and Equation 21, it can be seen that there is a small positive correlation between \( T_{\text{evap,in}} \) and \( \epsilon \) which means that the average condenser temperature should slightly increase but as this increase is much lower than the increase in the average evaporator temperature, the transmembrane temperature differential increases and hence the distillate production \( P_{\text{flux}} \) increases. The heat required \( Q_{\text{HX2}} \) still increases though, despite the increase in \( \epsilon \), as the magnitude of increase of \( T_{\text{evap,in}} \) is not matched by a similar increase in \( T_{\text{cond,out}} \).

This behaviour can also be explained using the developed mathematical model. With the rising \( T_{\text{evap,in}} \) and \( \epsilon \), the \( T_{\text{cond,out}} \) also increases (see Figure 15) but cannot match the increase in \( T_{\text{evap,in}} \) (see Figure 16).
The driving force hence also increases and the same can be represented by plotting the average temperature difference that exists between the evaporator and condenser channels as can be seen in Figure 17. It should be noted that the model used $T_{\text{cond}_{\text{in}}} = 20 \, ^{\circ}\text{C}$ and $\dot{V}_F = 200 \, \frac{l}{h}$ when generating Figure 15, Figure 16 and Figure 17.
3.1.3 Variation with condenser inlet temperature ($T_{cond_{in}}$)

The variation of the three main performance indicators $P_{flux}$, $STEC$ and $GOR$ with $T_{cond_{in}}$, at different $V_F$, keeping $T_{evap_{in}} = 70$ °C constant, is shown in Figure 18, Figure 19 and Figure 20 respectively.

![Figure 18](image1.png)

*Figure 18: Distillate flux vs condenser inlet temperature at evaporator inlet temperature of 70°C*

![Figure 19](image2.png)

*Figure 19: STEC vs condenser inlet temperature at evaporator inlet temperature of 70°C*
The driving force continuously decreases with an increase in the $T_{\text{cond in}}$ as the temperature difference in the evaporator and condenser channel narrows. Thus, the distillate production and flux $P_{\text{flux}}$ fall in Figure 18 although the decrease is relatively small. The heat required to drive the MD process must also decline as the $T_{\text{cond in}}$, and hence the $T_{\text{cond out}}$, draw closer to the $T_{\text{evap in}}$. It can be observed in Figure 19 and Figure 20 that the energy required to produce a unit volume of distillate continuously decreases with increasing $T_{\text{cond in}}$. Hence the effect of decreasing heat required is consistently larger than the effect of decreasing distillate production at higher $T_{\text{cond in}}$. Therefore, it can be summarised that the distillate production marginally decreases with increasing $T_{\text{cond in}}$ but the energy cost for producing a unit of distillate also falls.

Note that, from Table 4 and Equation 21, the effectiveness $\varepsilon$ is virtually independent of $T_{\text{cond in}}$. The mathematical model developed confirms the above findings with the driving force, and thus distillate production, along with the heat required predicted to fall (see Figure 21 and Figure 22)
The model assumed a $T_{evap_{in}} = 70 \, ^\circ C$ and $\dot{V}_F = 300 \frac{l}{h}$ while generating Figure 21 and Figure 22.

### 3.1.4 Repetition of experiment run

To check the accuracy of the experiments vis-à-vis experiment errors, variation of performance parameters and system stability over time, a random experiment run was selected and performed 3 times on different days. The experiment run chosen was;

$$T_{evap_{in}} = 70 \, ^\circ C, T_{cond_{in}} = 20 \, ^\circ C \text{ and } \dot{V}_F = 300 \frac{l}{h}$$

The results of the three runs are shown in Table 6 below. As can be seen, there were fairly small deviations between experiment runs given the magnitude of the respective errors in final values due to propagation of errors.

Table 6: Results from experiment run repetition

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<th>$\Delta$STEC</th>
<th>GOR</th>
<th>$\Delta$GOR</th>
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<th>$\Delta P_{flux}$</th>
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<td>kWh/m$^3$</td>
<td>-</td>
<td>-</td>
<td>l/(h·m$^2$)</td>
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3.1.5 Validation and comparison of experimental results

The results from the experiment campaign on the PGMD module yielded results that match well with expected trends and conclusions from previous studies and experimental works using the same type and construction of MD module i.e spiral wound PGMD [3], [8]–[10]. While the size of the module was slightly different from the ones used in other works, the performance indicators were within the expected range.

Experimental results were compared with findings from [8] and are presented in Figure 23, Figure 24 and Figure 25.

![Figure 23: Comparison of STEC between current study and [8]](image)

![Figure 24: Comparison of distillate fluxes between current study and [8]](image)
As can be seen from Figure 24, the $P_{flux}$ of the current PGMD module is consistently and considerably higher than the PGMD module tested in [8]. The $STE_C$ and $GOR$ of both modules are almost on par and no conclusive inference can be drawn from Figure 23 and Figure 25 regarding the relative energy performance of one module over the other.

In [10], a spiral wound PGMD module with a membrane area of 10 m$^2$ was found to have $P_{flux} = 1.33 \frac{l}{h \cdot m^2}$ and $STE_C = 142 \frac{kWh}{m^3}$ at $T_{cond_{in}} = 25$ °C, $T_{evap_{in}} = 80$ °C and $\dot{m}_F = 300 \frac{kg}{h}$ ($\equiv \dot{V}_F = 300 \frac{l}{h}$). By comparison, the current module, at the same operating conditions, had $P_{flux} = 2.904 \frac{l}{h \cdot m^2}$ and $STE_C = 208 \frac{kWh}{m^3}$.

Hence the distillate production rate of the current spiral wound PGMD module is significantly higher than that of earlier modules while its energy consumption is comparable. The improvement in the distillate production rate can be attributed to an increased driving force. The driving force may be increased due to a reduction in the temperature polarisation in the PGMD module channels which in turn might be due to changes in the design and construction of the spacers placed in the flow channels to increase the turbulence and hence reduce the temperature difference between the fluid bulk and the membrane surface i.e the temperature polarisation. Differences in the channel widths lead to different flow velocities, at the same volumetric flow rate, and this may also lead to changes in the driving force. Of course, the differences in the membranes themselves, each having their own permeability, tortuosity and porosity, cause variations in the mass transfer (as well as heat transfer) resistance which directly impacts the driving force. A more thorough analysis of the individual spiral wound PGMD modules needs to be carried out to reveal the exact cause(s) of this increased driving force however this is beyond the scope of the current work.
3.2 Analysis with Mathematical Model

3.2.1 Validation of mathematical model of solar loop

To validate the model of the solar collector field, experimental runs were made to test the accuracy of the theoretical predictions with experimental results. The overall efficiency of the solar collector field was estimated using Equation 10 and compared with experimental findings using Equation 12. An uncertainty analysis of the same can be found in Appendix D.

Figure 26 below shows the comparison of the theoretically and experimentally determined efficiency of solar collection during an hour’s long steady state experiment. The experiment was carried out on 02/02/2018 between 14:00-15:00 local time (no daylights savings). As the data is plotted for every second of the hour the individual data points are impossible to distinguish and this holds true for the error bars as well.

![Figure 26: Theoretical and experimental solar collector efficiency](image)

The average relative error was found to be 4 %. This was deemed to be within the acceptable limit of accuracy, given the propagation of error and uncertainties in the measurements themselves, and hence the mathematical modelling of the solar collector field is experimentally validated. The differences in the efficiencies arise because of the approximation of the total collector area as one large collector instead of treating it as 4 different collectors connected in series. Additionally, there is a capacitance effect where the collector outlet temperature does not change quickly with changes to the collector inlet temperature which continuously rises due to the charging tank. As the tank heats up, the temperature at the inlet to the heat exchanger from the tank increases and hence the solar collector inlet temperature also increases. However, the solar collector outlet temperature doesn’t rise immediately due to the thermal capacitance and this raises the average collector temperature. This capacitance effect provides another source of error besides ambient conditions such as wind, humidity etc which are ignored in the theoretical model.
3.2.2 Validation of MD module and loop

To validate the MD module and loop operation using the model, it is tested against the experimental data from the experiment runs. The following experiment runs were randomly chosen to test the accuracy of the model:

\[ T_{\text{evap}_{in}} = 70 \, ^\circ\text{C}, \, T_{\text{cond}_{in}} = 25 \, ^\circ\text{C}, \, \dot{V}_F = 200, 300, 400 \text{ and } 500 \, \frac{l}{h} \]

The parameter evaluated theoretically and experimentally is the heat required \( Q_{HX2} \) and the results from the trials can be seen in Figure 27.

![Figure 27: Theoretical and experimental heat required](image)

The average relative error was 4%. Given the error propagation and uncertainties in measurements (see Appendix D for a more detailed uncertainty analysis), this deviation falls within the acceptable range of error. Thus the mathematical model of the MD module can be said to be experimentally validated.
3.2.3 Minimum radiation required for MD operation

Using the mathematical model and applying some simplifying assumptions, it is possible to derive the minimum global solar radiation (on the titled plane/collector surface) required to operate the MD module at different conditions. The exact mathematical manipulations and assumptions made are discussed in detail in Appendix F. As can be observed in Table 7, the behaviour of minimum solar radiation required at various operating conditions is the same as that of heat required for MD operation $Q_{HX2}$.

Table 7: Minimum solar radiation required at different MD operating conditions

<table>
<thead>
<tr>
<th>$T_{\text{cond}_{in}}$</th>
<th>$T_{\text{evap}_{in}}$</th>
<th>$\dot{V}_F$</th>
<th>$G_T$</th>
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<td>l/h</td>
<td>W/m²</td>
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3.2.4 Auxiliary cooling required in solar loop for steady state operation

In practice it is often desired to operate in steady state conditions and not in fluctuating conditions brought on by variations in incoming solar irradiation. Given that, within the scope of this study, there is only a single heat source, the solar collector field, hence there is no possibility of additional heating if the irradiation is insufficient. However, cooling may be carried out in the case of excessive irradiation. To this end, the mathematical model is applied to determine the auxiliary cooling necessary to maintain a near-constant heat exchanger (on the solar side) input at higher than required solar irradiations for a given set of operating conditions (see Table 7). A detailed explanation on the mathematical manipulations as well as simplifying assumptions made can be found in Appendix F.

Table 8: Auxiliary cooling required for steady state operation at 700 W/m² total irradiation on slope

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<tr>
<th>$T_{\text{cond,in}}$</th>
<th>$T_{\text{evap,in}}$</th>
<th>$\dot{V}_F$</th>
<th>$Q_{\text{cooling}}$</th>
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4. Further Discussion

4.1 Operation with solar thermal collector field

This section will delve into general principles and guidelines to improved operation of a MD system in conjunction with a solar collector system. The following is based on operational experience as well as previous studies [5], [7], [11], [12], [17];

- First the desired steady state operation point for the MD module must be decided. It may be based on optimising $P_{flux}$, $STEC/GOR$ or both. This fixes the $T_{evap_{in}}$, $T_{cond_{in}}$ and $\dot{V}_F$.

- Matched flow on either side of a heat exchanger provides the best heat transfer rate and hence the flow rate in the solar collector loop should aim to match $\dot{V}_F$.

Depending on whether the overall system’s objective is for maximum daily production (operation spread over a longer time period per day) or simply for steady state operation during permitting sunshine hours, a storage tank or inertia tank may be used respectively. Each case must be treated separately because of their different objectives.

4.1.1 System with inertia tank

- The solar collector outlet temperature may then be fixed to a few degrees above the $T_{evap_{in}}$. This is because solar collector efficiency declines at higher operating temperatures as can be seen in Figure 28 below. While Figure 28 has been drawn using Equation 10 and the characteristics for the flat plate collector used in this study, the negative relation between mean collector temperature and collector efficiency holds true for all collectors. Besides for overcoming environmental heat losses, there is no reason to operate the solar collector at temperatures much higher than $T_{evap_{in}}$. 
Figure 28: Solar collector efficiency vs average collector temperature at 25°C ambient

- In order to limit the solar collector operating temperature, especially when the irradiation is higher than required, an auxiliary cooler must be employed. Of course, as discussed in Chapter 3.2.4, the cooling required is a function of the incident irradiation and is hence variable in nature. Some active variable control, such as a PID controller, must thus be implemented to deliver the necessary degree of cooling.

4.1.2 System with storage tank

- The solar collector temperature in this case does not need to be restricted and it may be allowed to rise, within limits of its operation, to charge the storage tank to a higher temperature than required for MD operation. High temperature tank charging allows for operation of the MD unit even after the irradiation levels fall below the minimum required.

- Of course during the course of tank charging the tank’s temperature will rise and so will its outlet temperature to the heat exchanger connected to the MD loop. In order to regulate this temperature, the return from the heat exchanger needs to be suitably mixed with the tank’s outlet to the MD loop. An active tempering valve needs to be utilised for this purpose and its setting needs to be actively controlled, potentially by PID control, to obtain the desired temperature to the heat exchanger with the MD loop.
4.2 Future areas of interest and investigation

With respect to the current specific PGMD module used in this study, a complete characterisation of the module, within its operational limits, would reveal its maximum $P_{\text{flux}}$, $\text{GOR}$ and minimum $\text{STEC}$. The module has the capacity to operate at $\dot{V}_F = 700 \frac{\text{l}}{\text{h}}$ which is far above the highest level used in this study. Moreover, the upper limit of $T_{\text{cond}_{\text{in}}}$ could be raised to analyse the performance of the module at elevated condenser temperatures which may be the case in regions (especially in the Arabian gulf) with higher seawater temperatures. The PGMD module’s $T_{\text{evap}_{\text{in}}}$ upper operational limit, provided by the manufacturer, was 90 °C which is not much higher than the 80 °C used in this investigation however is still an area for further study. Spiral wound modules are especially sensitive to the $T_{\text{evap}_{\text{in}}}$ because at elevated temperatures the seals and adhesives, used in its construction, are degraded and eventually fail causing leakages. It is for this reason that a 10 °C margin of safety was used. A full characterisation of the PGMD module would allow for an optimisation analysis into the optimum operation conditions for maximum distillate production and minimum energy consumption. Lastly, an investigation into the effects of inlet feed water salinity should be carried out to test the variation in performance at different concentration levels. This, along with an investigation with increased $T_{\text{cond}_{\text{in}}}$, would give an idea about the suitability for batch operation. Batch operation here refers to the mixing of the inlet feedwater with some proportion of the outgoing brine.

More generally, a comparative study should be made of different control strategies and configurations of MD systems coupled to solar thermal collectors vis-à-vis solar fraction, distillate production etc. A detailed investigation could also be carried out into the optimal placement of nodes on a storage tank and inertia tank which is important given the low temperature drop (around 8 °C) observed across the heat exchanger with the MD loop. To preserve the stratification of the tank, care must be taken to place the returning node at the appropriate height.

Electrical consumption is often ignored as it is much smaller in magnitude than the heat consumption. However this facet must not be completely neglected and future studies must consider it as well as it’s especially useful for off-grid stand alone system design and application [5], [6] which requires some source of electrical energy as well.

With respect to the mathematical model developed, it would be interesting to see the utility and insightfulness of employing the effectiveness of internal heat recovery as a means of modelling the MD module. As it was observed in this study, a linear regression analysis of the effectiveness provided a simple yet accurate tool to model the PGMD unit at different operating conditions. A more detailed mathematical model could be developed and would need to consider the storage/inertia tank operation as well as more thoroughly analyse the system’s heat losses to the environment and the heat transfer process through the heat exchangers. Besides this, the current model developed simply considered the PGMD module from a thermal energy consumption standpoint but a more complete model would have to consider the distillate production as well.
5. Conclusions

A complete characterisation of a precommercial spiral wound Permeate Gap Membrane Distillation (PGMD) module, for the purpose of seawater (3.5 %) desalination, has been carried out to determine the performance of the PGMD module at different operating conditions. The performance indicators utilised in this study were the distillate flux ($P_{\text{flux}}$), specific thermal energy consumption ($STE\text{C}$) and the gross output ratio ($GOR$). The operating variables varied were the evaporator inlet temperature ($T_{\text{evap}_i}$), condenser inlet temperature ($T_{\text{cond}_i}$) and the feed flow rate ($\dot{V}_F$). These three operating variables were set at the following levels;

\[
T_{\text{evap}_i} = 60 \, ^\circ C, 70 \, ^\circ C, 80 \, ^\circ C
\]

\[
T_{\text{cond}_i} = 20 \, ^\circ C, 25 \, ^\circ C, 30 \, ^\circ C
\]

\[
\dot{V}_F = 200 \, \frac{l}{h}, 300 \, \frac{l}{h}, 400 \, \frac{l}{h}
\]

Within this experiment matrix, the highest distillate flux $P_{\text{flux}} = 4.135 \, \frac{l}{h,m^2}$ was recorded at $T_{\text{evap}_i} = 80 \, ^\circ C, T_{\text{cond}_i} = 20 \, ^\circ C, \dot{V}_F = 400 \, \frac{l}{h}$. This equates to a distillate production rate of $21.3 \, \frac{l}{h}$. The lowest specific thermal consumption $STE\text{C} = 180 \, \frac{kWh}{m^3}$ and highest gross output ratio $GOR = 3.7$ simultaneously occurred at $T_{\text{evap}_i} = 80 \, ^\circ C, T_{\text{cond}_i} = 30 \, ^\circ C, \dot{V}_F = 200 \, \frac{l}{h}$.

These results were found to be consistent with previous studies and indicate a considerable improvement in the distillate production rate over previous PGMD modules of the spiral wound construction. There was no noticeable change in the specific thermal energy consumption in the studied PGMD module over previously studied modules.

Additionally, three experiment runs were made at a higher feed flow rate. The three experiment runs were at operating conditions;

1. \(T_{\text{evap}_i} = 70 \, ^\circ C, T_{\text{cond}_i} = 20 \, ^\circ C, \dot{V}_F = 500 \, \frac{l}{h}\)

2. \(T_{\text{evap}_i} = 60 \, ^\circ C, T_{\text{cond}_i} = 25 \, ^\circ C, \dot{V}_F = 500 \, \frac{l}{h}\)

3. \(T_{\text{evap}_i} = 70 \, ^\circ C, T_{\text{cond}_i} = 25 \, ^\circ C, \dot{V}_F = 500 \, \frac{l}{h}\)

Maximum distillate production rate of the PGMD module is expected to be higher as, according to the manufacturer’s specifications, it can operate at a maximum feed flow rate of $700 \, \frac{l}{h}$ and a maximum evaporator inlet temperature of $90 \, ^\circ C$. An increase in either of these operating parameters would lead to an increase in the distillate production rate.

A simplified generic mathematical model of a flat plate solar collector field coupled with a MD module was developed, and experimentally validated, to further analyse the overall system. The model was used to derive the minimum global irradiation (on the collector plane) required to operate the MD module at steady state as well as auxiliary cooling required in the solar collector.
field to regulate the operating temperature of the MD operation and ensure steady state conditions of operation.

Modelling of the spiral wound PGMD module utilised a novel method of evaluation namely through the effectiveness of internal heat exchange between the evaporator and condenser channels. This parameter, which was found to vary linearly with the evaporator inlet temperature and feed flow rate, could be used to accurately model the PGMD module (in temperature and energy terms).

Finally, general guidelines and principles toward optimum usage of a solar thermal collector system in conjunction with a MD system, using inertia or storage tanks, were presented and discussed.


6. References

Book

Book chapter

Journal Article


Conference Paper


Report


Web document


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7. Appendices

7.1 Appendix A

7.1.1 Technical details of experimental setup and its components

1. Solar collector field

The solar collector field consisted of four LBM 10 HTF manufactured by Wagner and Co. based in Madrid, Spain. Below is attached some technical data on the flat plate collector and a picture of the collector field itself.
2. Permeate Gap Membrane Distillation Module

The technical details of the permeate gap distillation module, as provided by the manufacturer, are:

Membrane Contactor module LS6.707.4 (stainless steel distillate spacer)

Configuration: LGMD (liquid gap MD)

Membrane area: 5.15 m²

Channel length: 6.3 m; Distillate channel spacer: stainless steel 316 mesh, 0.35 mm thickness

Channel widths 415 mm; Feed spacer: 2.85 mm; Membrane channel spacer: 2.85 mm

3. Tank loop

Tank volume = 1.5 m³

Direct no heat exchanger type tank.

Working fluid: Water at 2 bar pressure

Number of nodes = 4

7.1.2 Images/Pictures of experimental setup

Figure 29: Solar collector field (tank in distance)
Figure 30: Solar and tank loops
## 7.2 Appendix B

Table 9: Experiment input parameters with dispersion and standard deviations for all experiment runs

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<th>$T_{\text{evap,in}}$ (°C)</th>
<th>$V_F$ (l/h)</th>
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### 7.3 Appendix C

**Table 10: Date and time of experiment runs**

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7.4 Appendix D

7.4.1 Derivation of uncertainty/error of performance indicators

Equation 9 has been applied to the respective equations (Equation 2 for $P_{flux}$, Equation 4 for STEC, Equation 6 for GOR and Equation 7 for $\varepsilon$) to determine the final reduced uncertainty equation for each of the above performance indicator.

a) $P_{flux}$

Recalling the expression for $P_{flux}$;

$$P_{flux} = \frac{\dot{m}_p}{\rho_p \cdot A_M} \quad \text{Equation 2}$$

Where $\dot{m}_p$ and $\rho_p$ are measured and derived variables respectively and $A_M$ is constant. Applying Equation 9 to the Equation 2;

$$\Delta P_{flux} = \sqrt{\left(\frac{\partial P_{flux}}{\partial \dot{m}_p} \cdot \Delta \dot{m}_p\right)^2 + \left(\frac{\partial P_{flux}}{\partial \rho_p} \cdot \Delta \rho_p\right)^2}$$

$$= \sqrt{\left(\frac{1}{\rho_p \cdot A_M} \cdot \Delta \dot{m}_p\right)^2 + \left(-\frac{\dot{m}_p}{\rho_p^2 \cdot A_M} \cdot \Delta \rho_p\right)^2}$$

$$= \frac{1}{A_M} \cdot \sqrt{\left(\Delta \dot{m}_p\right)^2 + \left(-\frac{\dot{m}_p \cdot \Delta \rho_p}{\rho_p^2}\right)^2}$$

b) STEC

Recalling the expression for STEC;

$$\text{STEC} = \frac{\dot{V}_F \cdot \rho_F \cdot C_{PF} \cdot \rho_p \cdot (T_{evap_in} - T_{cond_out})}{\dot{m}_p} \quad \text{Equation 4}$$

Where $\dot{V}_F$, $T_{evap_in}$, $T_{cond_out}$ and $\dot{m}_p$ are measured variables and $\rho_F$, $C_{PF}$ and $\rho_p$ are derived variables. Applying Equation 9 to the Equation 4;

$$\Delta \text{STEC} = \sqrt{\left(\frac{\partial \text{STEC}}{\partial \dot{V}_F} \cdot \Delta \dot{V}_F\right)^2 + \left(\frac{\partial \text{STEC}}{\partial \rho_F} \cdot \Delta \rho_F\right)^2 + \left(\frac{\partial \text{STEC}}{\partial C_{PF}} \cdot \Delta C_{PF}\right)^2 + \left(\frac{\partial \text{STEC}}{\partial \rho_p} \cdot \Delta \rho_p\right)^2}$$

$$+ \left(\frac{\partial \text{STEC}}{\partial T_{evap_in}} \cdot \Delta T_{evap_in}\right)^2 + \left(\frac{\partial \text{STEC}}{\partial T_{cond_out}} \cdot \Delta T_{cond_out}\right)^2 + \left(\frac{\partial \text{STEC}}{\partial \dot{m}_p} \cdot \Delta \dot{m}_p\right)^2}$$
\[ \Delta GOR = \left[ \left( \frac{\Delta h_v}{V_F \cdot \rho_F \cdot C_{pf} \cdot (T_{evap\text{in}} - T_{cond\text{out}})} \right) \cdot \Delta \dot{m}_p \right]^2 + \left( \frac{\Delta h_v}{V_F \cdot \rho_F \cdot C_{pf} \cdot (T_{evap\text{in}} - T_{cond\text{out}})} \right) \cdot \Delta \dot{V}_F \right] ^2 + \left( \frac{\Delta h_v}{V_F \cdot \rho_F \cdot C_{pf} \cdot (T_{evap\text{in}} - T_{cond\text{out}})} \right) \cdot \Delta \rho_F^2 \]

\[ + \left( \frac{\dot{m}_p \cdot \Delta h_v}{V_F \cdot \rho_F \cdot C_{pf} \cdot (T_{evap\text{in}} - T_{cond\text{out}})} \right) \cdot \Delta \dot{m}_p \right]^2 + \left( \frac{\dot{m}_p \cdot \Delta h_v}{V_F \cdot \rho_F \cdot C_{pf} \cdot (T_{evap\text{in}} - T_{cond\text{out}})} \right) \cdot \Delta \dot{V}_F \right] ^2 + \left( \frac{\dot{m}_p \cdot \Delta h_v}{V_F \cdot \rho_F \cdot C_{pf} \cdot (T_{evap\text{in}} - T_{cond\text{out}})} \right) \cdot \Delta \rho_F^2 \]

\[ + \left( \frac{\dot{V}_F^2 \cdot \rho_F \cdot C_{pf} \cdot (T_{evap\text{in}} - T_{cond\text{out}})}{V_F \cdot \rho_F \cdot C_{pf} \cdot (T_{evap\text{in}} - T_{cond\text{out}})} \right) \cdot \Delta C_{pf}^2 + \left( \frac{\dot{V}_F^2 \cdot \rho_F \cdot C_{pf} \cdot (T_{evap\text{in}} - T_{cond\text{out}})}{V_F \cdot \rho_F \cdot C_{pf} \cdot (T_{evap\text{in}} - T_{cond\text{out}})} \right) \cdot \Delta \dot{V}_F \right] ^2 + \left( \frac{\dot{V}_F^2 \cdot \rho_F \cdot C_{pf} \cdot (T_{evap\text{in}} - T_{cond\text{out}})}{V_F \cdot \rho_F \cdot C_{pf} \cdot (T_{evap\text{in}} - T_{cond\text{out}})} \right) \cdot \Delta \rho_F^2 \]

\[ + \left( \frac{\dot{m}_p \cdot \Delta h_v}{V_F \cdot \rho_F \cdot C_{pf} \cdot (T_{evap\text{in}} - T_{cond\text{out}})} \right) \cdot \Delta \dot{m}_p \right]^2 + \left( \frac{\dot{m}_p \cdot \Delta h_v}{V_F \cdot \rho_F \cdot C_{pf} \cdot (T_{evap\text{in}} - T_{cond\text{out}})} \right) \cdot \Delta \dot{V}_F \right] ^2 + \left( \frac{\dot{m}_p \cdot \Delta h_v}{V_F \cdot \rho_F \cdot C_{pf} \cdot (T_{evap\text{in}} - T_{cond\text{out}})} \right) \cdot \Delta \rho_F^2 \]

\[ + \left( \frac{\dot{V}_F^2 \cdot \rho_F \cdot C_{pf} \cdot (T_{evap\text{in}} - T_{cond\text{out}})}{V_F \cdot \rho_F \cdot C_{pf} \cdot (T_{evap\text{in}} - T_{cond\text{out}})} \right) \cdot \Delta C_{pf}^2 + \left( \frac{\dot{V}_F^2 \cdot \rho_F \cdot C_{pf} \cdot (T_{evap\text{in}} - T_{cond\text{out}})}{V_F \cdot \rho_F \cdot C_{pf} \cdot (T_{evap\text{in}} - T_{cond\text{out}})} \right) \cdot \Delta \dot{V}_F \right] ^2 + \left( \frac{\dot{V}_F^2 \cdot \rho_F \cdot C_{pf} \cdot (T_{evap\text{in}} - T_{cond\text{out}})}{V_F \cdot \rho_F \cdot C_{pf} \cdot (T_{evap\text{in}} - T_{cond\text{out}})} \right) \cdot \Delta \rho_F^2 \]
d) \( \varepsilon \)

Recalling the expression for \( \varepsilon \);

\[
\varepsilon = \frac{T_{\text{cond\_out}} - T_{\text{cond\_in}}}{T_{\text{evap\_in}} - T_{\text{cond\_in}}}
\]

*Equation 7*

Where \( T_{\text{cond\_out}}, T_{\text{cond\_in}} \) and \( T_{\text{evap\_in}} \) are measured variables. Applying Equation 9 to Equation 7;

\[
\Delta \varepsilon = \sqrt{\left( \frac{\partial \varepsilon}{\partial T_{\text{cond\_out}}} \cdot \Delta T_{\text{cond\_out}} \right)^2 + \left( \frac{\partial \varepsilon}{\partial T_{\text{cond\_in}}} \cdot \Delta T_{\text{cond\_in}} \right)^2 + \left( \frac{\partial \varepsilon}{\partial T_{\text{evap\_in}}} \cdot \Delta T_{\text{evap\_in}} \right)^2}
\]

\[
= \sqrt{\left( \frac{1}{T_{\text{evap\_in}} - T_{\text{cond\_in}}} \cdot \Delta T_{\text{cond\_out}} \right)^2 + \left( \frac{T_{\text{cond\_out}} - T_{\text{evap\_in}}}{(T_{\text{evap\_in}} - T_{\text{cond\_in}})^2} \cdot \Delta T_{\text{cond\_in}} \right)^2}
\]

\[
+ \left( \frac{T_{\text{cond\_out}} - T_{\text{cond\_in}}}{(T_{\text{evap\_in}} - T_{\text{cond\_in}})^2} \cdot \Delta T_{\text{evap\_in}} \right)^2
\]
7.4.2 Derivation of uncertainty/error of solar collector efficiency

Equation 9 has been applied to the experimentally determined solar collector efficiency (given by Equation 12) and the theoretically determined efficiency (given by Equation 10) to calculate the uncertainty/error of the above two parameters.

a) Experimental $\eta_{solar}$

Recalling the expression for $\eta_{solar}$:

$$\eta_{solar\,exp} = \frac{m_{solar} \cdot C_{p\,col} \cdot (T_{solar\,out} - T_{solar\,in})}{A_{solar} \cdot G_T}$$  \hspace{1cm} \textbf{Equation 12}$$

This needs to be rewritten in terms of the measured variables as;

$$\eta_{solar\,exp} = \frac{V_{solar} \cdot \rho_{col} \cdot C_{p\,col} \cdot (T_{solar\,out} - T_{solar\,in})}{A_{solar} \cdot G_T}$$

Where $V_{solar}$ is the volumetric flow rate in the solar loop [l/s]

$\rho_{col}$ is the density of the fluid at mean collector temperature [kg/l]

Here $V_{solar}$, $C_{p\,col}, T_{solar\,out}, T_{solar\,in}$ and $G_T$ are measured variables, $\rho_{col}$ and $C_{p\,col}$ are derived variables and $A_{solar}$ is a constant. Applying Equation 9 to this;

$$\Delta \eta_{solar\,exp} = \sqrt{\left(\frac{\partial \eta_{solar\,exp}}{\partial V_{solar}} \Delta V_{solar}\right)^2 + \left(\frac{\partial \eta_{solar\,exp}}{\partial \rho_{col}} \Delta \rho_{col}\right)^2 + \left(\frac{\partial \eta_{solar\,exp}}{\partial C_{p\,col}} \Delta C_{p\,col}\right)^2 + \left(\frac{\partial \eta_{solar\,exp}}{\partial T_{solar\,out}} \Delta T_{solar\,out}\right)^2 + \left(\frac{\partial \eta_{solar\,exp}}{\partial T_{solar\,in}} \Delta T_{solar\,in}\right)^2 + \left(\frac{\partial \eta_{solar\,exp}}{\partial G_T} \Delta G_T\right)^2}$$

$$= \sqrt{\left(\frac{\rho_{col} \cdot C_{p\,col} \cdot (T_{solar\,out} - T_{solar\,in})}{A_{solar} \cdot G_T} \cdot \Delta V_{solar}\right)^2 + \left(\frac{V_{solar} \cdot \rho_{col} \cdot C_{p\,col} \cdot (T_{solar\,out} - T_{solar\,in})}{A_{solar} \cdot G_T} \cdot \Delta \rho_{col}\right)^2 + \left(\frac{V_{solar} \cdot \rho_{col} \cdot C_{p\,col} \cdot T_{solar\,out}}{A_{solar} \cdot G_T} \cdot \Delta C_{p\,col}\right)^2 + \left(\frac{V_{solar} \cdot \rho_{col} \cdot C_{p\,col} \cdot T_{solar\,in}}{A_{solar} \cdot G_T} \cdot \Delta T_{solar\,out}\right)^2 + \left(\frac{V_{solar} \cdot \rho_{col} \cdot C_{p\,col} \cdot T_{solar\,in}}{A_{solar} \cdot G_T} \cdot \Delta T_{solar\,in}\right)^2 + \left(\frac{V_{solar} \cdot \rho_{col} \cdot C_{p\,col} \cdot T_{solar\,out}}{A_{solar} \cdot G_T^2} \cdot \Delta G_T\right)^2}$$
\[
\frac{1}{A_{solar}} \left( \frac{\rho_{col} \cdot C_{p, col} \cdot (T_{solar, out} - T_{solar, in})}{G_T} \cdot \Delta V_{solar} \right)^2 + \left( \frac{V_{solar} \cdot C_{p, col} \cdot (T_{solar, out} - T_{solar, in})}{G_T} \cdot \Delta \rho_{col} \right)^2 \\
+ \left( \frac{\dot{V}_{solar} \cdot \rho_{col} \cdot (T_{solar, out} - T_{solar, in})}{G_T} \cdot \Delta C_{p, col} \right)^2 + \left( \frac{\dot{V}_{solar} \cdot \rho_{col} \cdot C_{p, col} \cdot \Delta T_{solar, out}}{G_T} \right)^2 \\
+ \left( \frac{\dot{V}_{solar} \cdot \rho_{col} \cdot C_{p, col} \cdot \Delta T_{solar, in}}{G_T} \right)^2 + \left( - \frac{\dot{V}_{solar} \cdot \rho_{col} \cdot C_{p, col} \cdot (T_{solar, out} - T_{solar, in})}{G_T} \cdot \Delta T_{solar, in} \right)^2
\]

b) Theoretical efficiency

Recalling the equation for \( \eta_{solar} \):

\[
\eta_{solar, calc} = \eta_0 \cdot \frac{K_{\tau}}{G_T} (T_{col} - T_{amb}) - \frac{k_1}{G_T} (T_{col} - T_{amb})^2
\]  
Equation 10

Where \( G_T, T_{col} \) and \( T_{amb} \) are measured variables, \( K_{\tau} \) is a derived variable and \( \eta_0, k_1 \) and \( k_2 \) are constants. Applying Equation 9 to Equation 10;

\[
\Delta \eta_{solar, calc} = \sqrt{\left( \frac{\partial \eta_{solar, calc}}{\partial K_{\tau}} \cdot \Delta K_{\tau} \right)^2 + \left( \frac{\partial \eta_{solar, calc}}{\partial G_T} \cdot \Delta G_T \right)^2 + \left( \frac{\partial \eta_{solar, calc}}{\partial T_{col}} \cdot \Delta T_{col} \right)^2} \\
+ \left( \frac{\partial \eta_{solar, calc}}{\partial T_{amb}} \cdot \Delta T_{amb} \right)^2
\]

\[
\left( \eta_0 \cdot \Delta K_{\tau} \right)^2 + \left( \frac{k_1}{G_T^2} (T_{col} - T_{amb}) + \frac{k_2}{G_T^2} (T_{col} - T_{amb})^2 \right) \cdot \Delta G_T \\
+ \left( \frac{k_1}{G_T} + 2 \cdot \frac{k_2}{G_T} (T_{col} - T_{amb}) \right) \cdot \Delta T_{col} \\
+ \left( \frac{k_1}{G_T} + 2 \cdot \frac{k_2}{G_T} (T_{col} - T_{amb}) \right) \cdot \Delta T_{amb} \right)^2
\]

65
7.4.3 Derivation of uncertainty/error of heat required for MD

The heat required by the PGMD module was calculated using the model and then validated against the experimentally determined values for the same operating conditions. Equation 16 and Equation 17 was used to determine the heat required for the experimentally and theoretically determined $Q_{HX2}$ respectively;

a) Experimental $Q_{HX2}$

Recalling the expression (Equation 16) for $Q_{HX2}$;

$$Q_{HX2} = \dot{m}_F \cdot C_{pf} \cdot (T_{evap_{in}} - T_{cond_{out}})$$  \hspace{1cm} \text{Equation 16}

Which, when expressed in terms of measured variables, becomes;

$$Q_{HX2} = \dot{V}_F \cdot \rho_F \cdot C_{pf} \cdot (T_{evap_{in}} - T_{cond_{out}})$$

Where $\dot{V}_F$, $T_{evap_{in}}$ and $T_{cond_{out}}$ are measured variables in the case of the experimental $Q_{HX2}$ and $\rho_F$ and $C_{pf}$ are derived variables. Applying Equation 9 to the above equation;

$$\Delta Q_{HX2} = \sqrt{\left(\frac{\partial Q_{HX2}}{\partial \dot{V}_F} \cdot \Delta \dot{V}_F\right)^2 + \left(\frac{\partial Q_{HX2}}{\partial \rho_F} \cdot \Delta \rho_F\right)^2 + \left(\frac{\partial Q_{HX2}}{\partial C_{pf}} \cdot \Delta C_{pf}\right)^2 + \left(\frac{\partial Q_{HX2}}{\partial T_{evap_{in}}} \cdot \Delta T_{evap_{in}}\right)^2 + \left(\frac{\partial Q_{HX2}}{\partial T_{cond_{out}}} \cdot \Delta T_{cond_{out}}\right)^2}$$

$$= \sqrt{(\rho_F \cdot C_{pf} \cdot (T_{evap_{in}} - T_{cond_{out}}) \cdot \Delta \dot{V}_F)^2 + (\dot{V}_F \cdot \rho_F \cdot C_{pf} \cdot (T_{evap_{in}} - T_{cond_{out}}) \cdot \Delta \rho_F)^2 + (\dot{V}_F \cdot \rho_F \cdot C_{pf} \cdot (T_{evap_{in}} - T_{cond_{out}}) \cdot \Delta C_{pf})^2}$$

b) Theoretical $Q_{HX2}$

Recalling the expression for $Q_{HX2}$ in terms of $\varepsilon$ (Equation 17) gives;

$$Q_{HX2} = \dot{m}_F \cdot C_{pf} \cdot (1 - \varepsilon) \cdot (T_{evap_{in}} - T_{cond_{in}})$$  \hspace{1cm} \text{Equation 28}

Which, when expressed in terms of set operating variables, becomes;

$$Q_{HX2} = \dot{V}_F \cdot \rho_F \cdot C_{pf} \cdot (1 - \varepsilon) \cdot (T_{evap_{in}} - T_{cond_{in}})$$

The set operating variables $\dot{V}_F$, $T_{evap_{in}}$ and $T_{cond_{in}}$ do not have any uncertainty as they are assumed input parameters in the theoretical model. The $\varepsilon$ may be determined using Equation 21
but as it is also a function of the input variables, it's uncertainty is only due to error in the statistical model i.e the root mean square error.

Applying Equation 9 to the above equation yields;

\[
\Delta Q_{HX2} = \sqrt{\left(\frac{\partial Q_{HX2}}{\partial \rho_F} \cdot \Delta \rho_F\right)^2 + \left(\frac{\partial Q_{HX2}}{\partial C_{p_F}} \cdot \Delta C_{p_F}\right)^2 + \left(\frac{\partial Q_{HX2}}{\partial \varepsilon} \cdot \Delta \varepsilon\right)^2}
\]

\[
= \sqrt{\left(\dot{V}_F \cdot C_{p_F} \cdot (1 - \varepsilon) \cdot (T_{\text{evap}_{in}} - T_{\text{cond}_{in}}) \cdot \Delta \rho_F\right)^2 + \left(\dot{V}_F \cdot \rho_F \cdot (1 - \varepsilon) \cdot (T_{\text{evap}_{in}} - T_{\text{cond}_{in}}) \cdot \Delta C_{p_F}\right)^2 + \left(- \dot{V}_F \cdot \rho_F \cdot C_{p_F} \cdot (T_{\text{evap}_{in}} - T_{\text{cond}_{in}}) \cdot \Delta \varepsilon\right)^2}
\]
### 7.5 Appendix E

Table 11: Variation of $UA_{HX2}$ at different operating conditions

<table>
<thead>
<tr>
<th>$T_{cond_{in}}$ °C</th>
<th>$T_{evap_{in}}$ °C</th>
<th>$V_F$ l/h</th>
<th>$UA_{HX2}$ W/°C</th>
</tr>
</thead>
<tbody>
<tr>
<td>20</td>
<td>60</td>
<td>200</td>
<td>113.002</td>
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<td>60</td>
<td>300</td>
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<td>300</td>
<td>191.150</td>
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<td>400</td>
<td>272.508</td>
</tr>
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</table>
7.6 Appendix F

7.6.1 Derivation of Minimum Radiation required for MD operation

For any MD operation, the three defined input parameters are the $\dot{V}_F$, $T_{\text{evap in}}$, and $T_{\text{cond in}}$. Starting from the MD loop, we know that the heat required for MD operation (as in Equation 17) is;

$$Q_{HX} = \dot{m}_F \cdot C_{PF} \cdot (1 - \varepsilon) \cdot (T_{\text{evap in}} - T_{\text{cond in}})$$

$$= \dot{V}_F \cdot \rho_F \cdot C_{PF} \cdot (1 - \varepsilon) \cdot (T_{\text{evap in}} - T_{\text{cond in}})$$

This heat is provided, via the heat exchanger, from the solar loop. Hence, ignoring heat losses, the same heat is transferred from the solar loop and can be expressed (Equation 14) as;

$$Q_{HX} = \dot{m}_{\text{solar}} \cdot C_{PHXsolar} \cdot (T_{HXi\text{n solar}} - T_{HXo\text{ut solar}})$$

Assuming matched flow across the heat exchanger i.e ($\dot{V}_{\text{solar}} = \dot{V}_F$), the $\dot{m}_{\text{solar}}$ can be calculated. Rearranging the above equation;

$$T_{HXi\text{n solar}} - T_{HXo\text{ut solar}} = \frac{Q_{HX}}{\dot{m}_{\text{solar}} \cdot C_{PHXsolar}}$$

Assuming that from the outlet of the solar collector field to the inlet of the heat exchanger, there is a temperature drop of 1.2°C. This value is chosen based on field experience and can be said to deviate from this value by, at most, 0.5°C.

Between the heat exchanger outlet and the solar collector field inlet there is a small heat loss. However this heat loss to the ambient is neglected as it is largely compensated for by the heat of compression which is added to the fluid by the pump. The error associated with this assumption is taken, by experience, as 0.3°C.

Hence the above equation can be rewritten as;

$$T_{\text{solar out}} - T_{\text{solar in}} = \frac{Q_{HX}}{\dot{m}_{\text{solar}} \cdot C_{PHXsolar}} - 1.2$$

Recalling Equation 12, the efficiency of the solar collector field can be expressed as;

$$\eta_{\text{solar}} = \frac{\dot{m}_{\text{solar}} \cdot C_{P\text{cot}} \cdot (T_{\text{solar out}} - T_{\text{solar in}})}{A_{\text{solar}} \cdot G_T}$$

The solar collector efficiency is also given by Equation 10 as;
\[ \eta_{solar} = \eta_0 \cdot K_{\tau \alpha} - \frac{k_1}{G_T} (T_{col} - T_{amb}) - \frac{k_2}{G_T} (T_{col} - T_{amb})^2 \]

In both the above expressions of \( \eta_{solar} \), the average collector temperature \( T_{col} \) is required and may be assumed to be around 10 °C above the average heat exchanger temperature on the MD side. Or in equation form;

\[ T_{col} \approx T_{evap_{in}} + T_{cond_{out}} + 10 \]

This is a reasonable assumption as it provides a sufficient temperature differential to drive the sensible heat exchange across the heat exchanger while simultaneously not being excessively high so as to reduce the efficiency of the solar collectors due to a high operating temperature. In practice, anywhere between 8 °C - 12 °C was used throughout the course of the experiments at the HX2.

The ambient temperature was assumed to be 25 °C.

From the two equations of \( \eta_{solar} \):

\[ \eta_0 \cdot K_{\tau \alpha} - \frac{k_1}{G_T} (T_{col} - T_{amb}) - \frac{k_2}{G_T} (T_{col} - T_{amb})^2 = \frac{m_{solar} \cdot c_{p_{col}} \cdot (T_{solar_{out}} - T_{solar_{in}})}{A_{solar} \cdot G_T} \]

Rearranging in terms of \( G_T \);

\[ G_T = \frac{m_{solar} \cdot c_{p_{col}} \cdot (T_{solar_{out}} - T_{solar_{in}})}{A_{solar} \cdot \eta_0 \cdot K_{\tau \alpha}} + k_1 \cdot (T_{col} - T_{amb}) + k_2 \cdot (T_{col} - T_{amb})^2 \]

For the sake of simplicity, the incident angle modifier is assumed as 1 i.e the calculation is done for solar noon. Hence the final reduced equation for \( G_T \) becomes;

\[ G_T = \frac{1}{\eta_0} \left( \frac{m_{solar} \cdot c_{p_{col}} \cdot (T_{solar_{out}} - T_{solar_{in}})}{A_{solar}} + k_1 \cdot (T_{col} - T_{amb}) + k_2 \cdot (T_{col} - T_{amb})^2 \right) \]
7.6.2 Uncertainty in calculated Minimum Radiation for MD operation

The expression for minimum radiation must first be given in terms of the modelled and derived variables;

\[ G_T = \frac{1}{\eta_0} \left( \dot{V}_{solar} \cdot \rho_{col} \cdot C_{pcol} \cdot \left( \frac{\dot{V}_F \cdot \rho_F \cdot C_{pf} \cdot (1 - \varepsilon) \cdot (T_{evap_{in}} - T_{cond_{in}})}{V_{solar} \cdot \rho_{col} \cdot C_{pHxsolar}} - 1.2 \right) \right) + k_1 \\
\cdot (T_{col} - T_{amb}) + k_2 \cdot (T_{col} - T_{amb})^2 \]

As \( \dot{V}_{solar} = \dot{V}_F \);

\[ G_T = \frac{1}{\eta_0} \left( \dot{V}_F \cdot \rho_{col} \cdot C_{pcol} \cdot \left( \frac{\rho_F \cdot C_{pf} \cdot (1 - \varepsilon) \cdot (T_{evap_{in}} - T_{cond_{in}})}{\rho_{col} \cdot C_{pHxsolar}} - 1.2 \right) \right) + k_1 \\
\cdot (T_{col} - T_{amb}) + k_2 \cdot (T_{col} - T_{amb})^2 \]

Applying Equation 9 to the above equation to determine its uncertainty;

\[ \Delta G_T = \sqrt{\left( \frac{\partial G_T}{\partial \rho_{col}} \Delta \rho_{col} \right)^2 + \left( \frac{\partial G_T}{\partial C_{pcol}} \Delta C_{pcol} \right)^2 + \left( \frac{\partial G_T}{\partial \rho_F} \Delta \rho_F \right)^2 + \left( \frac{\partial G_T}{\partial C_{pf}} \Delta C_{pf} \right)^2} \]

\[ \quad + \left( \frac{\partial G_T}{\partial \varepsilon} \Delta \varepsilon \right)^2 + \left( \frac{\partial G_T}{\partial C_{pHxsolar}} \Delta C_{pHxsolar} \right)^2 \]
7.6.3 Derivation of Auxiliary cooling required for steady state operation

As in the case for the derivation of the minimum radiation required for MD operation, the calculation must start at the MD loop within which lie the defined input parameters \( \dot{V}_F \), \( T_{\text{evap}_{\text{in}}} \) and \( T_{\text{cond}_{\text{in}}} \). The heat required for MD operation (as in Equation 17) is;

\[
Q_{\text{HX}} = \dot{m}_F \cdot C_{pF} \cdot (1 - \varepsilon) \cdot (T_{\text{evap}_{\text{in}}} - T_{\text{cond}_{\text{in}}})
\]

\[
= \dot{V}_F \cdot \rho_F \cdot C_{pF} \cdot (1 - \varepsilon) \cdot (T_{\text{evap}_{\text{in}}} - T_{\text{cond}_{\text{in}}})
\]

This heat is provided, via the heat exchanger, from the solar loop. Hence, ignoring heat losses, the same heat is transferred from the solar loop and can be expressed (Equation 14) as;

\[
Q_{\text{HX}} = \dot{m}_{\text{solar}} \cdot C_{pHXsolar} \cdot (T_{\text{HX in}_{\text{solar}}} - T_{\text{HX out}_{\text{solar}}})
\]

Assuming matched flow across the heat exchanger i.e. \( \dot{V}_{\text{solar}} = \dot{V}_F \), the \( \dot{m}_{\text{solar}} \) can be calculated. Rearranging the above equation;

\[
T_{\text{HX in}_{\text{solar}}} - T_{\text{HX out}_{\text{solar}}} = \frac{Q_{\text{HX}}}{\dot{m}_{\text{solar}} \cdot C_{pHXsolar}}
\]
Assuming that from the outlet of the solar collector field to the inlet of the heat exchanger, there is a temperature drop of 1.2 °C. This value is chosen based on field experience and can be said to deviate from this value by, at most, 0.5 °C.

Between the heat exchanger outlet and the solar collector field inlet there is a small heat loss. However this heat loss to the ambient is neglected as it is largely compensated for by the heat of compression which is added to the fluid by the pump. The error associated with this assumption is taken, by experience, as 0.3 °C.

Hence the above equation can be rewritten as;

\[ T_{solar_{out}} - T_{solar_{in}} = \frac{Q_{HX}}{\dot{m}_{solar} \cdot C_{p_{HX_{solar}}}} - 1.2 \]

The solar collector efficiency is also given by Equation 10 as;

\[ \eta_{solar} = \eta_0 \cdot K_{ta} - \frac{k_1}{G_T} (T_{col} - T_{amb}) - \frac{k_2}{G_T} (T_{col} - T_{amb})^2 \]

Here the ambient temperature is assumed to be 25 °C, the global tilted radiation is assumed as 700 W/m² and the incidence angle modifier is taken as unity for simplification. The average collector temperature may be assumed to be around 10 °C above the average heat exchanger temperature on the MD side. Or in equation form;

\[ T_{col} \approx \frac{T_{evap_{in}} + T_{cond_{out}}}{2} + 10 \]

This is a reasonable assumption as it provides a sufficient temperature differential to drive the sensible heat exchange across the heat exchanger while simultaneously not being excessively high so as to reduce the efficiency of the solar collectors due to a high operating temperature. In practice, anywhere between 8 °C - 12 °C was used throughout the course of the experiments at the HX2.

The heat from the solar collectors \( Q_{solar} \) can be expressed as;

\[ Q_{solar} = \eta_{solar} \cdot G_T \cdot A_{solar} \]

To determine the heat losses between the outlet of the solar collector field and the heat exchanger inlet, the solar outlet temperature as well as the heat exchanger inlet temperature must be determined. The solar outlet temperature can be calculated as the average collector temperature as well as the temperature difference across the collector field is known;

\[ T_{solar_{out}} = \frac{2 \cdot T_{col} + T_{solar_{out}} - T_{solar_{in}}}{2} \]

As mentioned before, the fluid is assumed to lose 1.2 °C to the environment and hence the heat loss (Equation 18) may be given as;

\[ Q_{loss} = 1.2 \cdot \dot{m}_{solar} \cdot C_{psf} \]
By the first law of thermodynamics, i.e. the law of conservation of energy, the heat into the system must be balanced by the heat leaving it and hence the auxiliary cooling required may be expressed as;

\[ Q_{\text{cooler}} = Q_{\text{solar}} - Q_{HX} - Q_{\text{loss}} \]

\[ = \eta_{\text{solar}} \cdot G_T \cdot A_{\text{solar}} - \dot{V}_F \cdot \rho_F \cdot C_{pF} \cdot (1 - \varepsilon) \cdot (T_{\text{evap in}} - T_{\text{cond in}}) - 1.2 \cdot \dot{m}_{\text{solar}} \cdot C_{\text{psf}} \]

7.6.4 Uncertainty in calculated Auxiliary Cooling required for steady state

Rewriting the previously derived equation for auxiliary cooling in terms of modelled and derived variables;

\[ Q_{\text{cooler}} = \eta_{\text{solar}} \cdot G_T \cdot A_{\text{solar}} - \dot{V}_F \cdot \rho_F \cdot C_{pF} \cdot (1 - \varepsilon) \cdot (T_{\text{evap in}} - T_{\text{cond in}}) - 1.2 \cdot \dot{V}_{\text{solar}} \cdot \rho_{\text{SF}} \cdot C_{\text{psf}} \]

Where \( \rho_{\text{SF}} \) is the density of propylene glycol at solar field temperature [kg/m\(^3\)]

As \( \dot{V}_{\text{solar}} = \dot{V}_F \);

\[ Q_{\text{cooler}} = \eta_{\text{solar}} \cdot G_T \cdot A_{\text{solar}} - \dot{V}_F \cdot \rho_F \cdot C_{pF} \cdot (1 - \varepsilon) \cdot (T_{\text{evap in}} - T_{\text{cond in}}) - 1.2 \cdot \dot{V}_F \cdot \rho_{\text{SF}} \cdot C_{\text{psf}} \]

Applying Equation 9 to the above equation;

\[
\Delta Q_{\text{cooler}} = \sqrt{\left(\frac{\partial Q_{\text{cooler}}}{\partial \rho_F} \cdot \Delta \rho_F\right)^2 + \left(\frac{\partial Q_{\text{cooler}}}{\partial C_{pF}} \cdot \Delta C_{pF}\right)^2 + \left(\frac{\partial Q_{\text{cooler}}}{\partial \varepsilon} \cdot \Delta \varepsilon\right)^2 + \left(\frac{\partial Q_{\text{cooler}}}{\partial \rho_{\text{SF}}} \cdot \Delta \rho_{\text{SF}}\right)^2 + \left(\frac{\partial Q_{\text{cooler}}}{\partial C_{\text{psf}}} \cdot \Delta C_{\text{psf}}\right)^2}
\]

\[
= \sqrt{(-\dot{V}_F \cdot C_{pF} \cdot (1 - \varepsilon) \cdot (T_{\text{evap in}} - T_{\text{cond in}}) \cdot \Delta \rho_F)^2 + (-\dot{V}_F \cdot \rho_F \cdot (1 - \varepsilon) \cdot (T_{\text{evap in}} - T_{\text{cond in}}) \cdot \Delta C_{pF})^2 + (\dot{V}_F \cdot \rho_F \cdot C_{pF} \cdot (T_{\text{evap in}} - T_{\text{cond in}}) \cdot \Delta \varepsilon)^2 + (-1.2 \cdot \dot{V}_F \cdot C_{\text{psf}} \cdot \Delta \rho_{\text{sf}})^2 + (-1.2 \cdot \dot{V}_F \cdot \rho_{\text{SF}} \cdot \Delta C_{\text{psf}})^2}
\]
Check-list before submitting your first draft

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☒ Use the check-list in [7] for the SI rules and style conventions.

☒ The Abstract should include the important parts of introduction, method, results and conclusion with focus on method and conclusions.

☒ The Abstract should stand alone from the report, thus no references or cross-reference to figures or tables should be made.

☒ All abbreviations have to be listed in alphabetical order in the Abbreviation section and have to be introduced the first time they are used in the text; same for the nomenclature list.

☒ Do not include a list of figures to the thesis.

☒ Does the Introduction give a good background to the research question and is it put into perspective and do you describe why it is relevant to make this investigation?

☒ Does the Method section clearly describe the procedure of the study and performed experiments and are the main simplifications and limitations of the method discussed.

☒ Preferably the used method should be motivated in relation to the aim of the study and other possible methods.

☒ Are the results clearly presented and easy to understand for the reader?

☒ Have you really looked at the results with critical eyes and tried to find contradictory data and unexplainable results or trends?

☒ Have you critically evaluated the results and critically discussed simplifications and uncertainties in the method and possible implications for the results due to these imperfections?
Have you critically evaluated the results and critically discussed simplifications and uncertainties in the method and possible implications for the results due to these imperfections?

Are your results put in relation to the results from other similar studies?

Excluding or avoiding the presentation of data which does not fit what you expect is strictly forbidden. Just if you know that there was a specific mistake in the measurement or the methodology for that particular data point, then the data could be excluded and the criteria to exclude data should be given in the Method section.

Recommendation for further work should be included in the thesis.

Are all figures, tables, equations, references and appendices referred to in the text? Use the Word cross-reference function.

A report is usually a formal text where personal words such as “I”, “we”, “us”, “our” etc. are seldom or never used. Therefore formulate without using these words, i.e. the passive form, e.g.:
Do not write:
I will investigate the effect of climate on xxxx.

Instead:
The effect of climate on xxxx will be investigated.

Original figures (copy/paste) used from other sources should be referred in the figure text and permission needs to be requested from the author. In that case the figure text will include “[ref] with permission from (the publisher/copyright owner)” This should be only done if it is really necessary to understand the context. Otherwise avoid it.

Redrawn figures used from other sources should be referred in the figure text. In that case the figure text is “Reprinted from [ref]”.

All statements and conclusions presented must be backed up with either a reference, a logical discussion, which explains the point of view or it should be concluded from your own results.

If you use labels in equations or figures they must be explained the first time they appear and then shown in a nomenclature list (with units) placed before the reference section.

Use consistent terminology if you talk about the same thing; e.g. do not use once “solar radiation” and somewhere else “sunlight”.
Summary of your thesis for the examiner

Right from the start of this thesis work, one of the main challenges was the balance between membrane distillation (which has little to nothing to do with solar energy by itself) and the solar thermal part. My local supervisor at the PSA, Guillermo Zaragoza, understood this and hence left it up to me to decide to what extent I wanted to include the solar part. Of course I couldn’t make it a purely solar thermal thesis because that would not work for the needs of the PSA and the company whose MD module I was testing. Mats Ronnelid, who was my supervisor at DU, was not very knowledgeable on this topic but all three of us agreed that, to some extent, the thesis should be a hybrid between solar thermal and membrane distillation. This is the main reason why along with performing the characterisation of the MD module (which in my opinion is a thesis in and of itself), I worked on developing a mathematical model to cover the whole operation and experimental setup including the solar thermal field.

Unfortunately I did not have the time to make the mathematical model in more depth and I would have liked to have been able to model the losses as well as the tank loop better but this was just not possible in the time frame. There were several delays in the experiment work with the final experiment only taking place on the 25th of May due to a combination of bad weather, control system (SCADA) problems, maintenance work on the system etc. Thankfully, Mats was flexible enough to allow me to finish my thesis in June despite these delays.

The evaluation that Mats has filled out was done before all the experiments had been complete and though the thesis that he read then was almost complete, I had added and edited some parts based on the points he mentioned in this assessment and comments.

I would’ve like to have done a multi-response optimisation analysis, and it would not have been very difficult to do using the desirability theory, but it would not have been so meaningful given that the characterisation was not done for the entire operation range of the MD module and the level of uncertainty in the results.

The only way to reduce this uncertainty, I was informed by Dr Zaragoza who was my local supervisor and very knowledgeable in the field, is to perform repetitions of experiments and use the average value of results obtained and the standard deviations as errors. Of course, because of time constraints this was not possible for me and I could only perform one or at most two repetitions of an experiment run.

I think that I have written a fairly comprehensive thesis, if not a bit long, and both my supervisors were happy with the work I did in the given time span. It was a great practical learning experience for me to work with the world class facilities at the PSA and after given me a brief introduction to the system, Guillermo left me to access and operate the system completely independently with little to no supervision. However we did discuss the results in depth and I hope that has come through in the results and discussion section.

Thank you for your time and sorry for making such a long thesis!