

**Solar-Powered Water Desalination using Multistage Circulating
Water Gap Membrane Distillation System**

BY

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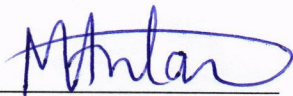
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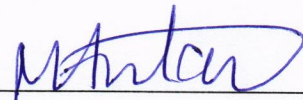
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DEDICATION

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LIST OF ABBREVIATIONS

A_f	:	Feed-Membrane Contact Area [m ²]
A_{gap}	:	Heat transfer area inside the water gap [m ²]
$A_{Membrane}$	:	Membrane vapor crossing area [m ²]
A_{plate}	:	Heat transfer area of the cooling plate [m ²]
$A_{support, solid}$	:	Area of the solid portion of the support plate [m ²]
A_v	:	Membrane effective permeation area [m ²]
ACO	:	Ant Colony Optimization
AGMD	:	Air Gap Membrane Distillation
ANN	:	Artificial Neural Networks
AGR	:	Air Gap Region
BTC	:	Base Test Case
$\dot{C}_{BD}$	:	Yearly Brine Disposal Cost [US\$/m ³ .Year]
$\dot{C}_{CT}$	:	Yearly Chemical Treatment Cost [US\$/m ³ .Year]
$\dot{C}_E$	:	Yearly Energy Cost [US\$/m ³ .Year]
$\dot{C}_L$	:	Yearly Labor Cost [US\$/m ³ .Year]
$\dot{C}_{MRC}$	:	Yearly Membrane Replacement Cost [US\$/m ³ .Year]

$\dot{C}_{SP}$	:	Yearly Maintenance and Spare-part Cost [US\$/m ³ .Year]
$\dot{C}_T$	:	Total Annual Cost [US\$/m ³ .Year]
CAPEX	:	Capital Expenditures [US\$]
CDI	:	Capacitive Deionization
CEMD	:	Continuous-Effect Membrane Distillation
CFPGMD	:	Counter-Flow Permeate Gap Membrane Distillation
COM	:	Cost of Membrane [US\$]
C_p	:	Heat Capacity [J/Kg.K]
CPGMD	:	Circulating Permeate Gap Membrane Distillation
DCMD	:	Direct Contact Membrane Distillation
DE	:	Differential Evolution
DHR	:	Double Heat Recovery
D_e	:	Equivalent Diffusion coefficient [m ² /sec]
D_h	:	Inner hydraulic diameter of the solar collector tube [m]
$D_{h,gap}$	:	Hydraulic diameter in the distillation gap [m]
D_i	:	Tube inner diameter [m]
D_k	:	Knudsen diffusion mode coefficient [m ² /sec]

D_m	:	Molecular diffusion mode coefficient [m^2/sec]
EES	:	Engineering Equation Solver
$\dot{E}_{in}$	:	Total power consumed by the MD unit [W]
ETSC	:	Evacuated Tubes Solar Collectors
F	:	Fin efficiency factor
F'	:	Collector efficiency factor
F''	:	Fluid flow factor
FPC	:	Flat Plate Solar Collector
F_R	:	Heat Removal Factor
GOR	:	Gain Output Ratio
GOR_{STD}	:	Gain Output Ratio for Solar Thermal Desalination
HRR	:	Heat Recovery Ratio
HWT	:	Hot Water Storage Tank
J_w	:	Permeate Flux [$Kg/m^2.hr$]
K_{insulation}	:	Thermal conductivity of thermal insulation [W/m.K]
K_p	:	Thermal conductivity of the absorber plate [W/m.K]
KSA	:	Kingdom of Saudi Arabia

L	:	Length of the distillation channels [m]
LEP	:	Liquid Entry Pressure [Pa]
LGMD	:	Liquid Gap Membrane Distillation
LFR	:	Linear Fresnel Reflectors
$\dot{m}$	:	Mass flow rate of fluid [Kg/Sec]
MD	:	Membrane Distillation
MED	:	Multi-Effect Distillation
MLP	:	Multi-layer Perceptron
MSF	:	Multi-Stage Flash Distillation
MVC	:	Mechanical Vapor Compression
Nu	:	Nusselt number
Nu_{f,i}	:	Nusselt Number for fluid inside tubes
Nu_{gap}	:	Nusselt number in the distillation gap
OPEX	:	Operating Expenditures [US\$]
PGR	:	Permeate Gap Region
P_{mf}	:	Vapor pressure on the hot side of the membrane [Pa]
P_{ms}	:	Vapor pressure on the cold side of the membrane [Pa]

PSO	:	Particle Swarm Optimization
PV-RO:		Photovoltaic-Powered Reverse Osmosis
PVDF	:	Polyvinylidene Fluoride
PTFE	:	Polytetrafluoroethylene
$\dot{Q}_{\text{Cooling,MD}}$	:	Required cooling thermal energy supplied to the MD system [W]
$\dot{Q}_{\text{Heating,MD}}$	:	Required heating thermal energy supplied to the MD system [W]
$\dot{Q}_c$	:	Heat transfer rate through the coolant channel [W]
$\dot{Q}_{\text{cooling,plate}}$	:	Heat transfer rate through the cooling plate [W]
$\dot{Q}_f$	:	Heat transfer rate in the feed channel [W]
$\dot{Q}_{\text{gap}}$	:	Heat transfer rate inside the water gap [W]
$\dot{Q}_{\text{support,plate}}$	:	Heat transfer rate through the support plate [W]
$\dot{Q}_{\text{Top}}$	:	Top surface heat loss from the solar thermal collector [W]
$\dot{Q}_u$	:	Useful energy gained from the solar thermal collector [W]
$\dot{R}_{MEM}$	:	Rate of Membrane Replacement [Year ⁻¹]
Ra	:	Rayleigh number
Re	:	Reynolds number
Re_{gap}	:	Reynolds number in the gap

RES	:	Renewable Energy Resources
RBF	:	Radial Basis Functions
RSM	:	Response Surface Methodology
RR	:	Feed Recovery Ratio
S	:	Absorbed solar energy
SEC	:	Specific Energy Consumption [W/m^3]
SEEC	:	Specific Electrical Energy Consumption [W/m^3]
SGMD	:	Sweeping Gas Membrane Distillation
SHR	:	Single Heat Recovery
SMD	:	Solar Membrane Distillation
SPMD	:	Solar Powered Membrane Distillation
STEC	:	Specific Energy Consumption [W/m^3]
STT	:	Solar Thermal Technologies
T_{PM}	:	Mean plate temperature [K]
T_a	:	Outside ambient temperature [K]
T_{bf}	:	Feed water bulk temperature [K]
T_{bc}	:	Coolant water bulk temperature [K]

T_{fm}	:	Average fluid temperature inside the collector tubes [K]
T_i	:	Fluid inlet temperature to the solar collector [K]
T_{mf}	:	Membrane surface temperature on the hot feed stream side [K]
T_{ms}	:	Membrane temperature from support plate side [K]
T_{pc}	:	Temperature at the cooling plate from the cold side [K]
T_{pg}	:	Temperature of the cooling plate from the permeate side [K]
T_{sg}	:	Temperature of the support plate surface from permeate side [K]
TVC	:	Thermal Vapor Compression
TRNSYS:		Transient System Simulation Tool
TDS	:	Total Dissolved Solids [PPM]
U_B	:	Bottom Loss Coefficient of the Solar collector [$W/m^2.K$]
U_L	:	Overall heat loss coefficient from the solar collector [$W/m^2.K$]
U_{Top}	:	Top heat loss coefficient from the solar collector [$W/m^2.K$]
UPC	:	Unit Production Cost [US\$/ m^3]
US\$/$m^3$:		US Dollars per Cubic Meter
$\dot{V}_d$	:	Volume flow rate of distillate water [m^3/hr]
VMD	:	Vacuum Membrane Distillation

WEN :	Water-Energy Nexus
WGMD:	Water Gap Membrane Distillation
$\dot{Z}_f$:	Annualized Capital Cost [US\$/m ³]
δ :	Thickness of the membrane [m]
$\delta_{\text{Insulation}}$:	Insulation Thickness the solar thermal collector [m]
δ_p :	Thickness of absorber plate of the solar collector [m]
δ_{plate} :	Thickness of the cooling plate of the membrane distillation [m]
δ_{support} :	Thickness of the support plate of the membrane distillation [m]
ΔH_v :	Enthalpy of vaporization for water [J/Kg]
β' :	Volumetric expansion coefficient [K ⁻¹]
γ_{wf} :	Activity coefficient of water in NaCl solution
ΔT :	Temperature difference [K]
ν :	Kinematic viscosity [m ² /Sec]
α :	Thermal diffusivity [m ² /Sec]
ε :	Emissivity [%]
η :	Efficiency [%]
θ :	Angle [rad]

μ	:	Viscosity [Pa.s]
μ_w	:	Viscosity of water at tube walls temperature [Pa.s]
ρ	:	Density [Kg/m ³]
σ	:	Stefan-Boltzmann constant [W/m ² .K ⁴]
τ	:	Transmittance [%]

ABSTRACT

Full Name : Abdul Hafiz Al Hariri
Thesis Title : Solar-Powered Water Desalination using Multistage Circulating Water Gap Membrane Distillation System
Major Field : Mechanical Engineering – Renewable Energy
Date of Degree : May – 2023

In recent years, the growing demand for freshwater has necessitated the development of more efficient and sustainable desalination technologies to satisfy the growing demand for freshwater. For this reason, this thesis is conducted to provide a sustainable desalination solution through developing and optimizing the performance of a multi-stage permeate water gap membrane distillation with gap circulation for the minimum production cost with providing its thermal power from solar energy.

Using theoretical models, experiments, and optimization techniques, the study enhances the performance of this desalination system. Detailed models were developed for the water gap membrane distillation (MD) process with gap circulation, solar thermal Flat Plate Collector (FPC), and a combined MD and FPC model. The system's performance, considering output flux, energy efficiency, and cost, was analyzed under various conditions. Based on optimum conditions, the study also suggested guidelines for small-scale solar membrane distillation units in rural areas.

The novel water gap MD design with circulated gap water showed significant improvement in output flux and energy efficiency compared to the conventional MD process. Artificial Neural Network (ANN) and Differential Evolution (DE) models were used to predict and

optimize freshwater production from the proposed desalination system, showing a great impact of the increased gap circulation rate to reduce the freshwater production cost. Potential approaches for energy recovery in the multi-stage desalination system were investigated, aiming to increase overall efficiency. Finally, a new circulating water gap is experimentally tested and compared to the conventional WGMD and another compact and energy-efficient circulating water gap multi-stage MD module was designed, manufactured to be tested for continuous operation.

ملخص الرسالة

الاسم الكامل: عبد الحفيظ الحريري

عنوان الرسالة: تحليلية المياه بالطاقة الشمسية باستخدام نظام التقطير الغشائي للمياه المتداولة متعدد المراحل

التخصص: الهندسة الميكانيكية - الطاقة المتجددة

تاريخ الدرجة العلمية: مايو - 2023

في السنوات الأخيرة ، استلزم الطلب المتزايد على المياه العذبة تطوير تقنيات تحليلية أكثر كفاءة واستدامة لتلبية الطلب المتزايد على المياه العذبة .لهذا السبب ، تم إجراء هذه الرسالة لتوفير حل مستدام لتحلية المياه من خلال تطوير وتحسين أداء تقطير غشاء متعدد المراحل يتخلل فجوة المياه مع دوران فجوة بأقل تكلفة إنتاج مع توفير الطاقة الحرارية من الطاقة الشمسية . باستخدام النماذج النظرية والتجارب وتقنيات التحسين ، تعزز الدراسة أداء نظام التحلية هذا . تم تطوير نماذج مفصلة لعملية تقطير غشاء فجوة الماء (MD) مع دوران الفجوة ، ومجمع الألواح المسطحة الحرارية الشمسية (FPC) ، ونموذج MD و FPC مدمج . تم تحليل أداء النظام ، مع الأخذ في الاعتبار تدفق المخرجات وكفاءة الطاقة والتكلفة ، في ظل ظروف مختلفة . بناءً على الظروف المثلى ، اقترحت الدراسة أيضًا مبادئ توجيهية لوحدات تقطير الغشاء الشمسي صغيرة الحجم في المناطق الريفية . أظهر تصميم MD الجديد لفجوة المياه مع فجوة المياه المتداولة تحسناً كبيراً في تدفق المخرجات وكفاءة الطاقة مقارنةً بعملية MD التقليدية . تم استخدام نماذج الشبكة العصبية الاصطناعية (ANN) والتطور التفاضلي (DE) للتنبؤ وتحسين إنتاج المياه العذبة من نظام التحلية المقترح ، مما يدل على التأثير الكبير لمعدل دوران الفجوة المتزايد لتقليل تكلفة إنتاج المياه العذبة . تم فحص الأساليب المحتملة لاستعادة الطاقة في نظام تحليلية المياه متعدد المراحل ، بهدف زيادة الكفاءة الإجمالية . أخيرًا ، تم اختبار فجوة مائية متداولة جديدة بشكل تجريبي ومقارنتها بـ WGMD التقليدي وتم تصميم وحدة MD أخرى متعددة المراحل مدمجة وذات كفاءة في استخدام الطاقة ، ليتم اختبارها للتشغيل المستمر .

CHAPTER 1

INTRODUCTION

1.1. Background

In 2020, different desalination technologies have produced a total amount of 97.2 million m³/day of freshwater globally [1]. This is because only 2.5% of the earth's available water is considered drinkable [2]. Desalination plays a crucial role in addressing the water shortage problem since freshwater is not accessible to many people due to limited renewable supplies and contamination with industries and human beings' wastewater. The physical properties of seawater, such as an average salt concentration of 35 g/L, a density of around 1025 g/L, and a freezing point of approximately -2 °C [3], are important to know for designing and developing water desalination systems. The water-energy nexus is a critical premise of global sustainable development as rapid population, and economic development will put significant stress on the planet's limited resources, including water and energy. The water sector requires energy for various applications, including water desalination and treatment, abstraction, distribution, disposal, and utilization, and investigations of the links between energy and water, known as the Water-Energy Nexus (WEN), have received much attention recently [4].

The Arabian gulf countries such as Saudi Arabia, United Emirates, Kuwait, and Qatar represent in chronological order represents 55% of the total global desalination production [5]. The modern activities observed in Saudi Arabia, and the harsh climate resulted in a

substantial increase in the water demand that may intimidate the water security in the Kingdom. Regardless of the high accessibility of potable water in Saudi Arabia, it is considered the most water-scarce country in the world [6] leading the desalination sector to thrive and to count 31 operating desalination plants in 17 locations, and other valued SR91 billion (\$24.3 billion) under construction desalination plants through 2020 [7]. The Kingdom's desalination plants consumes more than 20% of the total power production [8], [9].

Even though desalination helped Saudi Arabia to overcome the water scarce threatening the Kingdom's water security, it has come on a substantial financial cost because of the high capital and energy consumption requirements [10]. The substantial amount of energy consumption, consequently higher water production cost, intimidates the energy and water stability in the kingdom. The substantial amount of energy consumption, and consequently the higher water production cost, requires the adoption of an economic desalination practice that involves sustainability in terms of resources that could be achieved through the integration of Renewable Energy Resources (RES) such as solar, wind, and geothermal energy with water desalination systems, especially with areas that are rich in such renewable resources for the aim for producing water more efficiently in a reduced production cost. This could be an essential strategy to achieve the vision of Saudi Arabia that aims to reach sustainability in terms of resources and acceleration in terms of the energy transition at the end of 2030 and to attain Net-Zero carbon emissions in 2060 [11]. KSA is an area known for its high solar insolation, where the high availability of solar resources coincides with the intensive water requirements, making the study of solar water desalination an attractive area of study to produce water more efficiently at a reduced

production cost. The literature has reported different desalination techniques and different solar technologies to provide the required energy for the desalination process, which are described in the following sections.

1.2. Water Desalination Techniques

Desalination could be categorized into two main areas, the thermal technology area, and the membrane technology area. However, different emerging desalination technologies have been recently involved and developed by researchers, such as the MD technique, to overcome the drawbacks of the conventional techniques.

1.2.1. Thermal Water Desalination

Regarding thermal technology, liquid-vapor phase transformation of water is the key working concept. The process consists of evaporating the saltwater by adding a certain amount of heat and then condensing this vapor as a form of a freshwater product. The multi-effect distillation (MED), mechanical vapor compression (MVC), multi-stage flash distillation (MSF), and thermal vapor compression (TVC) could be considered as the main thermal desalination technologies [2], these require a large amount of latent heat to provide good water productivity.

1.2.1.1. MSF Desalination Process

MSF desalination involves heating seawater in the heating section, usually by waste steam source [12], followed by rapid cooling in a series of chambers in the heat recovery section. This causes the water to vaporize, or "flash," into steam, leaving dissolved salt and minerals to be disposed of as a form of leaving brine. In the heat recovery section, the thermal energy of the flashed water vapor is recovered by increasing the inlet sea water temperature, which

acts as a coolant in the recovery section, and feed water after being fully heated in the heating section. MSF is considered a reliable water desalination technique when integrated with thermal power plants, such as the unit installed in Ras El Kheir, KSA, that produces 228 million imperial gallons per day [13].

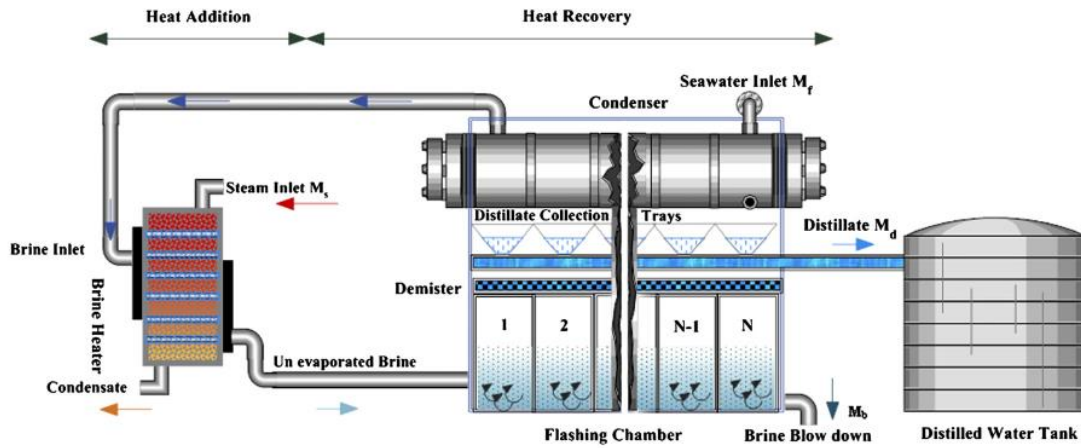


Figure 1.1 Once-through multi-stage flashing process [12]

1.2.1.2. MED Desalination Process

MED desalination is similar to the MSF, but utilizes multiple effects, or stages, to enhance the energy efficiency [14]. The process involves using the steam generated in one stage to heat seawater in the next stage, thereby reducing the energy required to produce freshwater[15]. MED desalination can produce freshwater at a rate of to 146,000 cubic meters per day (m^3/day), such that in Yanbu desalination plant, prior to the latest expansion project [16].

In the MED (Multiple Effect Distillation) process (as shown in Figure 1.2), seawater is introduced by mean of sprayers into each effect and then heated using steam from a power plant. The heated water in the first effect then becomes the energy source to heat the seawater in the second effect, where it turns into freshwater through condensation. This

process is repeated in subsequent effects, using the vaporized seawater as the energy source. This is the most efficient method of thermal desalination, and its performance improves as the inlet steam temperature increases [17].

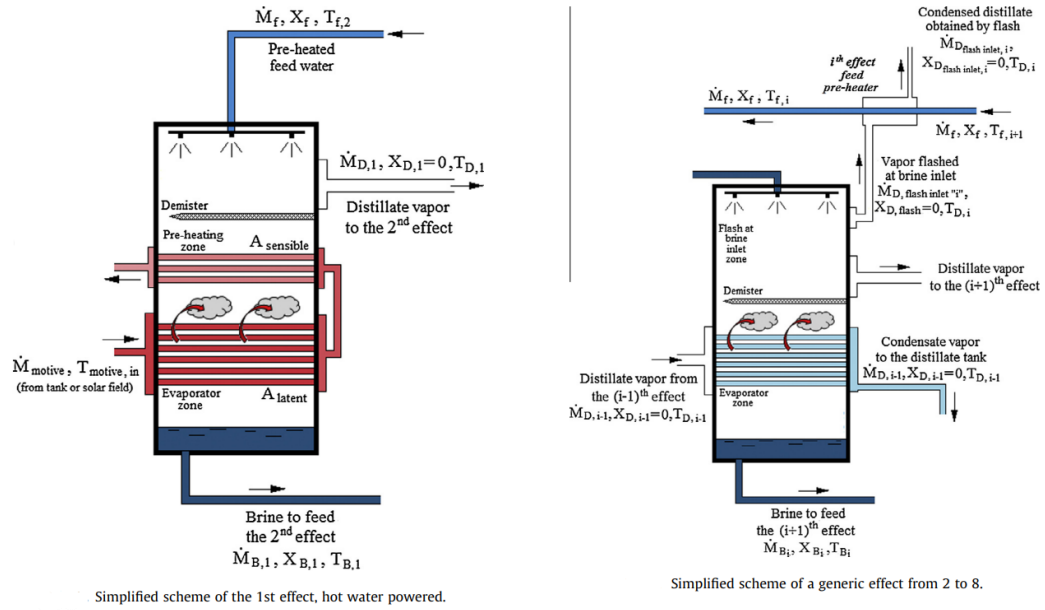


Figure 1.2 Forward-feed MED process [17]

1.2.2. Membrane Water Desalination

Membrane desalination is a process that utilizes a semi-permeable membrane to separate salt and other dissolved minerals from seawater, rendering it potable. Reverse osmosis (RO) is the dominant separation technology. RO is used to separate salt from brackish or seawater using a high-pressure process, as shown in Figure 1.3, that increases as the water salinity increases.

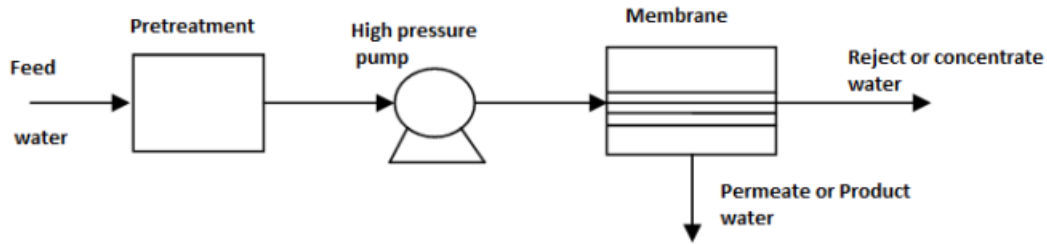


Figure 1.3 Schematic diagram of RO desalination process [18]

This pressure is required to reverse the natural osmosis pressure, Figure 1.4, to allow liquid to pass from saline solution to fresh permeate water. It is the most efficient process with a Specific Electrical Energy Consumption (SEEC) between 2 to 8 kWh/m³ and costs less than 1 USD/ m³ [19].

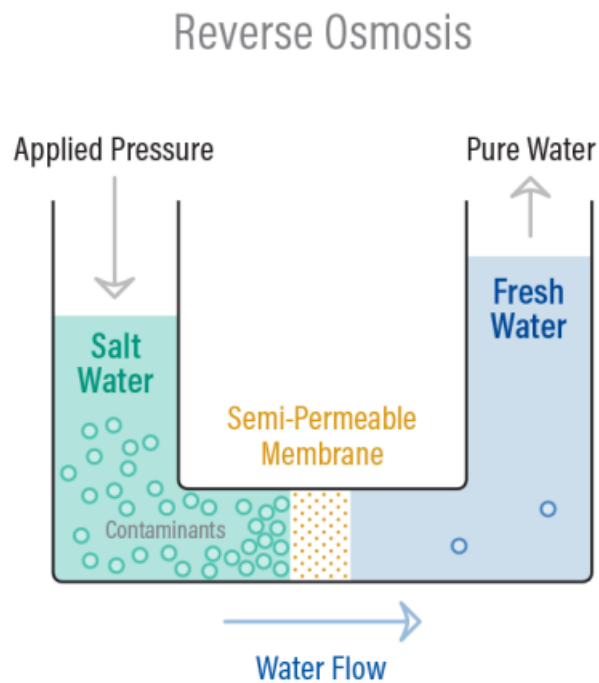


Figure 1.4 RO separation process

Considering its lower production cost, because of its lower energy consumption, RO is widely used in different desalination plants in Saudi Arabia such as the largest desalination plant in Ras Al Kheir producing more than 728 million liters (728,000 m³) per day [20].

Membrane desalination technologies are relatively simple and easy to operate and are becoming increasingly popular due to their ability to produce freshwater in different capacities at a relatively low cost. However, membrane desalination also requires adequate pre-treatment process, periodic membrane replacement, and it has limitation to treat water with higher salinities [21]. Hence, it is important to consider the system reliability, energy requirements and the cost of operation and maintenance when evaluating the feasibility of membrane desalination projects.

1.2.3. Emerging Water Desalination Techniques

Emerging desalination technologies are new and innovative ways to remove salt and other dissolved minerals from seawater, making it safe to drink. Some of the most promising emerging technologies include:

1. Forward osmosis (FO): FO is a process that utilizes a semi-permeable membrane and a draw solution to separate salt and other dissolved minerals from seawater. The process is less energy-intensive than traditional reverse osmosis (RO) desalination and has the potential to produce freshwater at a lower cost. FO process involves the osmotic dilution of the draw solution then generating fresh water from this diluted draw solution as shown in Figure 1.5. Saline feed water and the highly concentrated solute are separated by a semi-permeable membrane, the movement of water from the saline feed water (lower solute concentration) to the concentrated

solute is a natural process due to osmotic gradient, while retaining the solutes on both sides of the membrane.

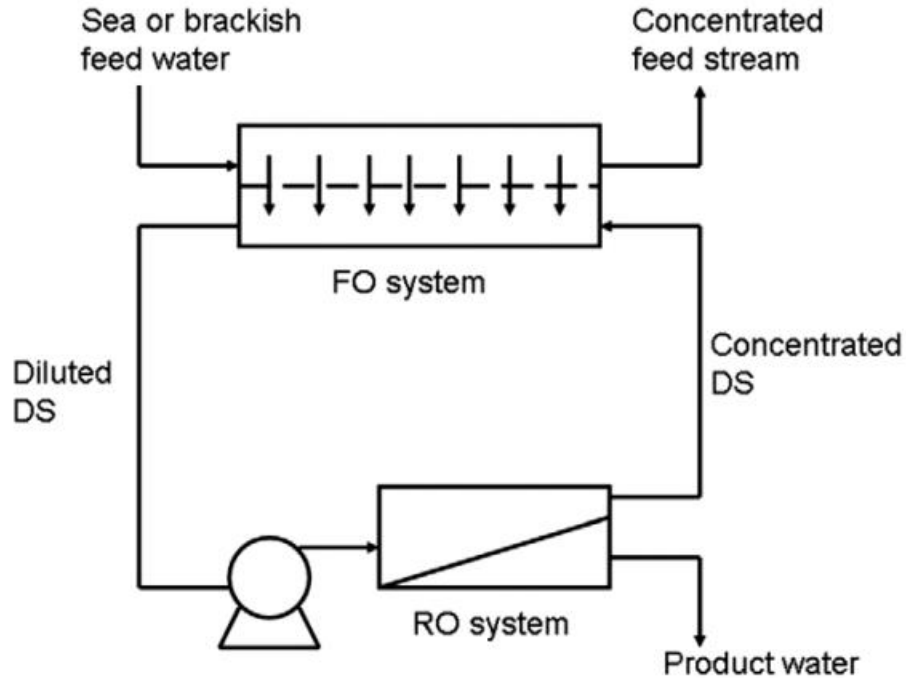


Figure 1.5 Simplified FO/RO desalination process using RO [22]

2. Capacitive deionization (CDI): CDI is an electrochemical process that utilizes electrodes to remove ions from seawater. The process is highly efficient in removing dissolved salts and can be powered by renewable energy sources, making it an attractive option for desalination in remote or off-grid locations.

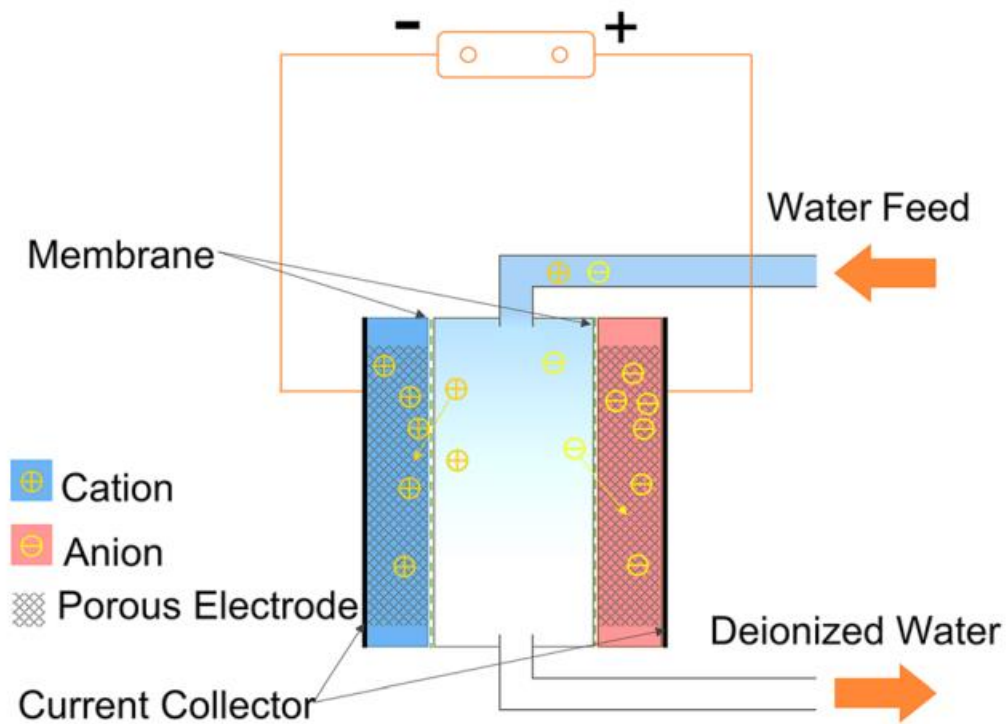


Figure 1.6 Conventional capacitive deionization process [23]

3. Solar-powered desalination: Several solar-powered desalination technologies have been developed in recent years, including solar stills, solar thermal desalination, and photovoltaic-powered reverse osmosis. These technologies utilize the energy from the sun to power desalination, making them an attractive option for desalination in sunny, arid regions, more details will be provided in the following sections.
4. Nanofiltration: Nanofiltration is a membrane filtration process that utilizes a semi-permeable membrane with small pores to separate dissolved salts and other impurities from seawater. This technology is more energy-efficient than traditional

reverse osmosis and can remove dissolved ions such as calcium, magnesium, and bicarbonates [24].

5. **Membrane Distillation (MD):** Membrane distillation (MD) is a thermal desalination process that uses a semi-permeable membrane to separate salt and other dissolved minerals from seawater, rendering it potable. The process is based on the principle of vapor-liquid separation. As a common process in different MD configurations, hot feed water (50–90 °C) is passed over a microporous hydrophobic membrane, Figure 1.7, that allows only vapor to cross and condense on the cold side as a freshwater product. The vapor crosses through the membrane pores because of the vapor pressure difference created by the temperature difference between the hot and cold sides of the membrane. The way the permeated vapor condenses on the cold side of the membrane depends on the MD configuration. There are four main MD configurations, Direct Contact Membrane Distillation (DCMD), Air Gap Membrane Distillation (AGMD), Sweeping Gas Membrane Distillation (SGMD), and Vacuum Membrane Desalination (VMD) [25].

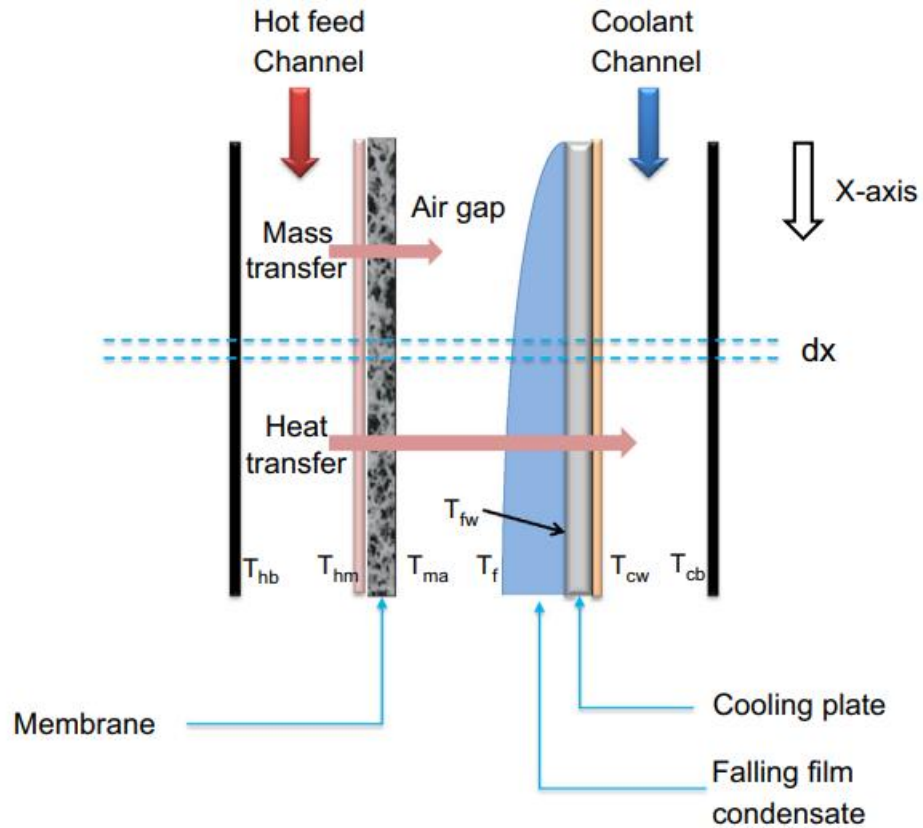


Figure 1.7 Mass and heat transfer across an AGMD [26]

Membrane distillation (MD) is considered an emerging desalination technology for several reasons:

- High Salt Rejection Ratio: MD can achieve up to 98% salt rejection making it a highly efficient desalination technology.
- Low Temperature Operation: MD operates at relatively low temperatures and pressures, making it less energy-intensive than traditional thermal and membrane desalination technologies with an ability to operate with low-grade heat source such as waste or solar thermal heat.
- Versatility: MD can be used to desalinate a wide range of water sources, including brine water, seawater, brackish water, and wastewater.

- Low fouling: MD is less prone to fouling, which is the buildup of impurities on the membrane surface, which reduces the efficiency of the process, and therefore, a lower maintenance is needed.
- Low cost: MD modules are easily manufactured with plastic materials without the need of large metal heat exchanger, making the capital cost of investment attractive comparing to conventional sources.
- Scalability: MD can be scaled up or down depending on the desired water production rate.

All these factors make MD an attractive option for desalination in a wide range of applications, from small-scale village systems to large-scale industrial plants. However, it is worth to mention that MD is still a relatively new technology, and more research is needed to optimize the process and reduce the costs of implementation. Also, the technology is still in the development stage and its feasibility and effectiveness are still being evaluated.

1.3. Solar Thermal Technologies

The efficient utilization of solar thermal energy as a part of renewable energy resources is being considered as a promising solution for the global warming challenges. This is because, the Sun releases a massive amount of radiation energy into the surrounding space, it reaches to 174 PW (1 PW = 10^{15} W) at the extraterrestrial atmosphere [27]. The energy arriving to the earth is attenuated by both the atmosphere (6% by reflection and 16% by absorption) and the clouds (20% by reflection and 3% by absorption) [28], as shown in Figure 1.8. Another 51% (89 PW) of the whole incoming solar energy reaches the oceans and the lands [27]. It is obvious that the total solar radiation existing on the Earth is still a

considerable amount, but because of the daily and season intermittency, the available solar energy must be collected through solar thermal collectors to be used directly or stored efficiently.

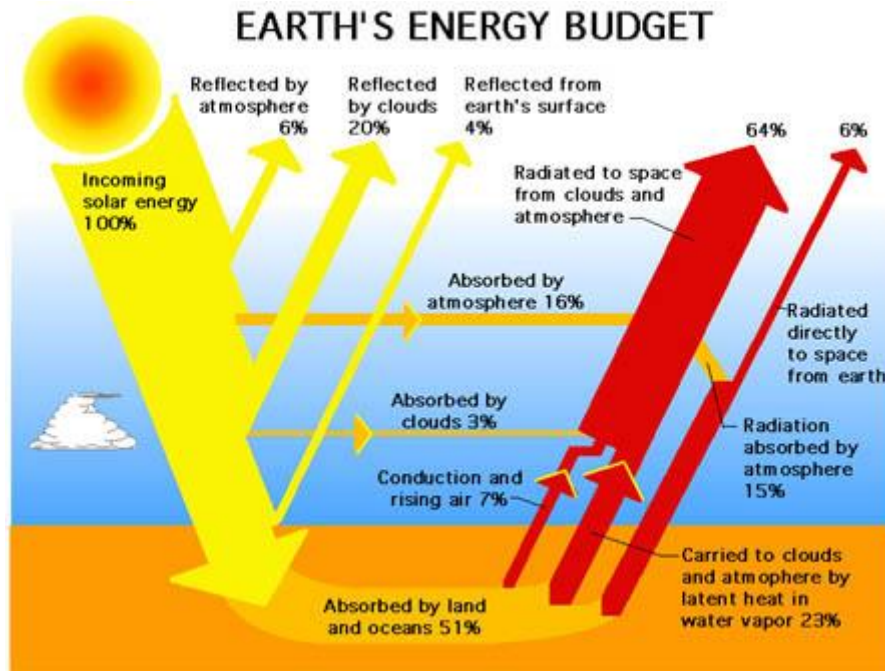


Figure 1.8 Energy balance for terrestrial solar radiation [28]

According to [29], solar thermal collectors are divided into two main categories, the non-concentrating (Low-Temperature Applications) and the concentrating collectors (High-Temperature Applications). A non-concentrating solar thermal collector has an equal intercepting area to its absorbing area, on the other hand, a concentrating solar collector usually has reflecting surfaces with concave shape to concentrate the solar irradiation to another smaller receiving area, resulting in an increased heat flux resulting to higher temperatures. Kalogirou [30] has divided solar collectors into tracking and non-tracking with an indication of the concentration ratio and temperature range as shown in the diagram in Figure 1.9.

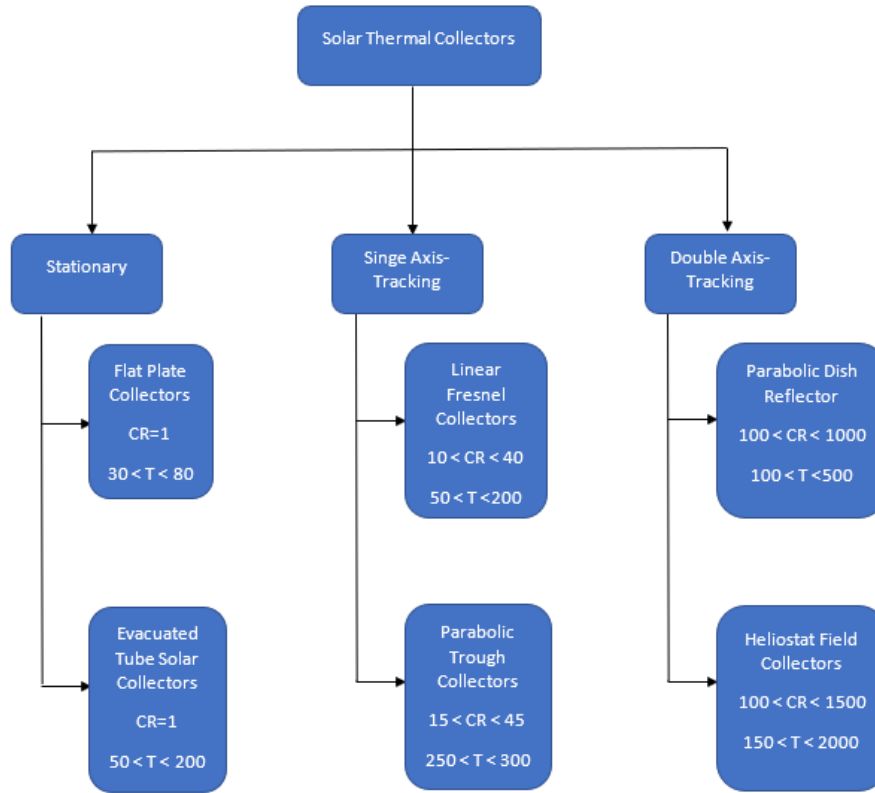


Figure 1.9 Solar thermal collectors classifications [30]

1.3.1. Non-concentrating Solar Collectors

- **Flat Plate Solar Collectors:**

Typically, a flat plate solar thermal collector consists of glazing covers made of single or multiple glass sheets with high short-wave radiation transmissivity and low long-wave radiation transmissivity, absorber plates, bottom and edge insulation layers, recuperating tubes (carrying heat transfer fluids) and other supplementary accessories. Glazing does not only reduce convection heat losses from the absorber plate, but also reduces irradiation losses from the absorber plate in a way like a greenhouse effect. The typical flat plate layout is presented in Figure 1.10.

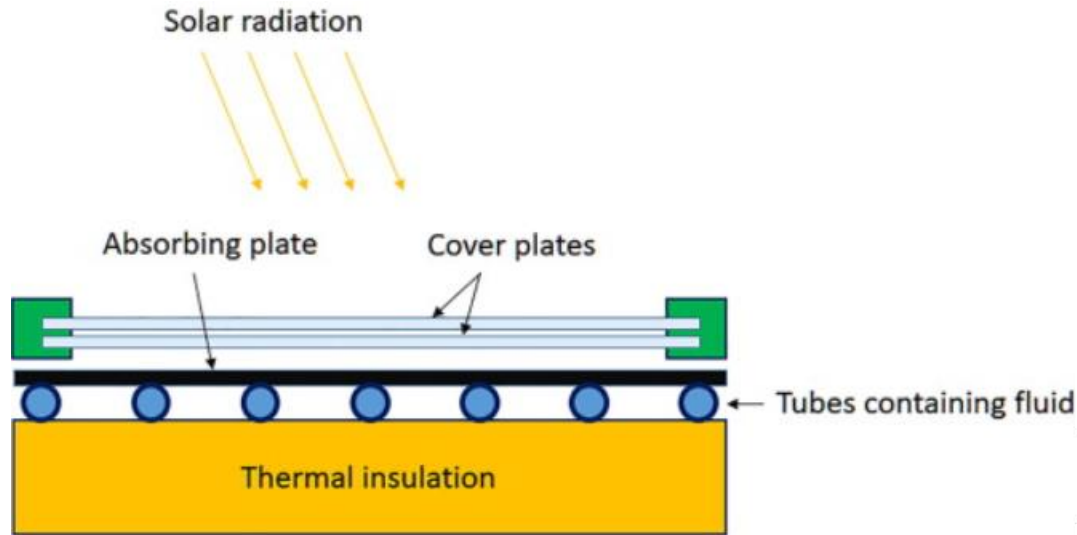


Figure 1.10 Solar thermal flat plate collector [31]

When solar radiation crosses the transparent covers to the blackened absorber plate with high absorptivity, a significant fraction of this energy is absorbed then transferred through heat transfer fluid tubes to be carried away for direct use or storage systems. The bottom side of the absorber plate and the edge casing are extremely isolated to eliminate the conduction heat losses. The fluid carrying tubes can be integrated with the absorber plate or welded with high conductivity materials.

According to review of Kalogirou [30], FPC are used in Thermosyphon Potable Water Heating, Direct Circulation Water Heating, Stored Water Heating, in Air and Water Space Heating, Solar Refrigeration, and thermal water desalination such as solar stills, MD, MSF, and MED.

- **Evacuated Tubes Solar Collectors:**

The literature of [32] claimed that many researchers found that ETSC has better performance and efficiency comparing with a typical FPSC, especially when operating at

low temperature and low solar insolation. Typically, as in Figure 1.11, an ETSC is formed of two parallel evacuated glass pipes. Usually, outer tube is transparent to allow light rays pass to through to be absorbed by the inner tube which is coated by selective material to increase the overall absorption. Both the outer and inner tubes come up with least reflection properties. A vacuum is created which allows the solar radiation to pass the outer tube to heat the inner tube but does not allow the heat to transfer outside the tubes. To obtain this vacuum in manufacturing, the two tubes must be fused together on top and the existing air is pumped out. Consequently, the heat remains inside the inner pipes which collects the solar radiation efficiently.

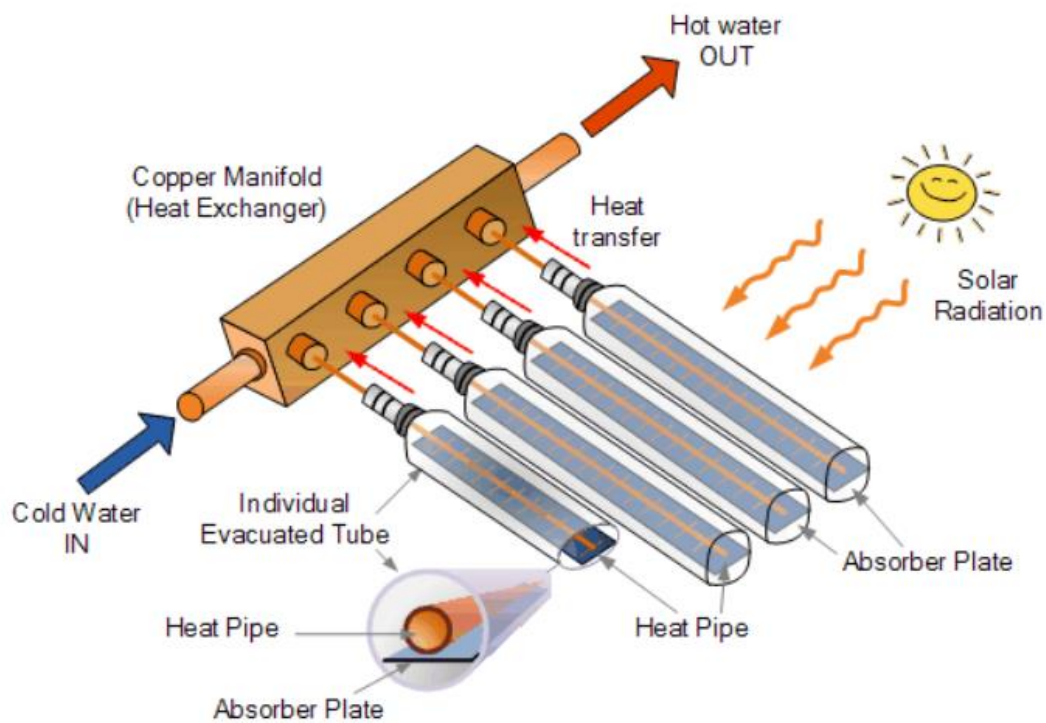


Figure 1.11 Typical evacuated tube solar collector configuration [33]

Mangal et al. [34] stated that FPC perform its peak energy output only at mid-day when the sun strikes perpendicularly the surface of the collector, in contrast, the ETSC are capable to passively track the sun during the whole day period owing to the cylindrical shape of evacuated tube. In other words, the incident angle of solar rays on the tubular shape is always equal to 90° throughout the entire day; hence the absorption of ETSC is always greater than of FPC. Since the Vacuum is created to reduce the convective and conductive heat transfer, the ETSC is less affected by low temperature and wind. In terms of maintenance, they reported that ETSC has a lower maintenance cost, because if one of the tubes is damaged it can be replaced separately without affecting the continuous operation. ETSC could be used for the same application of FPC except the Thermo-Syphon Water Heating and Air Heating Systems.

The efficiency of different solar collectors as a function of the ambient temperature and the available solar radiations are shown in Figure 1.12. It is shown that at high ambient temperature FPC could have higher efficiency than ETSC, whereas ETSC present a stable efficiency over a wide range of ambient temperature and it could keep high efficiency even at low temperatures. In hot region, such as the KSA, FPC could be an economic choice, this is proven by a research performed by Fahad Al Suleiman in [35], which revealed that FPC provided higher yearly savings and less payback period for the same heating application for different Saudi cities.

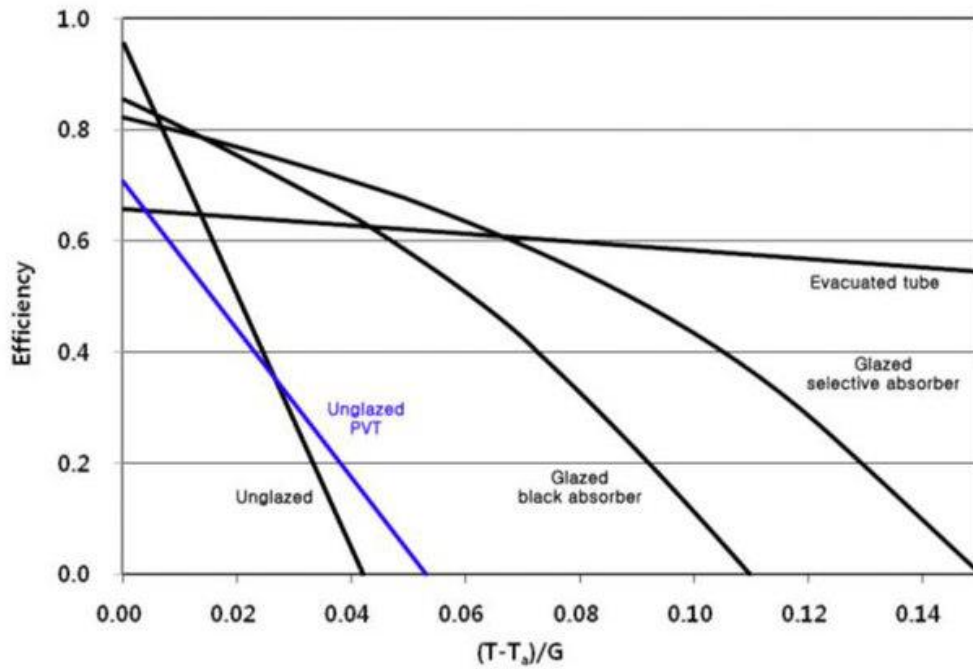


Figure 1.12 The Efficiency of different solar collectors as a function of the ambient temperature and the available solar radiations [36]

1.3.2. Concentrating Solar Collectors

Concentrated solar thermal collectors use mirrors to focus sunlight onto a receiver, where the solar energy is converted into heat. There are several different types of concentrated solar thermal collectors, each with their own advantages and disadvantages.

1. **Parabolic Trough Collectors:** These collectors use parabolic mirrors to focus sunlight onto a tube that runs along the focal point of the mirror. The heat collected in the tube is used to generate steam, which powers a turbine to generate electricity [37].

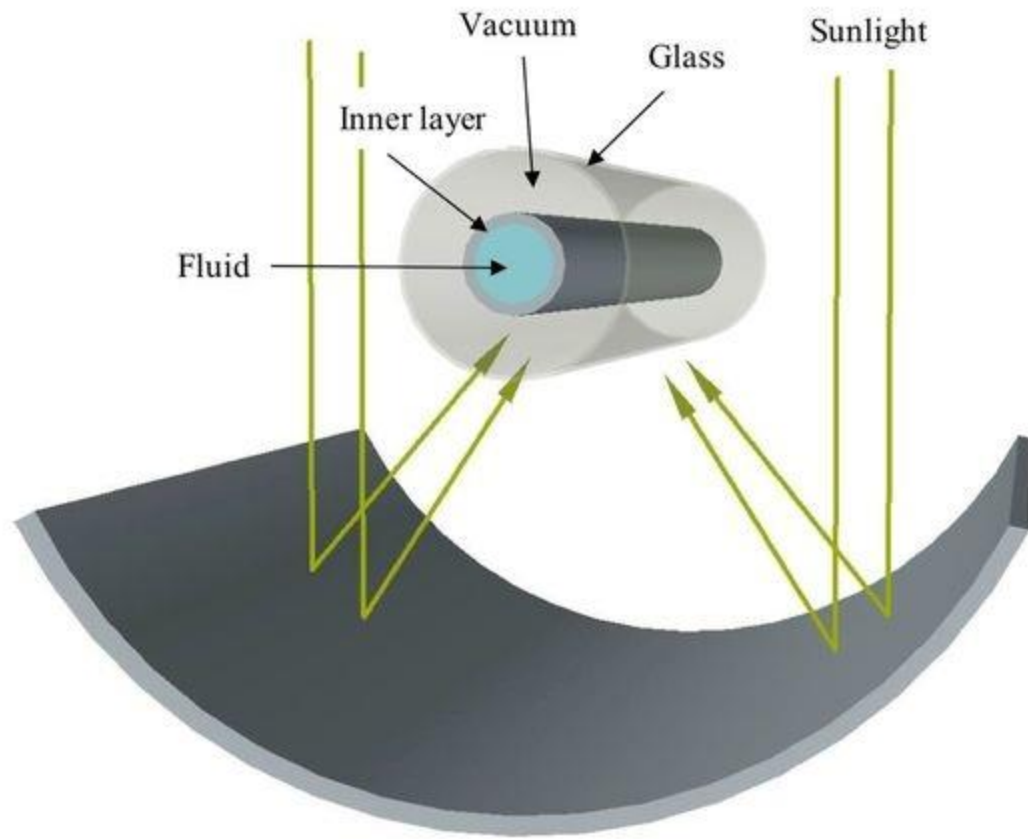


Figure 1.13 Parabolic Trough Collector Configuration [38]

2. **Linear Fresnel Reflectors:** These collectors use a series of flat mirrors to focus sunlight onto a receiver. They are similar to parabolic trough collectors, but use less material and are less expensive to build [39].

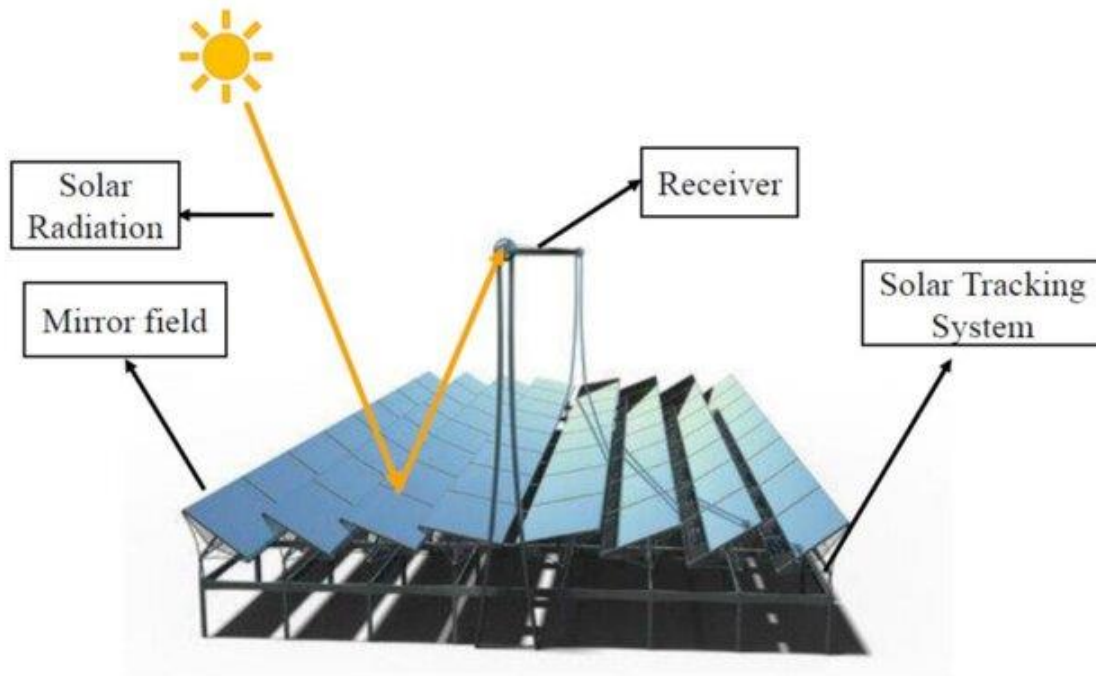


Figure 1.14 Linear Fresnel Collector Configuration [40]

3. **Dish/Engine Systems:** These collectors use a parabolic dish to focus sunlight onto a receiver that is connected to a Stirling engine. The heat collected by the receiver is used to power the engine, which generates electricity [41].
4. **Power Tower Systems:** These collectors use a large field of mirrors to focus sunlight onto a central receiver, which is located at the top of a tower. The heat collected by the receiver is used to generate steam, which powers a turbine to generate electricity [42].

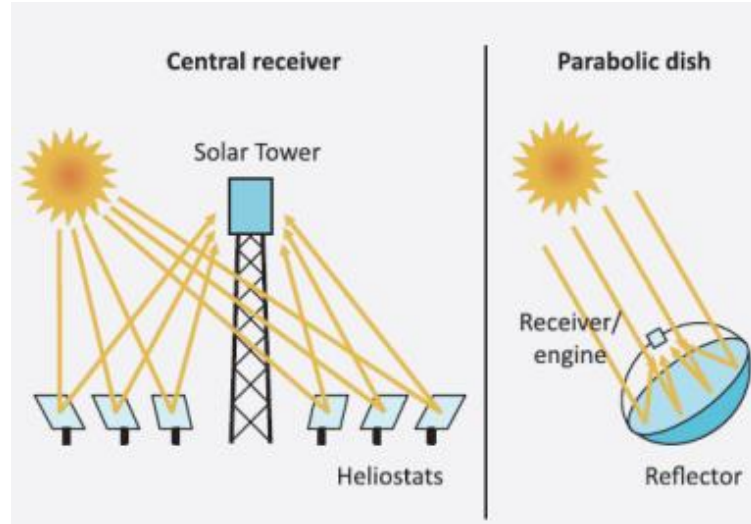


Figure 1.15 Power Tower System (Left), and Parabolic Dish (Right) [43]

Concentrated solar thermal collectors are versatile energy systems that can be utilized in a variety of settings, such as utility-scale power plants, industrial processes, commercial and residential buildings, agriculture, and remote areas. They are most found in sunny regions like deserts and Africa and Middle East but can also be used in other regions as well [37] [39] [41] [42]. In desalination, concentrated solar thermal collectors provide heat for thermal distillation such as multi-stage flash distillation, multi-effect desalination processes, which use heat to vaporize seawater and produce fresh water. The use of concentrated solar thermal collectors in desalination is considered a reliable source of fresh water in regions with limited power resources. Overall, concentrated solar thermal collectors provide an attractive option for desalination in regions with abundant sunlight and limited water resources [44].

1.4. Solar Energy Utilization in Water Desalination

Solar desalination involves the use of solar energy to convert seawater or brackish water into fresh water through various processes. This provides a sustainable and

environmentally friendly solution to the increasing demand for fresh water. The following three subsections will examine the different ways in which solar energy is utilized in the desalination process.

1.4.1. Photovoltaic-Powered Reverse Osmosis

Photovoltaic-powered reverse osmosis (PV-RO) is a technique that uses solar panels to generate electricity, which is then used to power the reverse osmosis (RO) process for desalination. The RO process is previously described, and it requires only electrical energy as an input [45]. By utilizing solar energy, the PV-RO process eliminates the need for fossil fuels and reduces the carbon footprint of desalination [46].

The basic components of a PV-RO system include solar panels, a battery system, a control system, and an RO system. The solar panels convert sunlight into electricity, which is stored in the battery system for use when the sun is not shining. The control system manages the flow of electricity between the battery and the RO system, ensuring that the RO process operates efficiently and effectively.

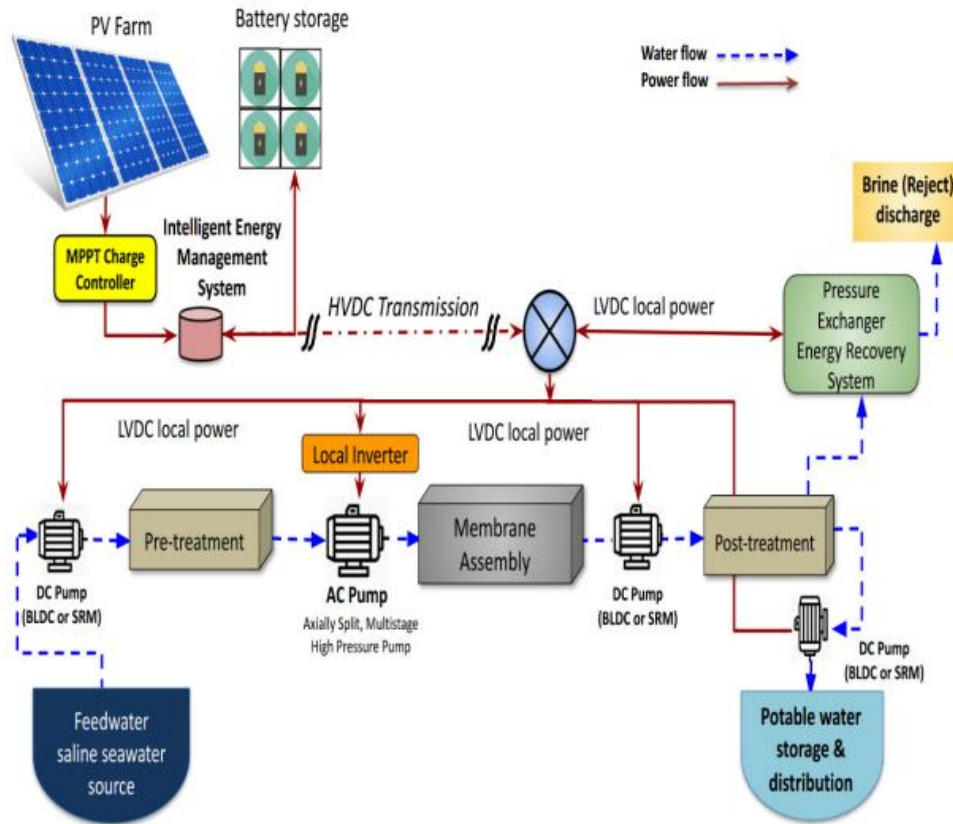


Figure 1.16 Stand-alone solar PV-RO desalination plant [47]

One of the main advantages of PV-RO is its efficiency and sustainability and it does not require regular fuel deliveries. Unlike other desalination techniques that rely on fossil fuels, PV-RO does not produce greenhouse gas emissions or contribute to climate change. Additionally, the use of solar panels provides a relatively low-cost and low-maintenance solution to the demand for fresh water [48].

However, there are also some limitations to PV-RO. For example, the effectiveness of the system may be limited by the availability of sunlight, that requires investing in storage batteries that increase the unit production cost of water or affect overall performance of the desalination process. Additionally, the initial costs of installing and maintaining a PV-RO

system can be high, making it less accessible to communities with limited financial resources [49].

Despite these limitations, PV-RO provides an important solution to the increasing demand for fresh water, especially in remote or underdeveloped areas where access to conventional energy sources may be limited. With advancements in technology and increased investment in renewable energy, it is likely that PV-RO will play an increasingly important role in the desalination of seawater and brackish water.

1.4.2. Solar Energy Utilization in Thermal Desalination

Solar energy utilization in thermal processes involves the use of solar collectors to heat water, which is then used to power the desalination process. This method is relatively low-cost and can be highly efficient, making it a viable solution to the increasing demand for fresh water. The efficient utilization of solar thermal energy to provide heat for different thermal desalination techniques is considered a promising solution for global warming challenges [27].

The desalination process through solar energy could be divided into two main categories, the direct and indirect processes.

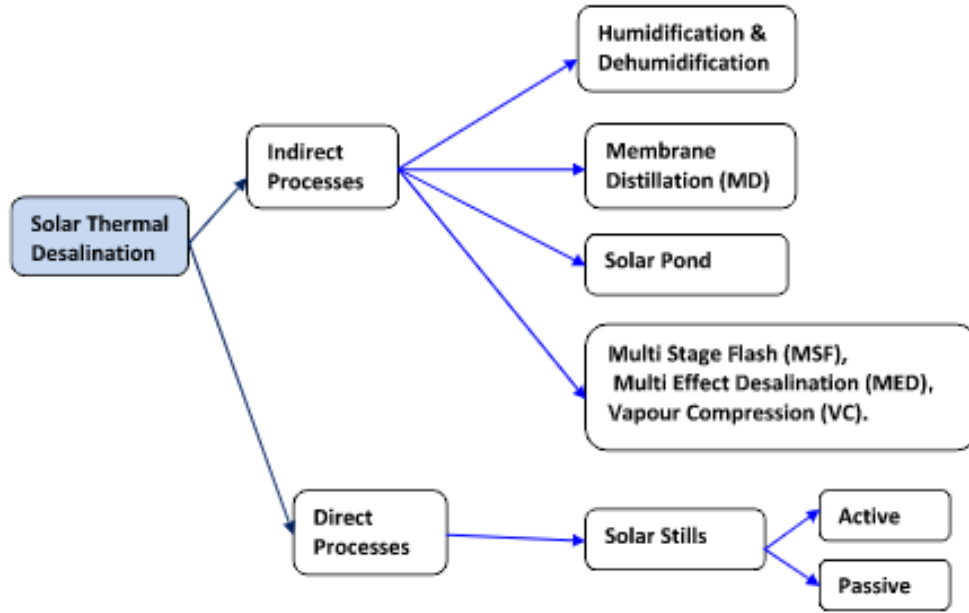


Figure 1.17 Direct/Indirect solar thermal desalination technologies classification [50]

In the direct desalination category, the same device is used to collect the heat from the sun for feed vaporization and condensation as a freshwater product [50], and this could be used only for small-scale desalination units such as a solar still [51]. However, the indirect process is known for consisting of two main subsystems, the solar energy collecting unit, and the desalination unit. For the solar energy collecting unit, solar heat is collected and transmitted to raise the temperature of feed water through different types of solar collectors such as the solar flat plate collectors (FPC), the evacuated tubes solar collectors (ETSC), and the concentrating solar collectors [51]. On the other hand, the desalination process occurs in a separate subsystem unit such as MD, MSF, and MED [52], this category could be used for medium to large desalination systems [51].

The authors also discuss the advantages and disadvantages of each method, as well as the challenges faced in terms of cost, efficiency, and sustainability.

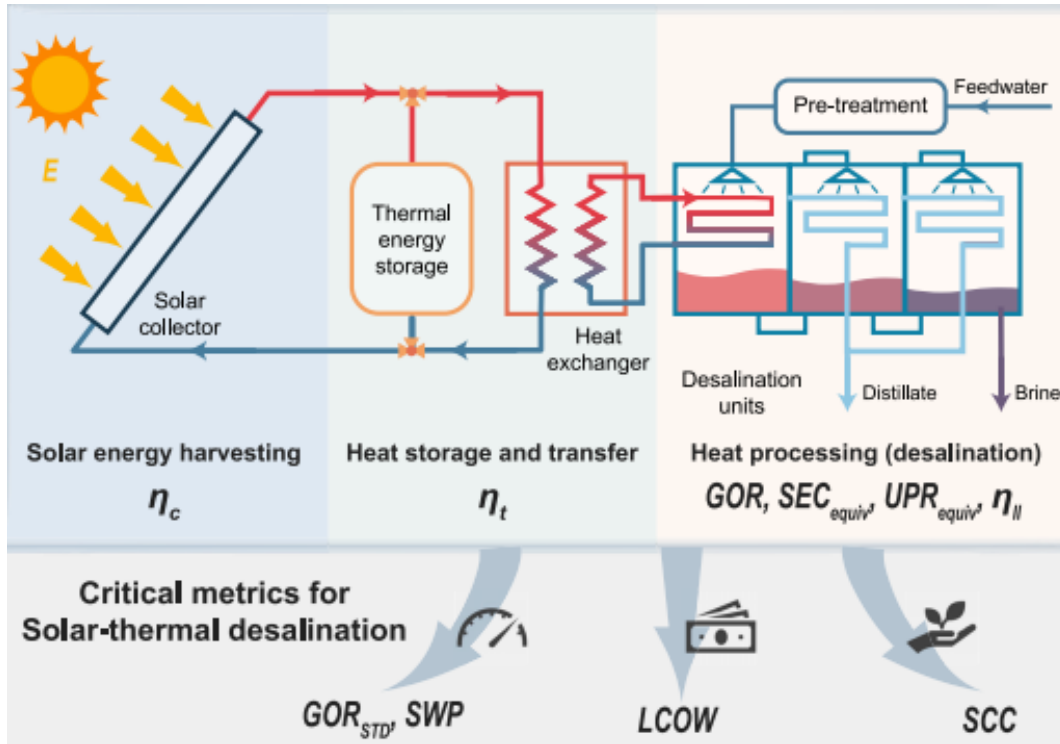


Figure 1.18 Energy flow in Solar-Thermal MED desalination unit [53]

The authors of [54] found that multi-effect distillation is one of the most efficient methods for solar thermal desalination and has the potential to reduce the cost of desalination significantly. However, this method also requires a large amount of energy, making it less suitable for use in remote and off-grid locations. Hybrid systems, on the other hand, can provide a more cost-effective solution by combining solar energy with other forms of energy, and have the potential to improve the overall efficiency of the desalination process. In conclusion, the authors argue that solar thermal desalination has the potential to play a significant role in addressing the global water crisis, especially in coastal regions where water scarcity is a major issue. However, further research and development is needed to

address the challenges faced by the technology and to improve its efficiency, cost-effectiveness, and sustainability.

1.4.3. Solar Thermal Energy for Membrane Processes

Solar Membrane Distillation (SMD) is a desalination technology that uses solar energy as the primary source of thermal energy to heat the feed water to the operating conditions (usually between 50 to 90 °C) [25]. This aims to convert saline or brackish water into potable water, especially in rural areas with limited access to conventional power sources [55].

The solar component of SMD is critical to its success, as it provides the energy required to heat the feedwater and drive the desalination process. A solar collector, such as a flat-plate collector or a parabolic trough etc. [56], is used to capture solar radiation and convert it into heat energy. This heat energy is then used to raise the feed temperature allowing the vapor to cross the membrane.

Solar collectors are designed to capture as much solar radiation as possible, while minimizing heat losses. Flat-plate collectors are commonly used in SMD systems due to their simplicity and low cost, while concentrated technologies such as parabolic troughs offer higher efficiency and more supplied heat [56]. In either case, the collector is placed in a location with high solar radiation, such as on a roof or in an open area and is typically oriented towards the sun to maximize exposure.

The solar collector is connected to a heat exchanger, where the heat transfer fluid (HTF) is heated by the collector. The HTF is then circulated through the heat exchanger, heating the feedwater in the process. The feedwater is then passed through a membrane distillation

module, where the purified water vapor is separated from the salt and other impurities. Moreover, solar PV panels could be used to power the circulating pump, however, the power consumed from the electrical pump could be neglected compared to the large amount of heat [57]. This process is completely shown in Figure 1.19.

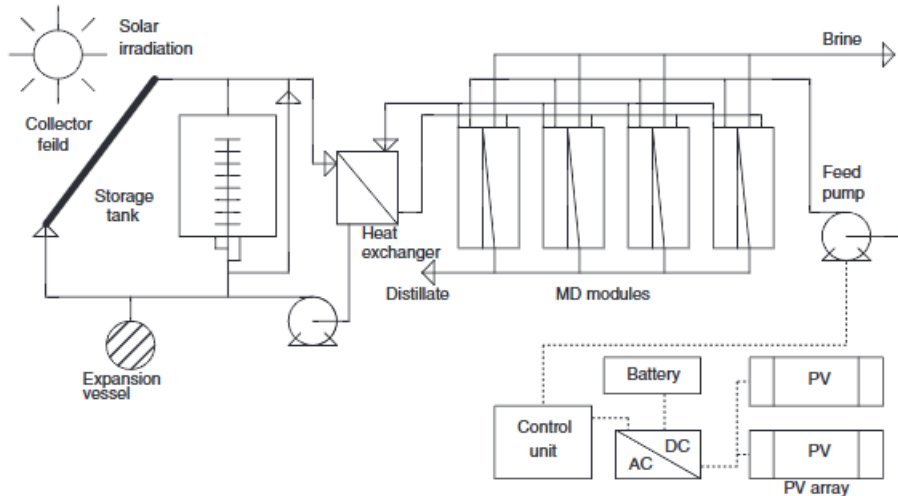


Figure 1.19 A complete solar-powered MD unit configuration [56]

The use of solar energy in SMD offers several advantages over other desalination technologies. Firstly, it is a renewable energy source that is widely available and has low operating costs. Secondly, solar energy is a clean and environmentally friendly energy source, reducing the carbon footprint of the desalination process. Finally, solar energy can be used in remote or underdeveloped areas, where access to conventional energy sources is limited.

In conclusion, the solar component of Solar Membrane Distillation is a critical aspect of the desalination process. By using solar energy as the main energy source, SMD offers a sustainable, energy-efficient, and cost-effective solution to water scarcity [58]. The proper

selection and implementation of solar collectors and heat exchangers may further improve the efficiency and performance to reduce the production cost of SMD systems, making them accessible to more people in need.

1.5. Artificial Neural Networks and Optimization Techniques

In recent years, there has been an increasing focus on the optimization of membrane distillation configurations to improve their performance for reduced water production cost. Several studies have explored the use of artificial neural networks (ANN) to model and optimize the membrane distillation process. ANN is a machine learning technique were developed in the early 1980s [59] that can be used to analyze complex systems, solve complicated multiple regression issues and make accurate predictions [60], making it an ideal tool for optimizing the MD process. Artificial Neural Network (ANN) models, including the Multi-layer Perceptron (MLP), solves linear and non-linear regression problems by effectively correlating and examining input-output variables, enabling quick and efficient creation of robust process designs [61] [62]. The MLP architecture consists of three layers: the input layer, hidden layers, and output layer. The ANN could be employed, to predict MD system performance for the purpose of optimization using different techniques.

Optimization techniques, including mathematical programming approaches like the Monte Carlo method [63], Particle Swarm Optimization (PSO) [64], and response surface methodology (RSM) [64] [65], as well as heuristic algorithms [66] [67] and metaheuristics like Ant Colony Optimization (ACO) [68] [69], can be used to optimize the performance of different MD configurations. In this study, Differential Evolution (DE) optimization is used, which is a stochastic and population-based metaheuristic algorithm used for

optimizing complex, nonlinear, and multi-modal problems. DE works by evolving a population of candidate solutions through a process of mutation, crossover, and selection, and it is particularly effective in handling high-dimensional and noisy search spaces. Compared to the other optimization techniques mentioned, DE is known for its fast convergence, robustness, and ease of implementation, making it a popular choice for various applications in engineering, and science [70]. The working principle of the DE optimization is presented in Figure 1.20.

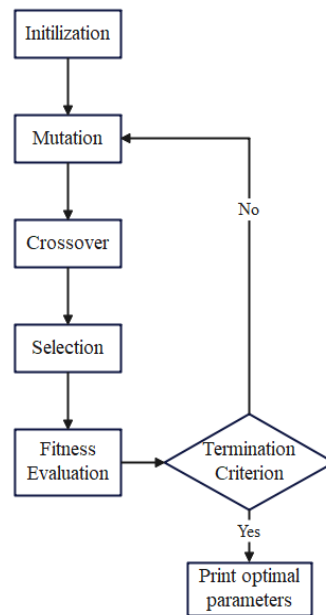


Figure 1.20 Flowchart of the DE optimization process

1.6. Research Gaps

In the past few years, membrane distillation (MD) technology has emerged as a promising alternative for water desalination and purification. Despite extensive research and development in this area, there remain several research gaps that require further exploration to optimize the performance and cost-effectiveness of MD systems. While most research has been performed on conventional MD configuration, only 0.45 % of research on MD is

based on PGMD [71], Figure 1.21, and very few of them are used to circulate the permeate water gap to enhance the heat and mass characteristics. The water gap membrane distillation with gap permeate being circulated was investigated and patented by Khalifa [72] [73], who only provided an comprehensive experimental analysis on a single stage unit .

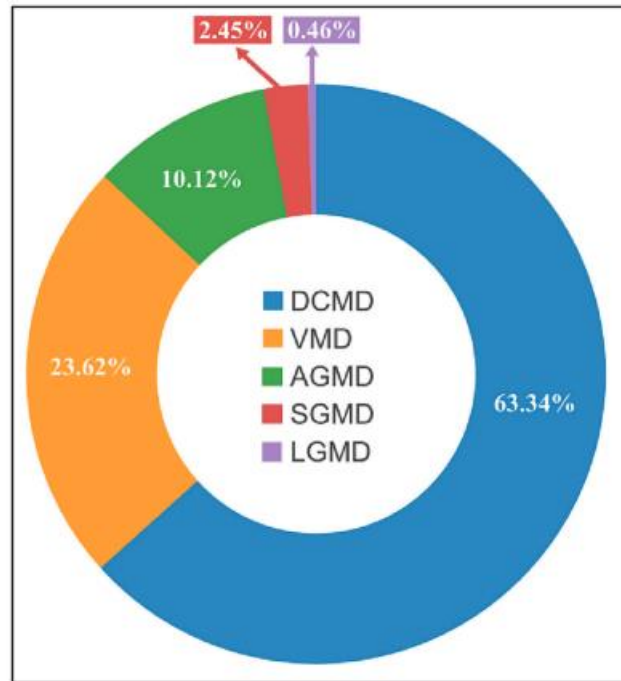


Figure 1.21 Percentage distribution of the investigation of conventional/modified MD configurations [71]

This dissertation aims to address these gaps by investigating three specific aspects of MD systems: the performance of a new permeate gap membrane distillation with water permeate gap circulation (PGMD-CG) unit, the optimization of circulating permeate gap membrane distillation (PGMD-CG) system using artificial neural networks (ANN) and differential evolution (DE) algorithms, and the development of a solar-powered circulating PGMD (PGMD-GC) system with different designs and operating parameters.

While previous studies have explored various configurations of MD systems, the performance of a new WGMD unit with circulating freshwater in the gap has not been theoretically evaluated. This gap in the literature hinders the understanding of the potential improvements in mass and heat transport coefficients, condensation rate, and freshwater production rate that could be achieved by incorporating circulation in the gap. To address this gap, this dissertation will develop a comprehensive theoretical model of the proposed PGMD unit and conduct a parametric study to evaluate the effects of various operating parameters on system performance.

Although the application of ANN in predicting the performance of various desalination technologies, including MD processes, has been widely explored, relatively few studies have utilized ANN for optimizing the operating conditions of these processes to maximize production and minimize energy consumption and production costs. Moreover, the integration of ANN with DE algorithms for optimization in MD systems, particularly in the context of PGMD-CG, has not been adequately addressed in the existing literature. This dissertation aims to fill this gap by developing an integrated ANN and DE optimization model to minimize the production cost of a PGMD-CG system, contributing to the design and operation of more efficient and cost-effective MD systems.

While the solar-powered MD systems have been investigated in previous studies, there is a lack of research on the development of a dynamic model to evaluate the performance of a solar-powered circulating PGMD (PGMD-GC) system with different designs and operating parameters. This research gap limits the potential for designing and implementing innovative solar-powered MD systems that can address practical challenges in water-scarce areas with high solar radiation. To fill this gap, this dissertation will

develop a novel dynamic model for a solar-powered PGMD-GC system combined with a flat plate collector (FPC), considering energy, heat, and mass balances, as well as water production costs using CAPEX and OPEX models. This study will also investigate the effects of permeate water gap circulation and the number of solar collectors on energy consumption, aiming to design a small-scale pilot system with reduced water production costs and the possibility of hybrid solar-electric powering for water supply in rural areas.

In addition to the previously identified research gaps, this dissertation will address the crucial gap concerning heat recovery in circulating water gap membrane distillation (CWGMD) systems. Despite the increased flux achieved by CWGMD processes compared to other MD configurations, high thermal energy consumption remains a significant challenge that hampers the large-scale implementation and economic feasibility of these systems in water desalination and purification applications. To tackle this research gap, the dissertation will explore various heat recovery configurations, with a focus on enhancing heat recovery using the circulating water permeate gap process. This approach aims to improve the energy efficiency, represented by specific thermal energy consumption (STEC) and gained output ratio (GOR), and overall performance of the CWGMD system by capturing and reusing waste heat generated during the distillation process.

By addressing these research gaps, this dissertation will contribute to advancing the field of MD technology and facilitate the development of more efficient, sustainable, and cost-effective solutions for water desalination and purification.

1.7. Aims and Objectives

This work aims to study and optimize the operating conditions of a solar-powered circulating multi-stage water desalination unit to reduce the overall production cost. To achieve this goal, the study proposes the following objectives:

1. Develop detailed theoretical models for the water gap membrane distillation process with gap water circulation, solar thermal Flat Plate Collector (FPC), and the integrated MD-Solar model.
2. Conduct a comprehensive study on the performance of the system, including output flux, energy efficiency, and cost estimation, at different operating conditions.
3. Implement an ANN and a DE optimization technique to optimize the operating parameter of a multi-stage system for the lower production cost as an objective function. The operating parameters are the feed flow rate and temperature, coolant flow rate, assuming an ambient water cooling, and the gap circulation rate.
4. Investigate possible approaches for energy recovery in the multi-stage desalination system to reduce the specific energy consumption (STEC) and increase the gained output ratio (GOR).
5. Design, manufacture, and experimentally test a new circulating water gap MD module at different operating conditions to evaluate the performance of the proposed system.

Overall, this study aims to contribute to the development of more energy-efficient and cost-effective solar-powered water desalination systems by combining theoretical and experimental studies to optimize the system performance.

1.8. Research Methodology

The proposed work is to be accomplished as per the following stages:

a) Background and literature review:

- Build a background on the importance of freshwater water supply to overcome the current scarcity.
- Study the current MD desalination techniques and energy requirements.
- Review of multistage designs and performances of membrane distillation systems.
- Perform a literature review on solar energy utilization in water desalination.
- Study the solar MD systems available in literature.
- Perform a literature review on the optimization techniques used in MD water desalination systems.

b) Theoretical work:

The theoretical work will be divided into the following main tasks.

Task 1: To develop a mathematical model of membrane distillation (MD) process with circulating water gap for water desalination, with validation. The mathematical model compromise mass and heat transfer equations that need to be solved simultaneously; this could be achieved through EES software. EES contains built-in thermodynamics libraries for accurate air and water properties.

Task 2: To develop an accurate theoretical model for flat plate solar collector, that could perform the gained solar radiation hour by hour solving heat transfer equation using EES. This model could predict the water temperature inside a storage a tank

for each hour. Validated through published experimental and theoretical works is to be considered.

Task 3: To integrate the MD model with the solar system model to determine the system performance, such as, permeate flux, productivity, the system efficiency, and the energy analysis.

Task 4: To perform a cost analysis of the multistage desalination process based on CAPEX and OPEX economical models.

Task 5: To optimize the design and operating conditions of the proposed MD System by studying and applying optimization techniques to maximize the productivity and/or minimize the energy consumption or cost.

c) Experimental work:

The experimental work will be divided into the following main tasks:

Task 1: To experimentally test the system performance parameters such as flux, productivity, STEC, GOR for the circulating water gap unit. Experiments will be conducted at different operating parameters such as feed temperature, feed flow rate, and gap flow rate.

Task 2: To Develop and manufacture an innovative design of a compact multi-stage circulating water gap MD unit. The compact design is to be made of low thermal conductivity materials and easy to manufacture.

1.9. Thesis Outline

The thesis begins with an Introduction, chapter 1, which provides the background of water scarcity and desalination, followed by an overview of water desalination

techniques. Solar thermal technologies and their classifications are discussed, along with the applications of solar energy in water desalination. This section also introduces artificial neural networks and optimization techniques, the aims and objectives of the study, research methodology overview, and an outline of the thesis.

In the Literature Review, in chapter2, conventional and modified membrane distillation configurations are examined, as well as multi-stage membrane distillation systems. Sustainable energy-driven membrane distillation and optimization methods in membrane distillation processes are discussed. The review also covers solar-powered membrane distillation studies, heat recovery configurations in membrane distillation, and identification of research gaps.

In chapter 3, the theoretical models are presented. This chapter introduces the theoretical model, modeling of membrane distillation processes, and energy efficiency assessment in membrane distillation. Solar thermal collector modeling integrated solar-membrane distillation modeling and energy efficiency, and cost estimation model development are discussed. This section also includes validation of models for circulating PGMD and FPC-storage tank.

The theoretical results are presented in chapter 4. Those results include the analyzes circulating water gap membrane distillation, the impact of operating parameters on permeate flux, and energy performance evaluation. Solar-powered membrane distillation system analysis, daily performance, energy analysis, and cost analysis are presented. This section concludes with sensitivity analysis and the design of a pilot MD system.

Experimental Setup & Analysis of the Multi-Stage System, in chapter 5, covers the description of the multi-stage circulating WGMD system, development of the experimental plan, and analysis of membrane characteristics. The description of experimental procedures, instrumentation and measurement details, and evaluation of experimental errors are also included.

Optimization of the Multi-Stage Circulating WGMD is presented in chapter 6. This chapter focuses on the development of the ANN model, analysis of variance in the ANN model, and application of the differential evolution optimization algorithm. Assessment of ANN performance and cost analysis and optimization of the system are discussed.

In chapter 7, the Assessment of Heat Recovery Configurations, the counter-flow multi-stage PGMD is presented, along with a description of the system and mathematical model. Results and discussion of the counter-flow multi-stage PGMD with and without gap circulation are provided, followed by an analysis of alternative designs and configurations, and assessment of increasing the area of PGMD.

Finally, in chapter 8, the Comparative Studies section includes a comparison of PGMD with and without gap water circulation, comparison of solar-powered multi-stage CPGMD with other solar-MD configurations, comparison of optimum water cost under different power conditions, comparison of heat recovery circulating PGMD with literature data, and a case study of multi-stage PGMD with and without heat recovery.

CHAPTER 2

LITERATURE REVIEW

The drawbacks of the two conventional desalination methods opened the door for researchers to develop other emerging desalination techniques, such as membrane distillation (MD), which is being developed for its tremendously advantageous and attractive features compared to the previously stated conventional techniques [74]. MD is a hybrid thermal-membrane emerging desalination technique. As a common process in different MD configurations, hot feed water (50–90 °C) is passed over a microporous hydrophobic membrane that allows only vapor to cross and condense on the cold side as a freshwater product. The vapor crosses through the membrane pores because of the vapor pressure difference created by the temperature difference between the hot and cold sides of the membrane. The way the permeated vapor condenses on the cold side of the membrane depends on the MD configuration.

The effectiveness of MD depends on several factors, including the MD configuration. This literature review focuses on various MD configurations, including multi-stage water gap membrane distillation, circulating water gap membrane distillation, sustainable energy-driven membrane distillation, and solar-powered membrane distillation. Each of these configurations has unique advantages and limitations and understanding their performance can help improve the overall effectiveness of MD technology. However, there is a research gap in this area, and this literature review aims to address this gap by providing a comprehensive analysis of the current state of knowledge on MD configurations.

2.1. Conventional Membrane Distillation Configurations

There are four main MD configurations, Direct Contact Membrane Distillation (DCMD), Air Gap Membrane Distillation (AGMD), Sweeping Gas Membrane Distillation (SGMD), and Vacuum Membrane Distillation (VMD) [25], as shown in Figure 2.1.

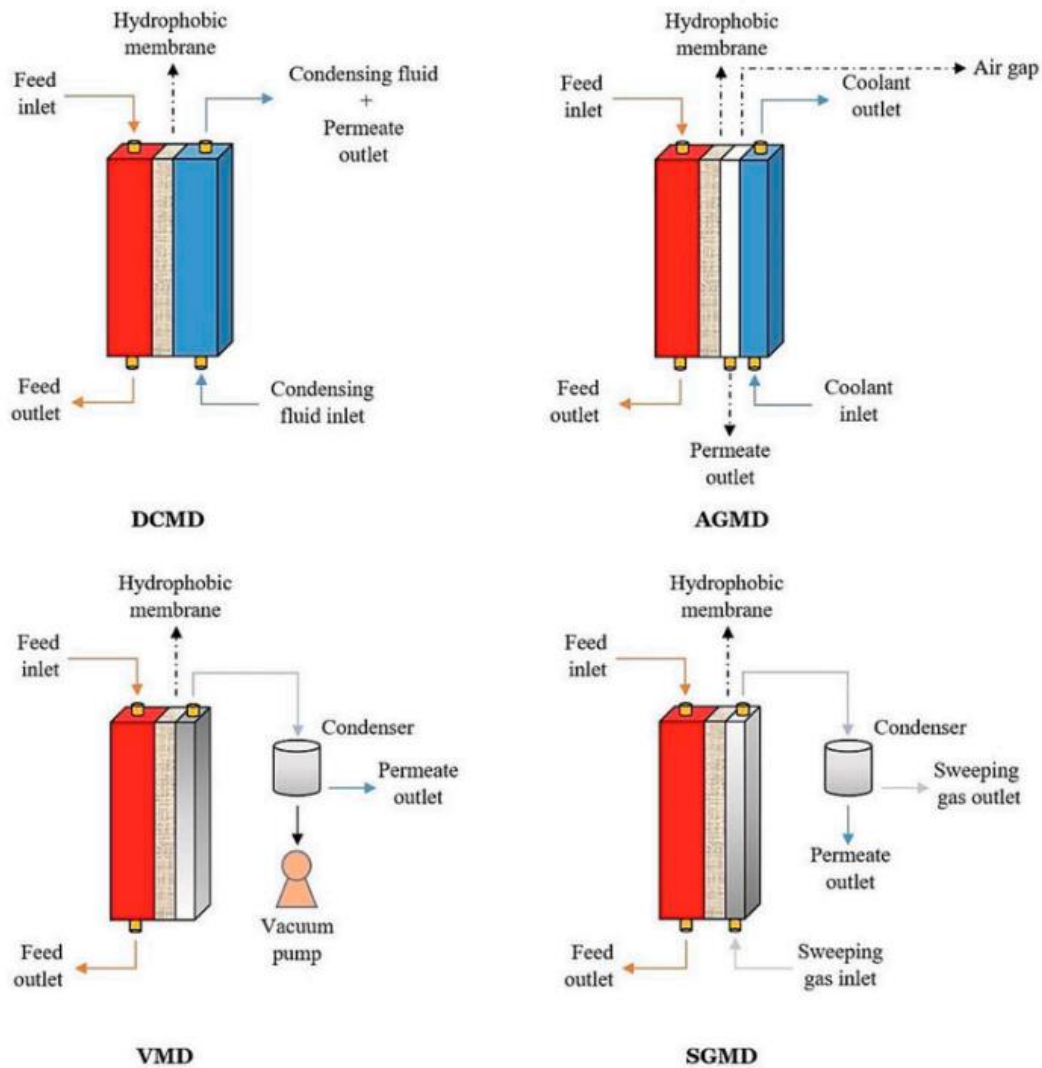


Figure 2.1 Typical MD configurations [75]

More than 60% of the Lab-scale studies are based on DCMD configuration, in which permeate vapor condenses directly on the cold side of the membrane, it is known for its

low energy efficiency due to the conduction losses through the membrane [76]. Khalifa et al. [77], have investigated theoretically and experimentally the performance of a DCMD, they found that the system had the best performance for a flux of 100 kg/hr.m^2 with operating conditions of 90°C feed temperature and 5°C coolant temperature. They also found that the thermal efficiency ranged between 70% to 95% with corresponding Gained Output Ratio (GOR) values between 0.8 and 1.2 for a feed temperature range between 40°C and 90°C respectively. For the same study, the most promising result was the ability of the system to deal with a high salt concentration of 100 g/l with a rejection factor of 98%. To reduce the conduction losses, the AGMD gap has been introduced [78]. In this configuration, water condenses in an air gap between the feed and coolant channels which reduces the conduction losses. However, it has a strong mass resistance to the vapor transfer across the gap before reaching the condenser surface [26], therefore it is recommended to optimize the air gap size. Kim et. Al. [79] have experimentally studied the performance of an AGMD with different conditions, such as the feed and coolant temperature, and the gap size. They found a maximum theoretical flux of 30 Kg/hr.m^2 . Experimentally, Pangarkar and Deshmukh [80] found a maximum total flux of 140 Kg/hr.m^2 for 4 stages of AGMD (35 Kg/hr.m^2 for each stage). A solution has been introduced to reduce the mass resistance of the stagnant AGMD, by sweeping an inert gas in the condensation channel which increases the mass flux and keeps a low conduction heat transfer, this technology is known as Sweeping Gas Membrane Distillation (SGMD). The main problem associated with this technology is that it requires the usage of an external condenser, which may lead to higher costs and lower condensation efficiency [76]. Similar to the SGMD, the VGMD extracts the vapor from the permeate side to an external

condenser by applying 5-10 KPa vacuum pressure [79]. The more the vacuum pressure the higher the permeate flux, but exceeding the applied pressure of the membrane Liquid Entry Pressure (LEP) could result in membrane wetting and performance deterioration [76].

2.2. Modified Membrane Distillation Configurations

Different improvements on the previous four typical MD configurations have been studied, water gap membrane distillation (WGMD), also known as Permeate Gap Membrane Distillation (PGMD), is one of these improvements, it has a similar configuration to the AGMD, but the gap is filled with stagnant permeate water. This decreases the resistance of vapor transfer through the membrane to the gap [81], compared to the AGMD, and consequently increases the flux of vapor crossing the membrane as well as the overall productivity. The water gap (WGMD) process is also referred to as liquid gap (LGMD) or permeate gap (PGMD) in literature.

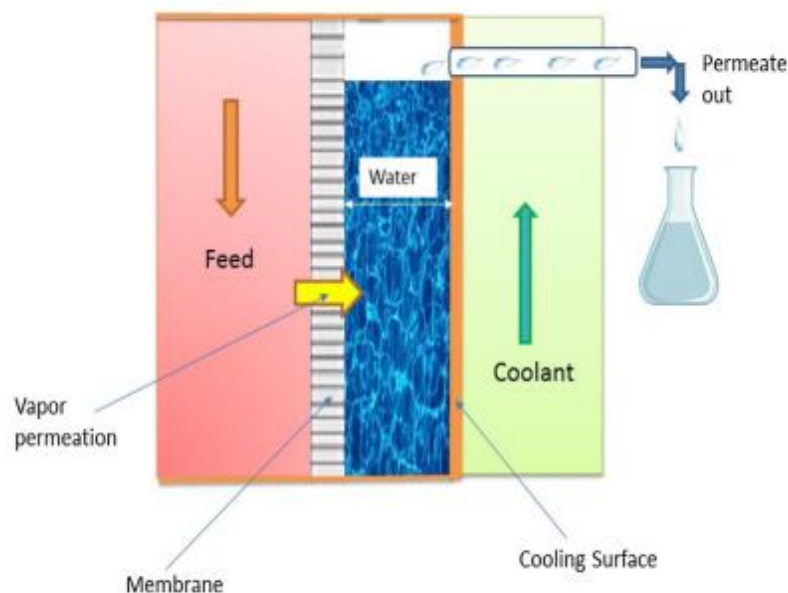


Figure 2.2 Typical Configuration of a Permeate Water Gap Membrane Distillation [81]

Abu-Zeid [82], experimentally compared the productivity, energy performance, and the GOR for both the air gap region (AGR) and permeate gap region (PGR) in hollow fiber MD setups. He found that the presence of the permeate gap showed a good enhancement in terms of water productivity. Khalifa [81] developed a module that can work as AGMD and WGMD to compare both performance parameters. He found that WGMD provided a 91% to 141% increase in the permeate flux compared to the AGMD for the same operating conditions because of the enhanced heat transfer and reduced vapor resistance across the membrane with the presence of water in the permeate side. In addition, he reported that the effect of gap thickness is negligible for the WGMD but effective in the AGMD owing to the thermal properties of water. On the other hand, despite the increase in productivity, the WGMD showed lower energy performance compared to the AGMD. Dahiru Lawal [83] experimentally claimed an enhancement in the flux and energy consumption for a single stage water gap MD system to achieve a maximum flux of 181 kg/h.m^2 with a 1129.27 kWh/m^3 energy consumption. Bartholomew et al. [84] used an optimization model to determine the water cost for three different gap material for the same single MD system, they found that water gap had a lower cost than air gap material because of the advanced heat transfer properties of water. Memon et al. [85] found that water gap MD presented 17-27% better performance compared to silica gel and zeolite filled gaps.

To enhance the flux in the typical WGMD, Khalifa [72] studied experimentally the effect of circulating the gap permeate water to increase the output flux by enhancing the heat transfer between the feed and permeate channels. The produced water quality is the same as the stagnant water gap with a TDS of 3-5 mg/L but an average increase in flux by 90%

for different operating conditions is observed due to the increased mass transfer as a result of the motion of gap water that increases the heat transfer by mass as well.

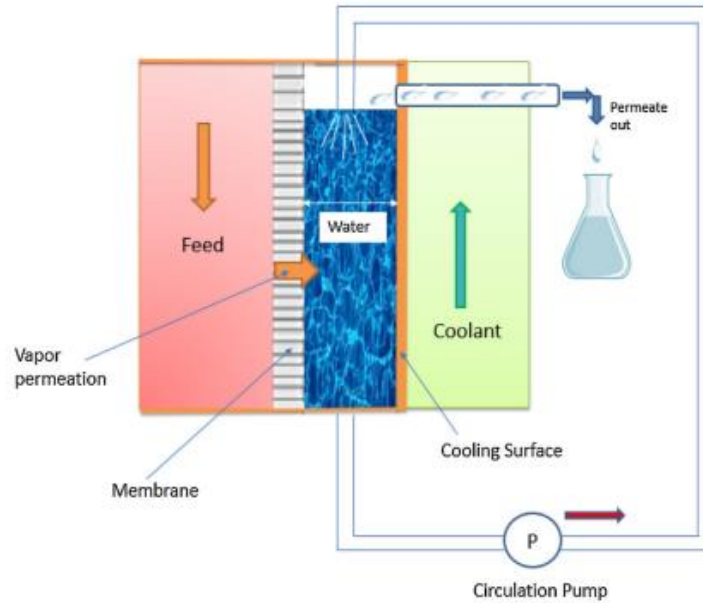


Figure 2.3 Permeate Water Gap Membrane Distillation with Gap Water Circulation Line [72]

For further flux enhancement, Khalifa in [86] studied experimentally the effect of cooling the circulating gap water in the cooling channel, despite the noticeable increase in flux among the stagnant and circulating water gap, a lower GOR has been obtained. This may be because of the conduction heat transfer from the feed channel to the coolant channel without applying a heat recovery technique.

2.3. Multi-Stage Membrane Distillation

The productivity of the MD process can be further improved using multi-stage modules, resulting in production cost reduction. Dudchenko et al. [87] found that multi-stage permeate gap MD provided a significantly lower cost than a single stage MD, they also claimed that multi-stage MD are cost-competitive solution for treating water with higher

salinities (greater than 125 g/L NaCl). Shamlou et al. [88] performed a study to optimize the performance of DCMD, AGMD, PGMD, CGMD, and VMD multistage MD systems, they found that increasing the number of stages enhances the energy performance, in terms of GOR and decreases the water production cost for the different configurations because of heat recovery effect. Sharan et al. [89] developed a multi-stage MD systems that could be integrated with supercritical water desalination system to develop a zero water discharge desalination system. Emad Ali [90] found that increasing the number of stages instead of increasing the stage number would provide a lower cost comparing to increasing the module size of a single stage. Dutta et al. [91] found that using multi-stage MD systems could reduce the energy consumption to 711 kWh/m³ rather than 1100 kWh/m³ for single stage DCMD when using a heat recovery.

Lawal and Khalifa [92] found that the productivity of the 2-stage air gap MD process is almost twice that of the one-stage unit. Geng et al. [93] showed that using highly concentrated RO brine as feed, a multi-stage air gap MD process can recover approximately 88% of the water. Pangarkara and Deshmukh [80] developed a theoretical model to assess the performance of a one-stage air gap MD unit, and then used an improved model to evaluate the performance of a four-stage air gap MD unit, which produced an output flux roughly 3.2 to 3.6 times that of the one-stage unit. Khalifa et al. [94] conducted an experimental investigation of multi-stage water gap and air gap MD processes, reporting that the output flux of the multi-stage water gap is higher than that of the multi-stage air gap system, with the 3-stage unit being about 2.6 to 3 times as productive as the one-stage unit.

2.4. Sustainable Energy Driven Membrane Distillation

Sustainable energy-driven membrane distillation (MD) is a water treatment process that uses renewable energy sources, geothermal, or waste heat as an energy source [95]. A sustainable process utilizes these sources to provide the necessary thermal energy to drive the MD process, instead of relying on fossil fuels. This process has a lower environmental impact compared to traditional desalination technologies because it does not require large amounts of electricity.

Geothermal MD has been widely investigated. Sarbatly and Chiam [96] evaluated energy use in cross-flow vacuum membrane distillation using lab-fabricated and commercial polyvinylidene fluoride (PVDF). When using warm geothermal water, 95% of the heating energy is saved, and the resulting drinking water quality is satisfactory. Moreover, they claimed that 20,000 m³/day VMD desalination plant operated with geothermal energy had lower water production costs (0.50 US\$/m³) compared to one without (1.22 US\$/m³). Three regions were studied for using geothermal energy in membrane distillation to obtain potable water and irrigation water: Cerro Prieto in Mexico, Momotombo in Nicaragua, and Lake Poopó in Bolivia. Membrane distillation was found to be effective in obtaining water of good quality with high retention of toxic solutes like arsenic, using geothermal energy or geothermal water as a heat source. Membrane distillation is suitable for use in areas with cheap sources of heat [97]. Unal et al. [98] examined boron removal from geothermal water using an air gap membrane distillation (AGMD) system. Six types of commercial membranes were tested, and the effect of operating parameters on permeate flux and rejection performance was investigated. Optimization experiments were performed using three saline water concentrations, and a real geothermal water was used to determine

AGMD system performance. All membranes achieved >99.5% boron removal efficiency, with permeate water boron concentrations <0.5 mg/L, meeting irrigation standards. The dominant factors affecting permeate flux were air gap width, feed velocity, and temperature, and feed water boron concentration did not affect permeate flux.

Waste heat powered MD are considered also a sustainable way to produce fresh water in areas where the resources are scarce. Baghbanzadeh et al. [99] explored small-scale desalination devices powered by waste heat from electric power generators to produce freshwater in areas affected by water stress. The technology relies on a passive, thermally-driven membrane distillation device powered by low-grade waste heat from small diesel engines. Field experiments show nearly 2.61 L/m²/hr freshwater production from seawater, and the proposed solution could be effective for field hospitals in remote or impoverished areas. Waste heat was used [100] to power a direct contact membrane distillation (DCMD) system for water recovery from saline waste. The pilot plant achieved 92.8% water recovery, producing high-quality distillate with 99.9% dissolved solids rejection. Membrane fouling only affected flux and thermal energy demand at the end of the trial. The available energy for a continuously operating 500 MW power station estimated a treatment potential of up to 8000 m³/day, making it practical for industrial, residential or agricultural water supply.

Solar powered MD is another sustainable desalination technique that is discussed in the following sub-section.

2.5. Optimization of MD Processes

Alawad et al. developed [101] a theoretical framework for enhancing the performance of permeate-gap membrane distillation. They used a heuristic differential evolution (DE) technique to optimize operational parameters such as feed and coolant temperatures, flow rates, and permeate gap thickness to increase permeate flux and decrease production costs and energy consumption. The optimization was performed for both single- and multi-objective functions, resulting in a 60% increase in the highest permeate flux value and a 38% and 31% decrease in production cost and energy consumption, respectively. The multi-objective model achieved a 51% higher optimized permeate flux and a specific energy consumption of 876 kWh/m³. Khalifa et al. [68] proposed heuristic optimization techniques based on a mathematical model that forecasts permeate flux using heat and mass transfer analysis coupled with two algorithms, ACO and PSO. The authors determined that PSO yielded superior results and was less computationally demanding than ACO, with less than a 3% deviation between the optimal flux values derived from both methods. RSM, a statistical modeling technique, was used to optimize a process or system with multiple input variables. Multiple studies have been conducted [102] [103] to optimize the performance of AGMD systems for water desalination. The impact of operating and design variables, including feed temperature, flow rate, coolant temperature, and air gap width, on AGMD performance, was examined. The Taguchi method and RSM were employed to optimize the AGMD system and generate regression models with permeate production as the primary focus. The results showed that the feed temperature and air gap width were the most significant variables affecting the AGMD performance. The optimal flux values ranged from 76.0457 to 76.998 kg/m².hr.

ANN models, including the Multi-layer perceptron (MLP), were developed in the early 1980s [59] to solve linear and non-linear regression problems by effectively correlating and examining input-output variables, enabling the quick and efficient creation of robust process designs [61] [62]. The MLP architecture comprises three layers: an input layer, a hidden layer, and an output layer. Different studies have implemented ANN to predict the performance of membrane distillation systems. However, limited work has been conducted on optimizing membrane distillation using ANN. Dragoi and Vasseghian [104] modeled mass transfer in a vacuum membrane distillation process to treat radioactive wastewater. The results showed that a feed-forward MLP-ANN with 10 neurons in the hidden layer, a fitting coefficient of 0.975, and the highest root mean squared error of 1.83% predicts the permeate flux with a satisfactory level of accuracy. Mittal et al. [105] studied the impact of the operating parameters on membrane fouling during vacuum membrane distillation using an ANN. An ANN with ten nodes was created using MATLAB's deep-learning toolbox to achieve a minimum mean square error value of 0.58. A correlation coefficient greater than 0.98 successfully confirmed the model predictions. Acevedo et al. [106] used ANN to improve a membrane distillation unit's distillate predictions in a trigenerational system. The performance of the ANN model was investigated and enhanced using a parametric analysis that varied the number of neurons in the hidden layer, the number of experimental datasets, and the use of various activation functions. Idris et al. [107] used an ANN to forecast the water flux from the DCMD of pretreated palm oil mill effluent. A feed-forward MLP-ANN model was created using 3 input data (feed velocity and feed temperature) and one set of output data (permeate flux) with 18 experimental data points. This model has one hidden layer and 13 hidden neurons. Their findings demonstrated that

feed temperature was the most sensitive parameter, with the lowest mean square error of 0.0034, and the training, validation, and testing regression coefficient (R) values were 0.9859, 0.9986, and 0.9984, respectively. Dudchenko and Mauter [108] estimated the physical parameters of membrane distillation, membrane permeability, heat transfer rate, and thermal conductivity using MLP algorithms and found excellent agreement with experimental data.

Khayet and Cojocaru [109] predicted air-gap membrane distillation performance using an ANN with 72 data points. They found an optimal permeate flux of 51.075 l/m².hr using the Monte Carlo stochastic optimization method for the given input variables, including the feed and coolant temperatures, flow rates, and air gap thickness. The same authors conducted a similar study for sweeping gas membrane distillation [110], where they employed an ANN model for desalination with 41 trials for training, six trials for validation, and six trials for testing. The airflow rate, feed flow rate, and feed inlet temperature were optimal, with a maximum flux of 8 L/m².hr. Shirazian and Alibabaei [64] developed a mathematical model using Volterra functional series theory to evaluate the impact of input parameters, such as cold feed inlet temperature, hot feed inlet temperature, and flow rate, on output parameters, such as distillate flux, cold feed outlet temperature, and gained output ratio, in an airgap membrane distillation system. The model was optimized using a PSO-controlled neural network and multivariable function approximation and was experimentally validated. Gil et al. [111] used RSM and ANN to predict a commercial-scale membrane distillation system performance for desalinating RO system brines. The results demonstrate that the ANN model can more accurately predict the behavior of the module for the entire range of input variables. Consequently, the

module operating at feed-water temperature and salinity of 70 °C and 105 g/L (concentrations that can be attained while desalting the RO brines) respectively was given the best working conditions using an ANN model. Kalla et al. [112] used the Taguchi optimization technique and an ANN to investigate the separation performance of an HCl/water mixture using AGMD. They developed an ANN model that effectively matched the experimental data, with a fitting coefficient R^2 of 0.998. Cao et al. [113] used an ANN to model and simulate a VMD desalination system and showed that the correlation coefficient, which measures the percentage match between experimental data and ANN predictions, was greater than 0.994. The optimal operating conditions for the VMD process were identified using a performance analysis of the ANN model with a maximum permeate flux of 14.66 L/m².hr and an acceptable coefficient of variation (CV) value based on the experiment. Mirbagheri et al. [114] used an ANN genetic method with simultaneous down- and up-aeration in a membrane bioreactor to examine and predict membrane fouling using MLP and radial basis functions (RBF). The RBF-ANN and MLP-ANN both had successful training models. However, the MLP-ANN-optimized ANN prediction had the best performance.

2.6. Solar Powered Membrane Distillation

Khalifa et al. [115] experimentally studied the performance of three-stages DCMD, they found that parallel MD connections provided higher production flux compared to series connection, they found that one solar collector provided 9 L/Day of freshwater product. Lee et al. [116] performed a mathematical model to investigate the monthly and hourly performance of a dynamic solar powered DCMD with an energy recovery scheme, they dynamically varied the number of stages in use based on the provided feed temperature to

increase the water productivity, they found only a slight increase in the water production when they applied the dynamic operation with a percentage of 7%. However, a 31% to 45 % increase in the thermal efficiency has been recorded. Li and Lu [117] modeled fully solar-powered stand-alone sweeping gas membrane distillation desalination system for island and coastal households. In this study sun was the only power source, they used solar thermal collector to provide thermal energy and a solar photovoltaic array to supply direct current to the operating pumps. The system provided a daily water production of 9.98 kg/day in January and 23.26 kg/day in July with an estimated cost of 18.34 US\$/m³. Sandid et al. [118] studied experimentally and theoretically the multi-stage AGMD process using flat plate and evacuated tube collectors. They claimed that the permeate flux with the evacuated tube collectors was 18.81%–30.44% higher than flat plate collectors with a maximum GOR of 4.4. Their proposed system provided 28.78 m³ /year of fresh drinking water at a cost of 14.73 USD/m³.

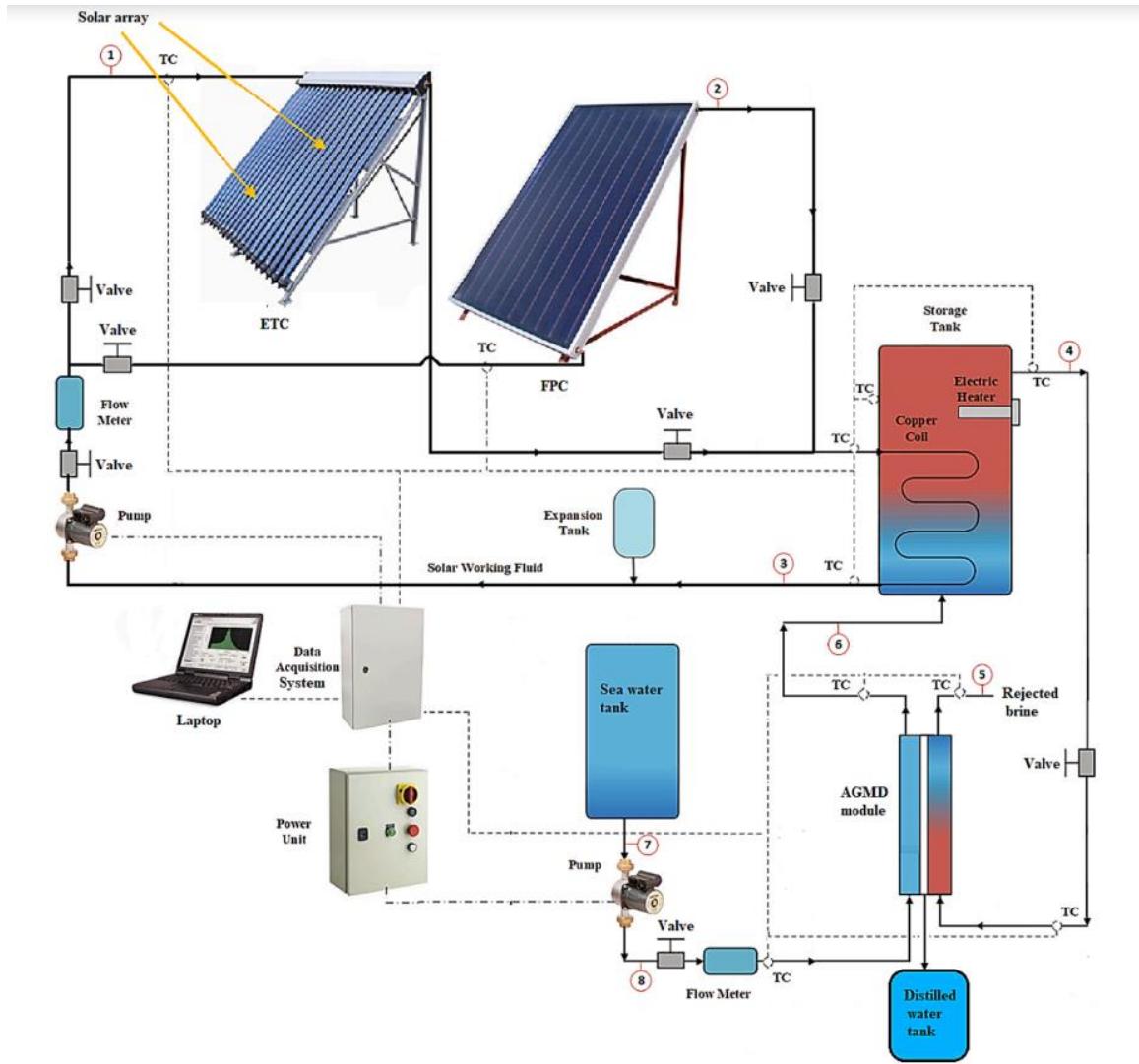


Figure 2.4 Schematic of Solar Powered Spiral Wound AGMD [118]

Both FPC [30] and Evacuated Tubes Solar Collectors ETSC [32] could be used in the indirect solar thermal desalination, they could provide heated water up to 100 °C [29] conforming with the heating temperature of the feed solution as stated in the previous literature for different MD configurations. Both collectors absorb the sun's heat through a heat transfer fluid, which in turn could be used directly as feed water to the MD process or indirectly through a heat exchanger [119][120].

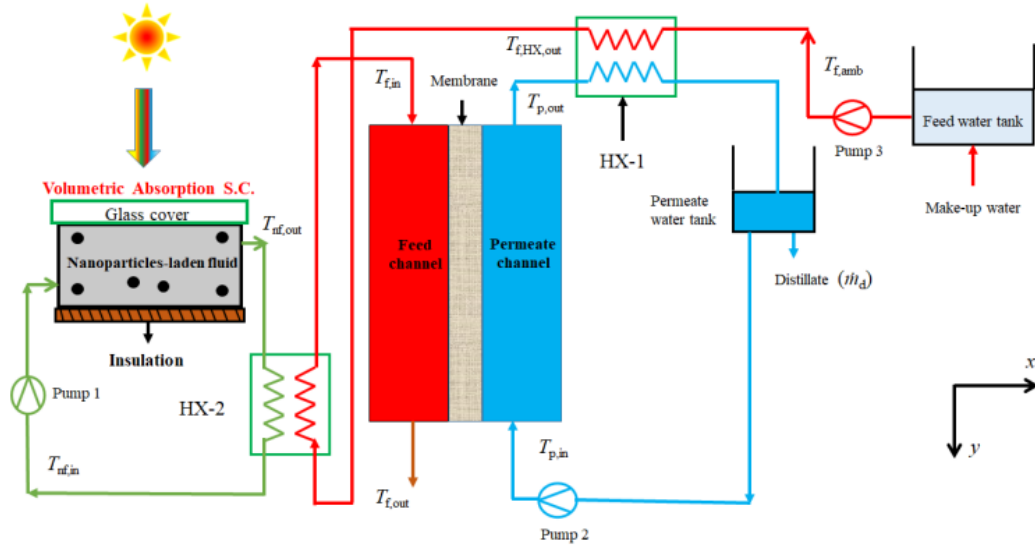


Figure 2.5 Solar Powered DCMD [120]

Even though ETSC could perform more efficiently and especially in cold climate conditions [32], Al Suleiman et. Al [121] found that FPC is providing more yearly savings and less payback period than ETSC for the same heating application for different cities in the KSA, which is considered a hot region. Solar thermal technologies have been integrated with different MD configurations.

Theoretically, using numerical analysis for the distillation process and TRNSYS software for the integration with the solar collector, a 22.6 m^2 of FPC has been integrated with a 7.2 m^2 spiral wound membrane in direct contact configuration to produce a flux of $19.7 \text{ L/m}^2/\text{d}$ [122]. A recent study in China [123], on the current and future perspectives of the distributed solar membrane distillation, found that a solar intensity of 10 kWh/m^2 distributed over 10 hours could provide 10-15 L/day for a collecting surface area of 5 m^2 as a rough estimation for a solar thermal collector having a thermal efficiency of 50% and

using the thermodynamics limits of evaporation. A theoretical study [124] found that an ETC collecting area of 4 m^2 and a total hollow-fiber membrane area of 0.6 m^2 produce 10 to 23 L/day of freshwater from an SGMD.

Experimentally, A 10 m^2 of spiral wound AGMD has been integrated with 5.7 m^2 FPC to produce an average of $2.5 \text{ L/m}^2/\text{day}$ [125]. For a 20 m^2 of FPC collecting area, a 2.8 m^2 of Flat-sheet AGMD produced a flux of $6.5 \text{ L/m}^2/\text{hr}$, a 9 m^2 of Flat-sheet WGMD produced a flux of $3.5 \text{ L/m}^2/\text{hr}$, and 10 m^2 spiral wound WGMD produced a flux of $3.4 \text{ L/m}^2/\text{hr}$ [126]. A 500 m^2 of stationary solar collectors are used to produce $86 \text{ L/m}^2/\text{day}$ from a 2.8 m^2 of an AGMD membrane [119]. Banat et al. [127] studied the performance of a large solar-driven MD unit with internal heat recovery in Aqaba, Jordan. The experimental setup consists of two separate loops, the solar loop and the desalination loop connected by a heat exchanger. The solar loop consists of 72 m^2 of collecting area and 3 m^3 of heating storage tank whereas the desalination loop consists of an effective area of 10 m^2 of spiral wound membrane sheets. The results showed a productivity between 2 to 11 L/day for each square meter of solar collecting area with an average energy consumption between 200-300 kWh/m³. Without any heat recovery regime, a 0.25 m^2 of flat sheet membrane integrated with an ETSC of 2.16 m^2 could produce $4 \text{ L/m}^2/\text{hr}$ for a STEC of 750 kWh/m^2 [128]. Bouguecha et al., [129] used eight FPCs with an overall area of 20 m^2 in DCMD having a membrane area of 3.39 m^2 to obtain a maximum flux of $8.8 \text{ L/m}^2/\text{hr}$.

The techno-economic analysis for solar-powered membrane distillation (SPMD) has been investigated in different studies. Lopez et.al. [74] theoretically investigated in the techno-economic analysis of a 1000 L/day WGMD and AGMD pilot models powered by different heat sources scenarios, they found that water production costs between 1.56 and 7.53

€/m³ for WGMD and 2.38 to 9.60 €/m³ for AGMD. They claimed the most feasible scenarios were for solar and waste heat MD powered sources. Banat and Juwaid [130] experimentally found the estimated cost of production of potable water production from an SPMD in Jordan is between 15 to 18 US\$/m³ depending on the reliability of the desalination plant, they claimed that the unit price decreases as the membrane and plant's lifetime increases. For a fully solar-powered SGMD in Honk-Kong, Li& Lu [131] found theoretically the cost of water production to be 18.34 US\$/m³. They used a mathematical model solved using Fortran language, but they did not show any relation that calculates the heat loss coefficient from the solar thermal collector. Moore et al. [132] found an optimized cost of 85 US\$/m³ for solar-powered SGMD.

2.7. Heat Recovery Configurations

Few implementations of heat recovery configurations have been researched and investigated as a potential solution to mitigate the high energy consumption of membrane distillation (MD) systems in water desalination and purification applications. Najib et al. [133] investigated the effect of recovering heat from the permeate of a DCMD, they achieved a maximum water recovery ratio of 6.5% and a performance ratio of 0.3 kg/MJ, which increased to 1.15 kg/MJ with the activation of heat recovery. Lisboa et al. [134] conducted a theoretical analysis on the energy efficiency of a direct contact membrane distillation (DCMD) module with external heat recovery for desalination applications, as shown in Figure 2.6. They found a maximum flux of 6 kg/hr.m² with a GOR that can exceed a value of 4.0 for 100% effectiveness of energy recovery.

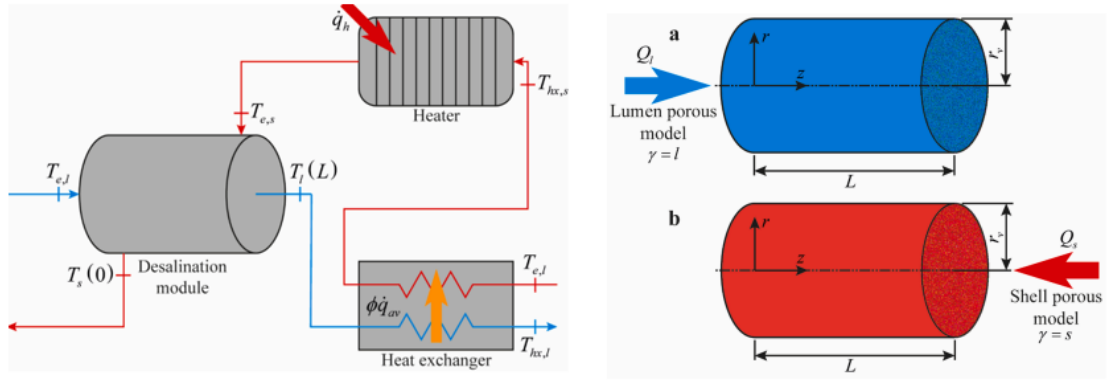


Figure 2.6 Counter-flow feed and permeate cooling in DCMD with external heat recovery [134]

Kim et al. [135] developed a multi-stage vacuum membrane distillation system, which incorporates a heat recovery unit (HRU) to recover energy from permeate vapor and transfer it to feed seawater. A commercial shell-and-tube capillary membrane module with polypropylene hydrophobic fibers is employed for system design. The system is examined with varying numbers of HRUs using a mathematical model. A 24-stage solar-assisted VMD system with 10 HRUs yields 3.37 m³/day of water production, a 34% increase compared to a single HRU system. For a non-solar VMD system, overall specific thermal energy consumption (OSTEC) drops by 20% as HRUs increase from 1 to 10.

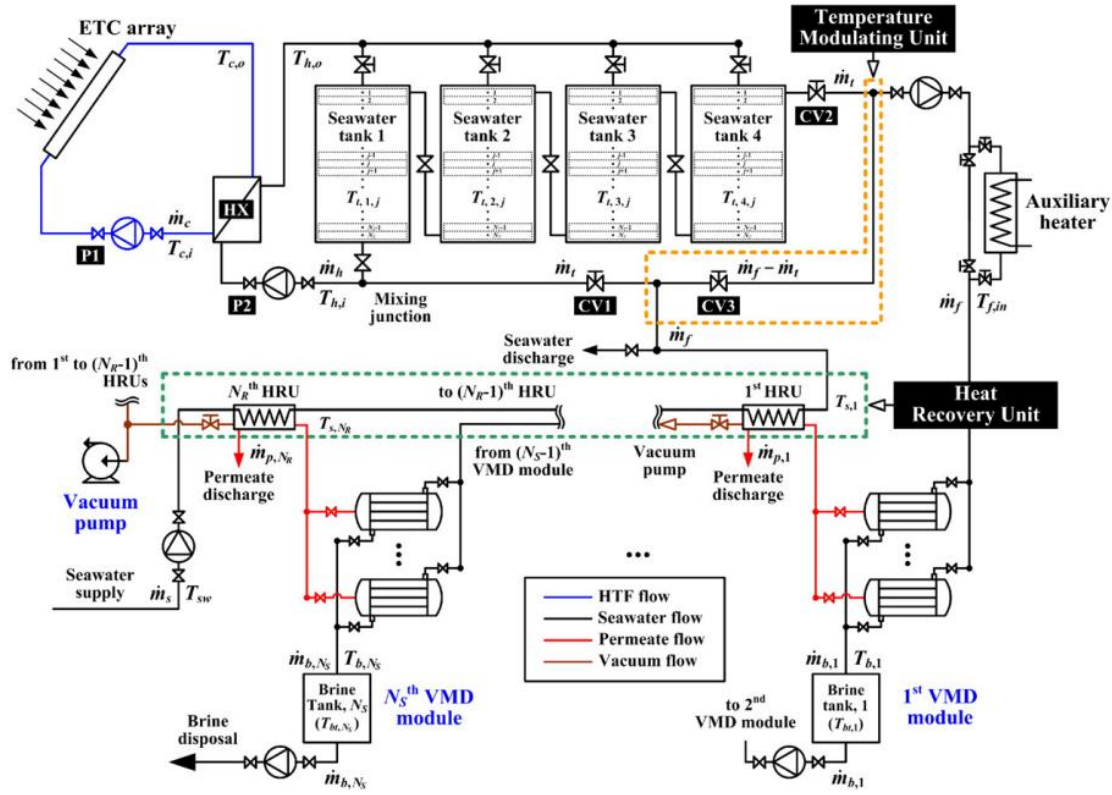


Figure 2.7 VMD with external heat recovery as described in [135]

Gandour et al. [136] presents an experimental analysis of a desalination system consisting of two air gap membrane distillation (AGMD) modules. A custom-built test rig was developed for this study, with the modules able to be connected in various configurations. The base test case (BTC) involves connecting the modules in parallel, ensuring identical operating conditions for feed and coolant flows in each module. The coolant flow condenses water vapor in the air gap before being discharged at a relatively high temperature, which can be harnessed to enhance the system's thermal efficiency and water productivity. To utilize this waste heat, two additional configurations were tested: double and single heat recovery (DHR and SHR). In DHR, the coolant flow exiting both modules are employed to heat the feed flow of the second module. In contrast, the SHR utilizes the

coolant flow from only the first module (at a relatively high temperature) to heat the feed flow of the second module. The system's performance was assessed using water productivity and Gain Output Ratio (GOR) under varying feed flow rates and salinity levels. The findings indicate that the DHR configuration improved system productivity by 27% compared to the BTC, while the SHR configuration increased productivity by 41%. Yao et al. [137] innovated a Continuous-Effect Membrane Distillation (CEMD) Process Based on Hollow Fiber AGMD Module with Internal Latent-Heat Recovery, as presented in Figure 2.8. Performance was measured by flux, performance ratio (same as GOR), and evaporation efficiency. The maximum performance ratio of 13.8 was achieved within the experimental range. A theoretical model based on transport equations was established to predict performance and aligned well with experimental data.

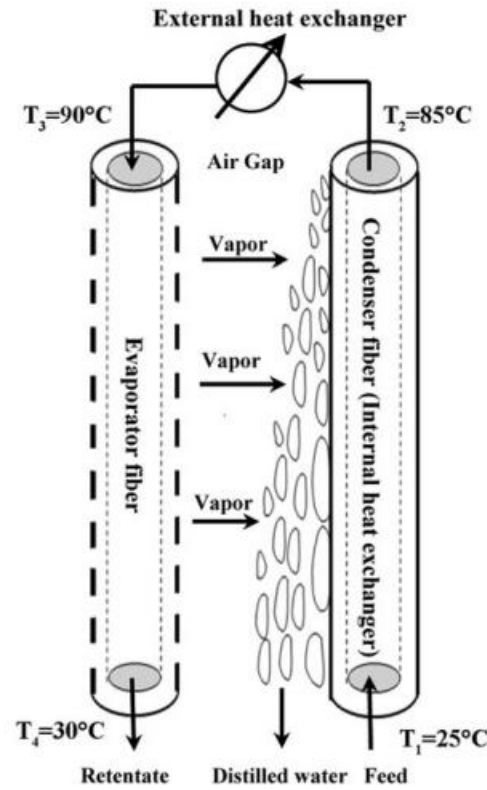


Figure 2.8 CMED configuration [137]

Burrieza et al. [138] examined the performance of two pre-commercial air gap membrane distillation (AGMD) desalination modules developed by Singapore-based Keppel Seghers, as shown in Figure 2.9. These modules were tested under real conditions for over 2400 hours across two years, coupled with a solar field using static collectors. One module features a compact single design, while the other comprises three stages connected in series, with both modules having a total membrane surface of 9 m^2 . The research focuses on characterizing the performance of the modules, including distillate production and quality, thermal efficiency, and recovery ratio, in relation to operational parameters. The study found minimum specific thermal energy consumption values of 1805 kWh/m^3 for the compact prototype and approximately 294 kWh/m^3 for the multi-stage module.

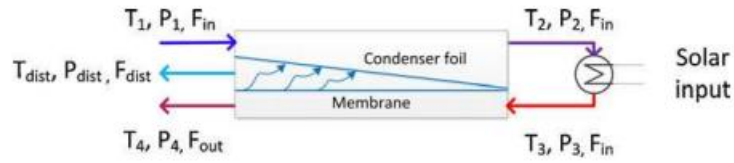


Figure 2.9 Scheme for the energy and mass balance. T, P and F stand for temperature, pressure and volumetric flow rate, respectively in [138]

Cheng et al. [139] investigated self-developed hollow fiber membrane modules with internal energy recovery, testing them in both AGMD and PGMD configurations, as shown in Figure 2.10. They found a better performance of the PGMD compared to the AGMD in terms of flux, GOR, and heat recovery with a maximum observed flux of 6.5 kg/hr.m^2 and GOR Value of 2.5 for the PGMD configuration.

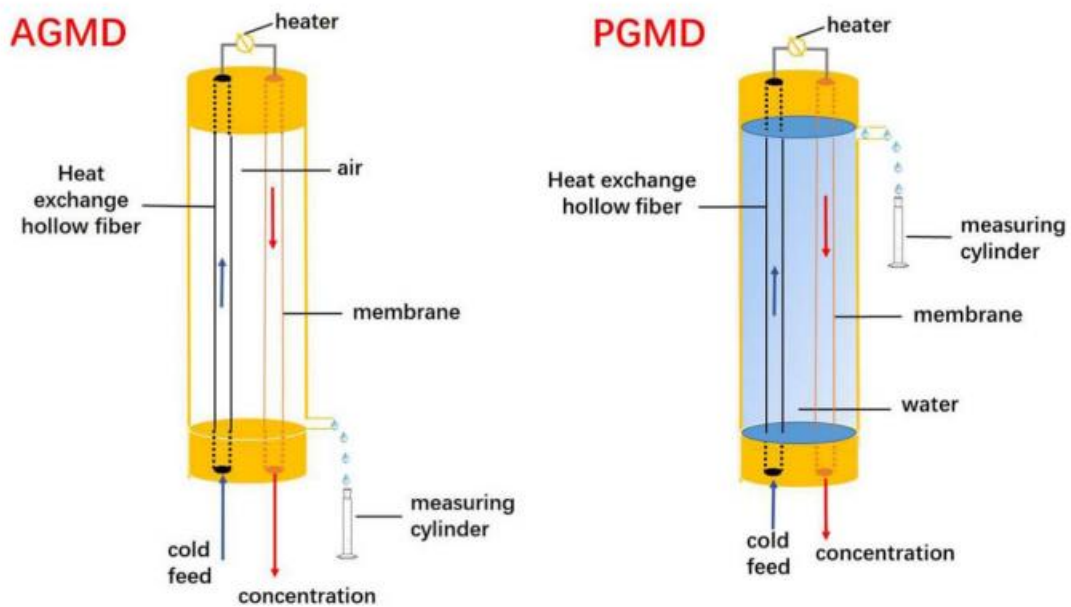


Figure 2.10 Schematic diagram of the counter-flow feed and coolant streams as for the AGMD and PGMD configurations used in [139]

CHAPTER 3

THEORETICAL MODEL

3.1. Introduction

This chapter presents the theoretical model that forms the basis of the research. It discusses various models that are critical to understanding the solar-powered water desalination process. These models include the modeling of the MD process, cost estimation model, optimization model of the MD process, solar thermal collector modeling, and integrated solar-MD modeling.

For the circulating WGMD sub-system, a one-dimensional mode of heat and mass transfer inside the unit is simulated to determine the resultant permeate flux and other performance parameters such as the gained output ratio (GOR) and the specific thermal energy consumption (STEC). The circulation of gap water provides forced convection in the water gap that increases the heat transfer coefficient between the hot and cold sides rather than pure conduction [23], this has the main effect to increase the vapor transfer through the membrane.

The cost estimation model uses the CAPEX/OPEX model to estimate the costs associated with the capital and operating expenditures required to build and run the desalination system. This cost estimation model provides valuable insights into the overall feasibility of using the solar-powered circulating water gap membrane distillation.

A dynamic heat transfer model is used to simulate the performance of the FPC connected to a hot water storage tank (HWT) in which the gained energy from the collector is stored efficiently in the tank and transferred by a heating coil to the feed water of the membrane unit through a hot water circulation pump. The model calculates the hourly energy performance of the solar thermal collector, providing a more accurate understanding of its performance over time.

An integrated solar-MD model is employed to bring together all of the models discussed and understand how they work together in the solar-powered water desalination process. This model correlates the hourly solar energy performance to hourly water production from the MD process. It provides a comprehensive understanding of the desalination process and aids in optimizing it for maximum efficiency and cost-effectiveness.

The experimental validation of the MD and solar models is conducted with a 5% deviation, providing confidence in the accuracy of the models and the validity of the research. By exploring these various models in depth, the chapter lays the groundwork for the subsequent performance analysis, which enables the drawing of conclusions regarding the feasibility of using the solar-powered water desalination process in real-world applications.

3.2. Modeling of MD Process

In this part, the mass and heat transfers inside the MD unit are simulated, based on a one-dimensional model, to determine the rate of distillate obtained, and to evaluate the energy efficiency of the WGMD process with the gap water circulation. The circulation of the gap water results in forced convection being the mode of heat transport within the gap rather than pure conduction or natural convection [23]. Heat is transported from the hot stream to

the cold stream within the MD unit in different forms. Figure 3.1 depicts the module's internal design and the expected distribution of the temperature across the module which is associated with mass and heat transfer modes throughout the WGMD unit with gap circulation. For clarity, Figure 3.2 represents a schematic of the WGMD cell with the feed, coolant, and gap circulation loops.

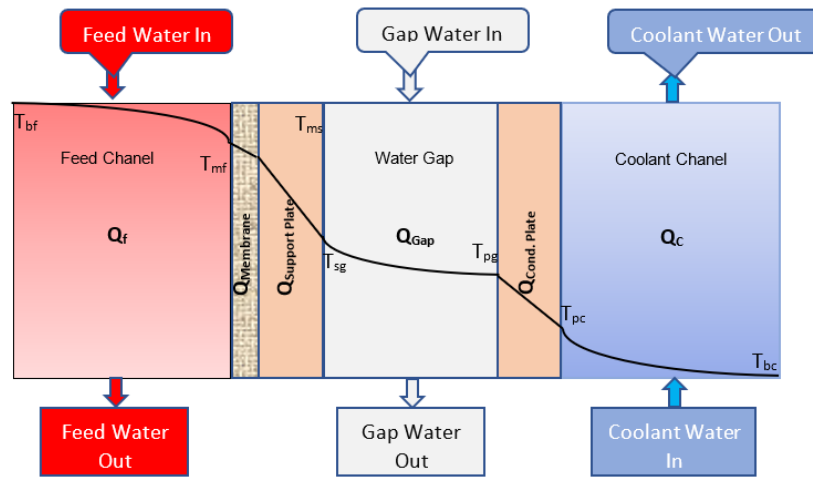


Figure 3.1 Temperature distribution across the different components of the MD-system

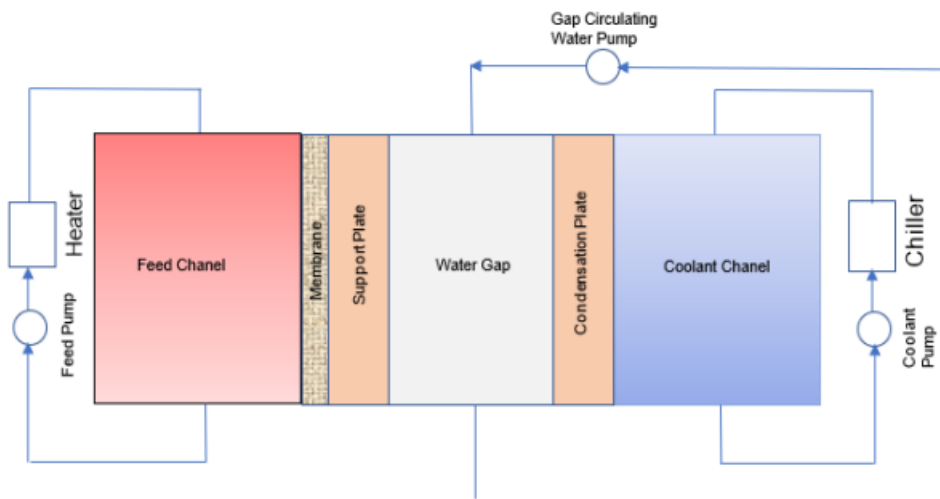


Figure 3.2 Schematic of the WGMD module with feed, coolant, and gap circulation loops

Feed solution flows over the membrane in the feed channel resulting in a convection heat transfer from the hot feed stream to the hot membrane surface, which can be expressed as Newton's law [140] as follows:

$$\dot{Q}_f = h_f * (T_{bf} - T_{mf}) * A_f \quad (1)$$

h_f is the convective heat transfer coefficient in the feed channel, T_{bf} is the temperature of the bulk feed water, T_{mf} is the membrane surface temperature on the hot feed stream side, and A_f is the fluid contact area between the feed channel and the membrane.

The heat Q_f represented in equation (1) is then transferred by conduction and mass vapor through the membrane to the permeate gap, and mathematically can be expressed as:

$$\dot{Q}_{Membrane} = \dot{Q}_{Conduction} + \dot{Q}_{Evaporation} \quad (2)$$

$$\dot{Q}_{Conduction} = \left(\frac{K_{m,c}}{\delta} \right) * (T_{mf} - T_{ms}) * A_v \quad (3)$$

$$\dot{Q}_{Evaporation} = J_w * \Delta H_v * A_v \quad (4)$$

$k_{m,c}$, and δ are the thermal conductivity and the thickness of the membrane, respectively, T_{ms} is the membrane inner face temperature on the contact side with the support plate, J_w is the mass flow rate of the distillate/permeate product per square meter of membrane area, ΔH_v is the enthalpy of vaporization for the water, and A_v is the effective mass transfer area (area of permeation) through the membrane, which is controlled by the design of the support plate.

The mass flow rate of the distillate/permeate product, J_w , which is a form of vapor crossing the hydrophobic membrane, is because of the difference in vapor pressure resulting from

the temperature difference between the two sides of the membranes. The model of the mass transfer crossing the membrane is expressed as follows [141]–[143]:

$$J_w = D_e * (P_{mf} \cdot \gamma_{wf} \cdot X_{wf} - P_{ms}) \quad (5)$$

$$D_e = \left(\left(\frac{\alpha}{D_k} \right) + \left(\frac{1-\alpha}{D_m} \right) \right)^{-1} \quad (6)$$

$$P_{mf} = \exp \left(23.1964 - \left(\frac{3816.44}{T_{mf} - 46.13} \right) \right) \quad (7)$$

$$P_{ms} = \exp \left(23.1964 - \left(\frac{3816.44}{T_{ms} - 46.13} \right) \right) \quad (8)$$

γ_{wf} is the activity coefficient of water in NaCl solution, and x_{wf} is the mole fraction of water in the feed solution. The vapor pressure on the hot side (P_{mf}) and cold on the cold side (P_{ms}) of the membrane are mainly functions of temperatures. D_e is the diffusion coefficient which is combining the Knudsen and molecular diffusion modes of vapor transfer through the membrane pores [141]–[143].

To avoid the deflection of the membrane due to the pressure applied by the hot feed stream, a support plate is installed. The heat received from the membrane is passed through the support plate to the water gap by three distinct means: conduction through the plate's solid portion, conduction from the membrane directly to the water inside the gap via the support plate's open area, and heat that is carried by the water vapor passing through the support plate's open holes. It can be represented as follows [144]:

$$Q_{support-plate} = \frac{k_{support}}{\delta_{support}} * (T_{ms} - T_{sg}) * A_{support,solid} + \frac{k_{mc}}{\delta} * (T_{mf} - T_{ms}) * A_v + (J_w \times \Delta H_v) * A_v \quad (9)$$

k_{support} is the thermal conductivity of the support plate material, δ_{Support} is the thickness of the support plate, T_{sg} is the temperature of the support plate surface on the contact side with the water gap.

For the circulating water gap, the heat transfer inside the water gap is assumed to be forced convection, and it can be expressed as follows:

$$\dot{Q}_{gap} = h_{gap} * (T_{sg} - T_{pg}) * A_{gap} \quad (10)$$

T_{pg} is the temperature of the cooling plate surface in contact with the water gap, A_{gap} is the heat transfer area inside the water gap.

h_{gap} is the coefficient of convective heat transfer in the water gap, which is given by the following expression:

$$h_g = \frac{Nu_{gap} \cdot k_{gap}}{D_{h,g}} \quad (11)$$

Nusselt number in the gap (Nu_{gap}), in this case, is calculated as follows:

▲ For laminar flow inside the gap (when Reynolds number less than 2300)

$$Nu_{gap} = 1.86 * (Re_{gap} * Pr_{gap} * \frac{D_{h,gap}}{L})^{0.4} \quad (12)$$

▲ For turbulent flow inside the gap (when Reynolds number more than 2300)

$$Nu_{gap} = 0.036 * (Re_{gap})^{0.96} (Pr_{gap})^{0.33} * (\frac{D_{h,gap}}{L})^{0.055} \quad (13)$$

For stagnant water, it is believed that heat is moved across the gap by conduction or natural convection in the conventional WGMD process [145][146][144]. The primary variables

influencing the significance of natural convection, as measured by the Rayleigh number, are the gap width, the temperature variation across the gap, and the gap direction with respect to gravity. Accordingly, the rate of heat transport in the gap is assessed as [147]:

$$Q_{gap} = \left(\frac{k_{eff}}{\delta_{gap}}\right) * (T_{sg} - T_{pg}) * A_{gap} \quad (14)$$

When natural convection is taken into account, the effective conductivity of the water gap is denoted by the symbol k_{eff} . The value of k_{eff} depends on the water gap conductivity, denoted by k_{gap} , and the Nusselt number of gap water, denoted by Nu_{gap} , as shown in the following equation [144]:

$$K_{eff} = K_{gap} \cdot Nu_{gap} \quad (15)$$

For pure conduction, when the Rayleigh number is less than 2000, in this case $Nu_{gap} = 1$ [146]. For Rayleigh number values more than 2000, Nu_{gap} can be evaluated using the following equations [144]:

$$\left. \begin{aligned} Nu_{gap} &= 0.42 (Ra)^{\frac{1}{5}} (Pr)^{0.012} \left(\frac{L}{\delta_{gap}}\right)^{-0.3} \quad , \quad 10^4 \leq Ra \leq 10^7 \\ Nu_{gap} &= 0.46 (Ra)^{1/3} \quad , \quad 10^7 \leq Ra \leq 10^9 \end{aligned} \right\} \quad (16)$$

The MD unit is equipped with a copper plate to cool the MD gap. Heat is transferred through the cooling plate material by conduction, and thus the rate of heat transfer through the cooling plate is given as:

$$\dot{Q}_{cooling-plate} = \left(\frac{k_{plate}}{\delta_{plate}}\right) * (T_{pg} - T_{pc}) * A_{plate} \quad (17)$$

K_{plate} and δ_{plate} are the thermal conductivity and the thickness of the cooling plate respectively, T_{pc} is the temperature at the cooling plate surface in contact with the cold stream, A_p is the heat transfer area of the cooling plate.

Heat transfer through the coolant channel, according to Newton's law of cooling, is calculated as:

$$\dot{Q}_c = h_c * (T_{pc} - T_{bc}) * A_c \quad (18)$$

h_c is the convective heat transfer coefficient in the coolant channel, T_{bc} is the bulk temperature of the cooling stream.

For turbulent and laminar flows, the heat transfer coefficients in the feed, gap, and cooling channels (h_f , h_g , and h_c) are determined based on Nusselt number correlations Nu for laminar and turbulent flows in channels as presented in [23].

3.3. MD Energy Efficiency

For the MD system, the energy efficiency may be evaluated by calculating the gained output ratio (GOR) and the specific energy consumption (SEC). The gained output ratio represents the ratio of the energy used in vapor production (evaporation) to the ratio of the energy consumed by the MD system, it could be approached like the thermal efficiency of the MD unit. The greater the GOR value, the more efficient the system. The following equation expresses the gained output ratio [148];

$$GOR = \frac{J_w \cdot \Delta H_v \cdot A_v}{\dot{E}_{in}} \quad (19)$$

$\dot{E}_{in}$ is the total power consumed by the MD unit, which is given as follows:

$$\dot{E}_{in} = \dot{Q}_{Heating,MD} + \dot{Q}_{Cooling,MD} \quad (20)$$

Where, $Q_{Heating,MD}$ & $Q_{Cooling,MD}$ are the required heating and cooling thermal energy supplied to the MD system.

The specific thermal energy consumption (STEC) of the MD system is defined as the amount of the required thermal energy to generate one cubic meter of freshwater [kWh/m³].

$$STEC = \frac{\dot{E}_{in}}{\dot{V}_d} \quad (21)$$

$\dot{V}_d$ is the volume flow rate of distillate water in m³/hr.

3.4. Solar Thermal Collector Modeling

A solar collector is a special kind of heat exchanger that exchanges heat from a distant source to interacting fluids rather than exchanging heat between two separate fluids. In the current model, a single glass FPC is considered as presented in Figure 3.3.

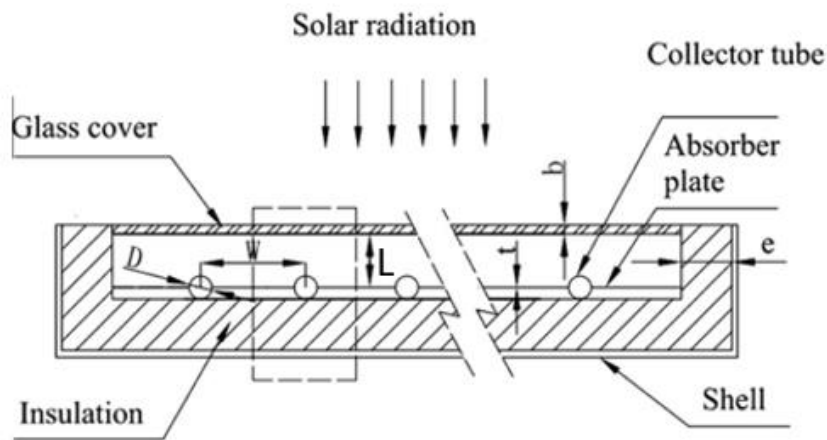


Figure 3.3 Flat plate solar collector cross-sectional view [149]

At steady state operation, the useful energy gain Q_U in (Watts) from a solar collector is determined from an energy balance that relates it to the incident beam and diffuse solar energy, thermal losses, and optical losses as in equation 22.

$$Q_U = A_C * [S - U_L * (T_{PM} - T_a)] \quad (22)$$

Where S (W/m^2) is the absorbed solar energy, U_L in ($W/m^2.K$) is the overall heat loss coefficient from the plate by convection, radiation, and conduction, T_{PM} is the mean plate temperature and T_a is the outside ambient temperature.

Equation (22) calculates the useful energy as a function of the mean plate temperature which cannot be measured for the performance determination. Therefore, part of the model analysis is to reformulate this equation as a function of fluid inlet temperature T_i by introducing a new term which is the heat removal factor F_R . The new resulting equation is as follows:

$$Q_U = A_C * F_R * [S - U_L * (T_i - T_a)] \quad (23)$$

The components of equation (23) are to be determined separately. The model determination has certain assumptions and limitations. This model is for a sheet and parallel tube model operating under steady-state conditions. the headers provide a uniform flow to tube risers and have negligible heat transfer area. The heat flow through the glass cover is one-dimensional with a neglected temperature difference between their surfaces with opaque characteristics to infrared radiations and negligible heat absorption. The temperature gradient between the tubes is neglected and the temperature gradient in the flow direction could be treated separately. The effect of dust and shading is not considered.

To simplify the mathematical calculations of a complete heat transfer circuit similar to the one described in [150], the overall heat loss coefficient U_L is obtained.

Physically, the total heat loss coefficient U_L is the summation of the bottom heat loss coefficient U_B and the top heat loss coefficient U_{Top} . Mathematically, it can be written as in equation (65).

$$U_L = U_B + U_{Top} \quad (24)$$

Where U_B is the conductive bottom heat loss coefficient resulting from high-thickness insulation, where the convective and radiative heat losses from the bottom could be neglected [150].

$$U_B = \frac{K_{insulation}}{\delta_{Insulation}} \quad (25)$$

Where $K_{insulation}$ is the thermal conductivity of thermal insulation in (W/m.K) and $\delta_{Insulation}$ is the thickness of the thermal insulation.

The U_{Top} coefficient is the equivalent radiative and convective heat transfer from the absorber plate to the ambient air. It can be determined when the value of the average plate temperature is first guessed and then by solving the following four equations of the top surface heat transfer simultaneously [150].

$$Q_{Top} = U_{Top} * (T_p - T_a) \quad (26)$$

$$Q_{Top} = (h_{r,p-cover} + h_{c,p-cover}) * (T_p - T_c) \quad (27)$$

$$Q_{Top} = (h_{r,cover-a} + h_{c,cover-a}) * (T_c - T_a) \quad (28)$$

For equations 26-28, h_r is designated for the radiative heat transfer coefficient and h_c is designated for the convective heat transfer coefficients.

For the radiative heat transfer coefficient, it can be determined using the following formulas [150]:

$$h_{r,p-cover} = \frac{\sigma * (T_p + T_c) * (T_p^2 + T_c^2)}{\frac{1}{\varepsilon_p} + \frac{1}{\varepsilon_c} - 1} \quad (29)$$

$$h_{r,cover-a} = \frac{\sigma * \varepsilon_c * (T_c + T_s) * (T_c^2 + T_s^2) * (T_c - T_s)}{(T_c - T_a)} \quad (30)$$

The convective heat transfer coefficients are mentioned as follows [150]:

$$h_{c,cover-a} = 5.7 + 3.8 * V \quad (31)$$

Where V is the ambient air velocity in m/sec. The convective heat transfer coefficient between the double glass could be neglected.

The convective heat transfer coefficient between the absorber plate and the inner glass could be represented as:

$$h_{c,p-cover} = \frac{Nu_{p-c} * K_{a,p-c}}{L} \quad (32)$$

Where L is designated for the space distance between the plate and the cover glass, $K_{a,p-cover}$ represents the air conductivity at the air temperature between the absorber plate and the cover glass, and $Nu_{p-cover}$ is the Nusselt number between the absorber plate and the inner glass, which can be found using the following correlation [150]:

$$Nu = 1 + 1.44 * \left[1 - \frac{1708 * \sin(1.8\beta)}{Ra * \cos(\beta)} \right] * \left[1 - \frac{1708}{Ra * \cos(\beta)} \right]^+ + \left[\left(\frac{Ra * \cos(\beta)}{5830} \right)^{\frac{1}{3}} - 1 \right]^+ \quad (33)$$

In equation 33, the + up script means that the calculated value inside the square bracket shall be positive, otherwise the resultant inside the bracket is equal to zero. β is considered as the collector tilt angle, and Ra is the Rayleigh Number which is a dimensionless number that represents the effect of natural convection and defined as is defined as:

$$Ra = \frac{g * \beta' * \Delta T * L^3}{\nu * \alpha} \quad (34)$$

Where g is the gravitational acceleration in m/sec^2 , β' is the volumetric expansion coefficient ($\beta' = 1/T_{a,i}$), considering the air between the plate and the cover as an ideal gas), ΔT is the temperature difference between the absorber plate and the glass cover, ν is the kinematic viscosity in m/sec^2 and α is the thermal diffusivity in m/sec^2 .

The correction factor F_R named the heat removal factor and is defined as the ratio of the actual useful energy to the useful gain in case the whole collector area is at the temperature of the inlet fluid [150]:

$$F_R = \frac{\dot{m} * C_p}{A_C * U_L} * [1 - EXP(\frac{-A_C * U_L * F'}{\dot{m} * C_p})] \quad (35)$$

Where, $\dot{m}$ is the mass flow rate of the fluid flowing across the collector in (Kg/sec), C_p is the specific heat capacity of this fluid in (KJ/Kg), A_C is the collector area in m^2 , and F' is the collector efficiency factor, it is calculated based on the geometry of the collector as:

$$F' = \frac{1/U_L}{W [\frac{1}{U_L * [D + (W - D) * F' + \frac{1}{C_b + \pi * D_i * h_{fi}}]}]} \quad (36)$$

Where, W is the spacing of the tubes (m), D is the tube nominal diameter (m), D_i is the tube inner diameter, h_{fi} is the inside fluid heat transfer coefficient in ($W/m^2.K$), C_b is the

bond conductance, and F is the fin efficiency factor and it is purely related to the collector geometry, and represented as:

$$F = \frac{\tanh \left[\frac{m(W-D)}{2} \right]}{\frac{m(W-D)}{2}} \quad (37)$$

Where m is a constant required to solve the initial second-order heat transfer equation and defined as:

$$m = \sqrt{\frac{U_L}{K_P \delta_P}} \quad (38)$$

K_P , and δ_P are the thermal conductivity and the thickness of the absorber plate respectively.

The average fluid temperature inside the tubes required to calculate the inside flow heat transfer coefficient is then represented as:

$$T_{fm} = T_{fi} + \frac{\frac{Q_U}{A_C}}{F_R U_L} (1 - F'') \quad (39)$$

Where F'' is another factor required to calculate the fluid mean temperature in the tubes and it is called the fluid flow factor.

$$F'' = \frac{F_R}{F'} \quad (40)$$

The internal heat transfer coefficient of the flow inside tubes could be calculated as:

$$h_{fi} = \frac{Nu_{fi} K_{fm}}{D_h} \quad (41)$$

D_h is the inner hydraulic diameter in (m), K_{fm} is the thermal conductivity of water at fluid mean temperature in (W/m².k), and Nu_{fi} is the Nusselt Number and it is equal to 3.7 for fully developed laminar flow (Re<2300), and for developed turbulent flow it is calculated as [150]:

$$Nu_{fi} = \frac{\left(\frac{f}{8}\right) * (Re - 1000) * Pr}{1.07 + 12.7 \sqrt{\left(\frac{f}{8}\right) * \left(Pr^{\frac{2}{3}} - 1\right)}} \left(\frac{\mu}{\mu_w}\right)^{0.25} \quad (42)$$

Where, Pr is Prandtl Number and μ is the viscosity of water at the fluid mean temperature, μ_w is the viscosity of water at the tube walls temperature. The tube wall temperature is assumed to be equal to the mean absorber plate temperature and f is the friction factor for smooth pipes and it is defined as:

$$f = (0.79 * \ln(Re) - 1.64)^{-2} \quad (43)$$

The solution is an iterative process, it is a common practice to assume a plate temperature for the first hour as 10 °C greater than the fluid inlet temperature, then it could be calculated for the next hour as:

$$T_{pm} = T_{fi} + \frac{\frac{Q_U}{A_C}}{F_R * U_L} (1 - F_R) \quad (44)$$

3.5. Integrated Solar-MD Modeling & Energy Efficiency

The integration model of both solar-MD systems is presented in this section, the system configuration is presented in Figure 3.4.

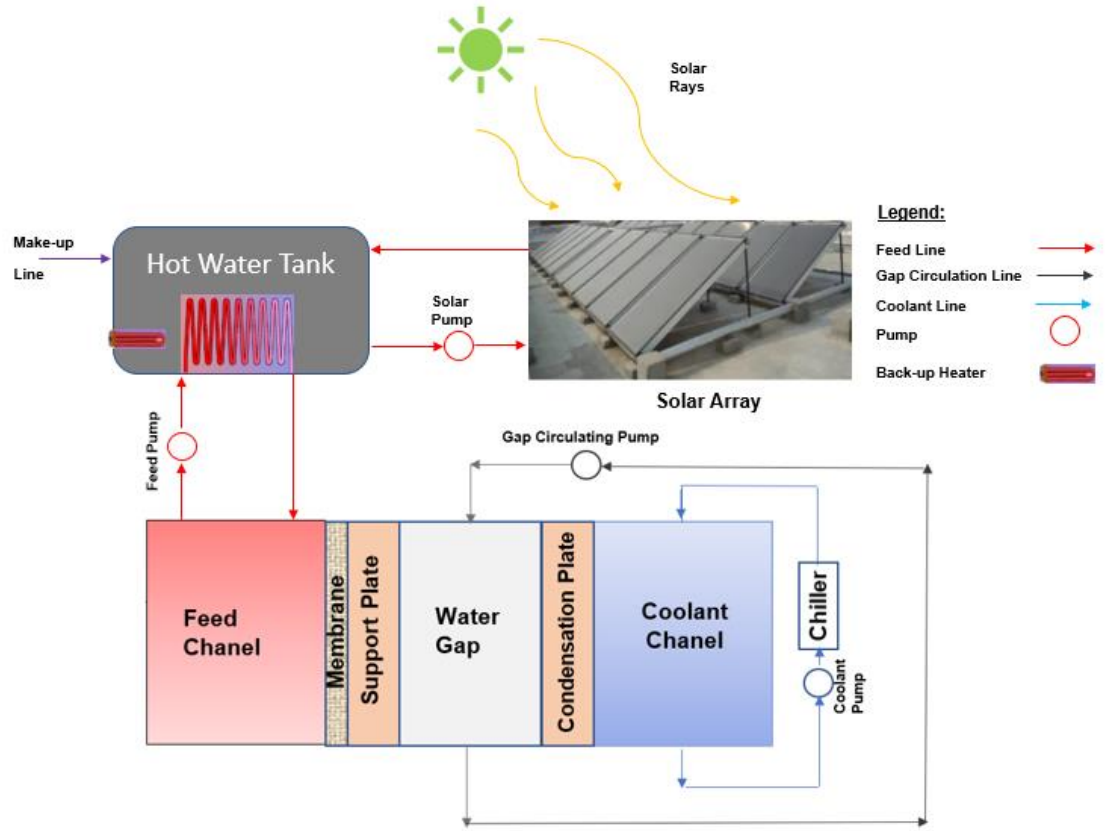


Figure 3.4 Configuration of the solar-powered circulated water gap WGMD system

For the system integration, the storage tank shown in Figure 3.4 has been considered as a control volume, the circulation water through the FPC increases the internal energy of the storage tank, whereas the Heat Exchanger (HE) connected to the MD Module consumes thermal energy from the tank.

The energy balance equation could be presented as:

$$Q_U - Q_{Tank\ Loss} - Q_{Heating,MD} = m_{WT} * C_{p,WT} * (T_{Water,Tank,(H+1)} - T_{Water,Tank,(H)}) \quad (45)$$

Where, m_{WT} is the mass of water inside the storage tank, $C_{p,WT}$ is the specific heat of water inside the storage tank at $T_{Water,Tank,(H)}$ which is the temperature in the tank at a certain

time hour “H”, $Q_{Tank\ Loss}$ is the heat losses from the storage tank, and $Q_{Heating,MD}$ is the consumed energy by the MD system for the heating requirement.

For the solar FPC, the thermal efficiency could be defined as the useful gained solar energy to the available incident solar. It can be expressed as:

$$\eta_{Solar} = \frac{Q_U}{G * A_c} \quad (46)$$

Where G is the available solar intensity in (W/m^2).

Equations 19, 21, and 46 describe the efficiency of each system separately, hence to relate the amount of energy consumed in the water product to the available solar energy, a special term has been introduced which is the GOR_{STD} , which refers to the GOR for Solar Thermal Desalination (STD) [53]:

$$GOR_{STD} = \frac{J_w * \Delta H_v * A_v}{G * A_c} \quad (47)$$

For solar-thermal desalination, the GOR value described in equation (19), considers all the thermal energy entering the desalination device only. Whereas GOR_{STD} presented in equation (47) is the metric evaluating the useful energy used in water production to the available solar energy input. In other words, the GOR_{STD} describes the whole picture of the solar desalination process and this value could be referred to as:

$$GOR_{STD} = GOR * \eta_{Solar} \quad (48)$$

3.6. Cost Estimation Model

References defined the production cost to the Unit Production Cost (UPC) of water in US\$/m³. The predicted cost is based on the Capital Expenditures (CAPEX) and Operational Expenditures (OPEX) economic model. To begin with, the UPC could be expressed as [151], [152], [153] :

$$UPC = \left(\frac{\dot{C}_T}{\dot{V}_{\text{freshwater}}} \right) \quad (49)$$

Where, $\dot{V}_{\text{freshwater}}$ is the yearly water production, A_{Membrane} is the membrane vapor crossing area, f is the availability factor and 8760 is the number of hours in the year.

and $\dot{C}_T$ is the total annual cost which can be represented as follows:

$$\dot{C}_T = \dot{Z}_f + \dot{C}_{\text{MRC}} + \dot{C}_L + \dot{C}_{SP} + \dot{C}_E + \dot{C}_{Ch} + \dot{C}_{BD} \quad (50)$$

The membrane replacement cost $\dot{C}_{\text{MRC}}$, the labor cost ($\dot{C}_L$), the maintenance and spare-part cost ($\dot{C}_{SP}$), the energy cost for the heat process and pumping operation ($\dot{C}_E$), the chemical treatment cost ($\dot{C}_{Ch}$), and the brine disposal cost $\dot{C}_{BD}$ are considered for the annual operational cost.

$\dot{Z}_f$ is the annualized capital cost, which is mainly a function of the system lifetime and the yearly interest rate for investment, such that:

$$\dot{Z}_f = Z \times \alpha \quad (51)$$

Where, Z is the initial capital expenditure cost and α is the amortization factor which can be expressed as:

$$\alpha = \frac{i \times (i+1)^n}{(i+1)^n - 1} \quad (52)$$

Where, i is the annual investment interest rate and n is the system lifetime.

The cost of the membrane replacement could be determined using the following formula:

$$\dot{C}_{\text{MRC}} = \dot{R}_{\text{Mem}} \times \text{COM} \quad (53)$$

$\dot{R}_{\text{Mem}}$ is the rate of membrane replacement and COM is the cost of the membrane.

3.7. Models Validation

Both the circulating WGMD and the solar FPC models have been experimentally validated independently. The solar model has been validated against the data published in the literature, whereas the MD model has been validated through experimental data conducted by Khalifa [72].

3.7.1. Validation of Circulating WGMD Model

The WGMD model results have been validated with experimental data [72] for different operating interests, and the validation results are presented in Figure 3.5. The validation has been performed by varying the main operating conditions such as the feed flow rate and temperature, and coolant flow rate and temperature. All comparisons have been performed based on 8 mm gap thickness. The first comparison is made under a coolant temperature of 15 °C, a feed flow rate of 5.3 L/min, a coolant flow rate of 4.3 L/min, a gap circulation flow rate of 2.35 L/min, and an 8 mm gap thickness. The model has presented an accuracy of 5% for predicting the permeating flux at various feed temperatures. Another comparison has been performed for varying the feed flow rate with a coolant and feed temperatures of 18 °C and 70 °C respectively, a coolant flow rate and gap circulation rate of 3.5 L/min and 2.35 L/min respectively, a maximum difference of 10% with the experimental results has been observed. Another comparison for the variation of cooling flow rate revealed a maximum deviation of 4% with the experimental data using the conditions of 18 °C coolant temperature, 70 °C feed temperature, 3 L/min feed flow rate, 2.35 L/min circulation flow rate, and 8 mm gap thickness. Regarding the effect of the circulation rate, the comparison has been for the range of 1.7 to 4.1 L/min while keeping the 15 °C coolant temperature, the 70 °C feed temperature, the 4.3 L/min coolant flow rate,

the 5.3 L/min feed flow rate, and the 8 mm gap thickness. For all values of the circulation flow rate, the model showed accuracy with a maximum error of about 6% at the low flow rate of 1.7 L/min.

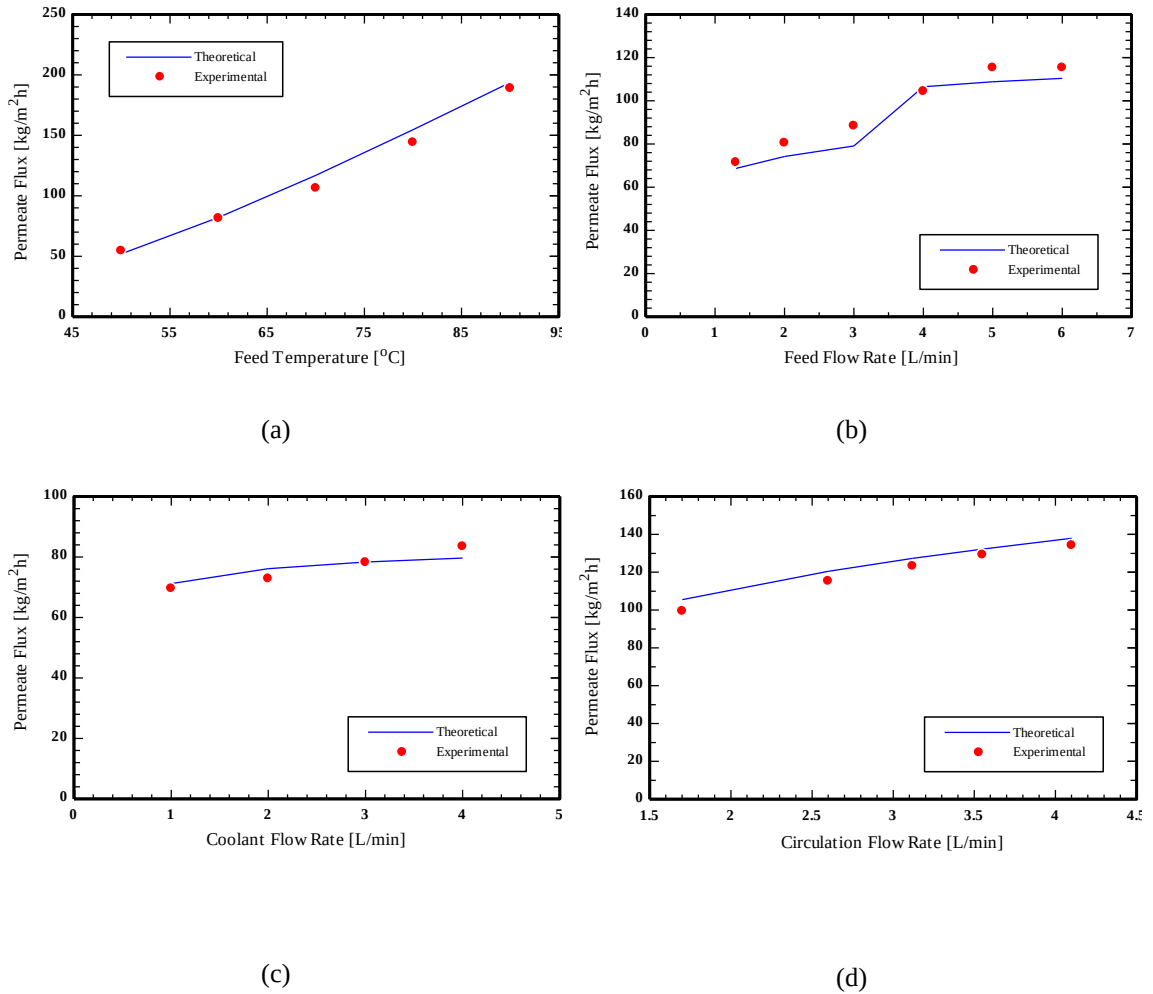


Figure 3.5 Model validation of the circulating WGMD at different operating parameters

3.7.2. Validation Integrated FPC-Storage Tank Model

The flat plate collector has been validated using two experimental studies presented in [154], [155]. The first study [154], consists of experimental work to test the efficiency of flat plate solar FPC working with different nano-fluids, the experiment has been performed

based on ASHRAE standards. For this experiment the circulating water flow rate through the FPC and the internal heat exchanger of a 41 L hot water storage tank with a fixed circulation rate of 1 L/min ($\dot{m} = 0.0166 \text{ Kg/s}$), the load is considered to be a cold water stream passing through the hot water tank to control the inlet fluid temperature to the FPC, which is measured as shown in Figure 3.6.

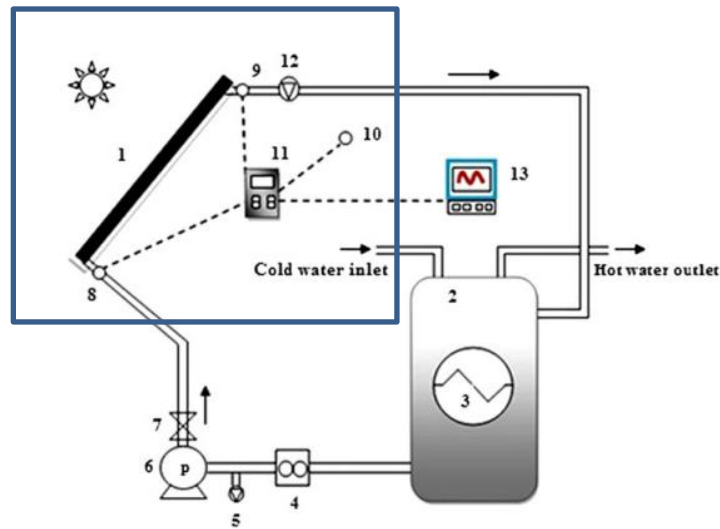


Figure 3.6 Setup configuration for the performance of a flat plate solar collector [154]

In the experimental work presented in [52], the authors provided the inlet and outlet temperature to the FPC, T_8 , and T_9 respectively, the hourly ambient temperature, the average solar intensity, and the solar collector specification as presented in Table 3.1. These data are used as input for the present simulation model, except the FPC exit temperature T_9 was obtained as an output to validate the present model. The obtained output T_9 is presented in Figure 3.7.

Table 3.1 Flat Plate Solar Collector Specification [154]

Specification	Dimension
Occupied Dimension [cm]	200x94x9.5
Absorption Area [m ²]	1.51
Glass Thickness [mm]	4
Riser Pipe [mm]	Diameter = 10 Thickness = 0.9
Absorption Sheet	Thermal Emission= 7% Solar Absorption =96.2 % Coting Method: Vacuum Magnetron Sputtering

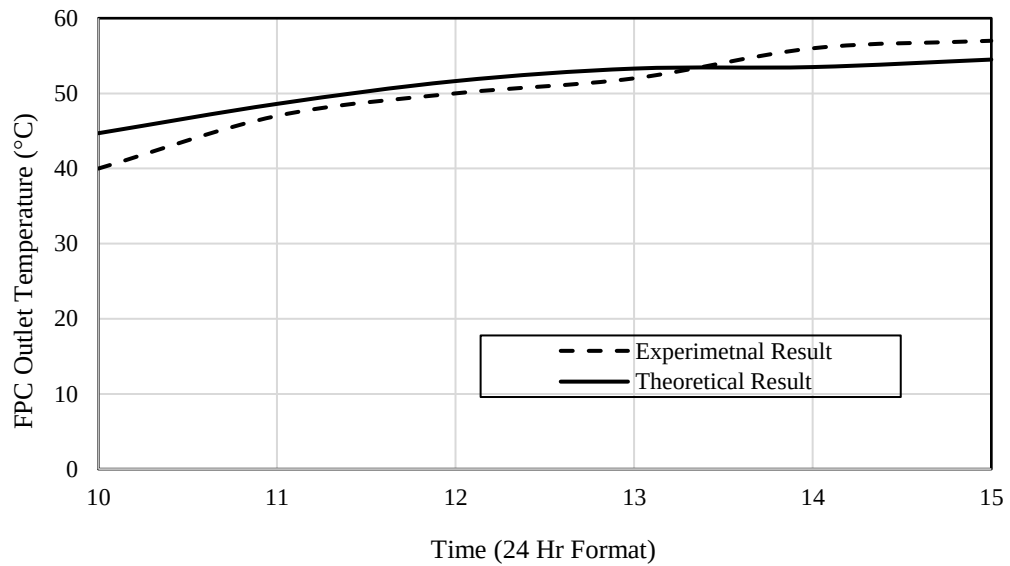


Figure 3.7 . Experimental vs theoretical FPC outlet temperature (T9) validation based on the data in [120], with inlet temperature (T8) set as input to the model.

The validation with this experimental work is just for the FPC as a heat exchanger that gains and loses heat. It can be noticed that the largest error was 10 % at 10:00 am, this is because the mean plate temperature was first assumed 10 °C greater than the water inlet temperature, then when it is calculated by the simulation model the results showed a maximum error of 3.5%.

After the FPC Model has been validated, it is required to validate the integrated FPC-Tank subsystem, equation 45, which is considered to have a small heat loss from the tank, in addition to considering that the average tank temperature is the inlet temperature to the FPC. The tank-collector model has been validated against the data published in [155] and presented in Figure 3.8. The experiment was conducted using two FPC with a collecting area of 3m^2 connected in parallel to a 140 L storage tank. In this experiment, the mass flow rate for the circulation loop has been recorded and used as an input to the simulation model. The ambient temperature and available solar intensity and the solar FPC specifications have been taken as input whereas the output of the model was the average tank temperature. Both theoretical and experimental data showed the same trend with a maximum error of 4% at 15:00.

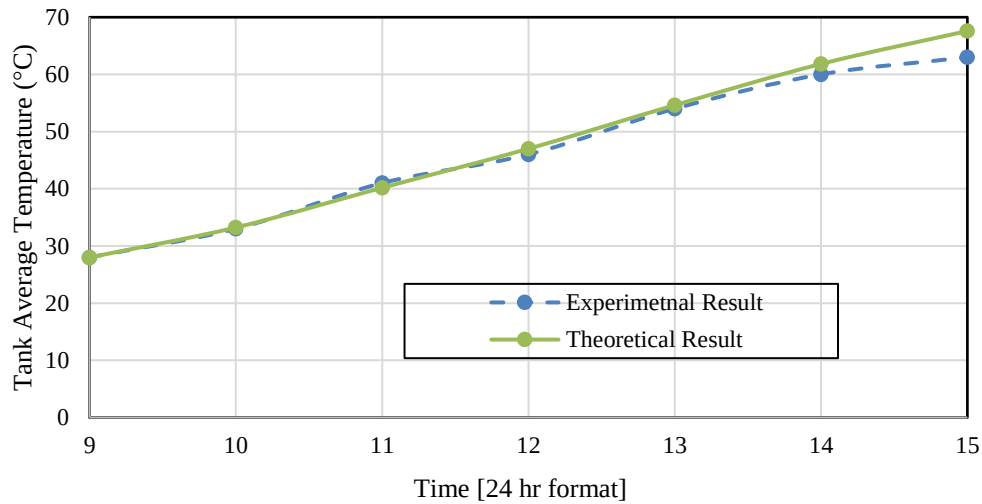


Figure 3.8 Experimental Vs Theoretical Integrated Average Tank Temperature Validation Based on Data in [155].

CHAPTER 4

THEORETICAL RESULTS

4.1. Circulating Water Gap Membrane Distillation

A comprehensive parametric study is conducted to determine the effect of gap circulation on the performance of the water gap MD system. The experimental investigation that was conducted by Khalifa [72] examined the primary operating parameters, including feed inlet temperature, water gap width, coolant inlet temperature, hot and cold water flow rates, and gap circulation flow rate; however, the tested ranges of the operating parameters were constrained by setup capabilities and system operation constraints. For example, the tested feed inlet temperature ranged between 50–90°C, the water gap sizes were 8–16 mm, the cold water inlet temperature ranged between 10–18°C, the feed flow rate was between 1.3 and 6 L/min, the coolant flow rate was between 1 and 4 L/min, and the gap circulation flow rate was between 1.7 and 4.1 L/min. The same operating parameters are examined in this analysis; however, more comprehensive ranges are investigated such that for the same experimental feed temperature range, the coolant temperature increases from 5–30°C, the feed, coolant, and circulation flow rates are set from 1–10 L/min, and the gap size is increased from 2–50 mm. The analysis is performed at a constant feed concentration of 150 ppm and a membrane area of 0.0066 m². The specific energy consumption, production cost, gained output ratio, and output flux are reported and discussed with operating parameters to evaluate the performance of the circulated WGMD unit. The results are then compared to the baseline WGMD module design.

In this study, the water in the gap is moving (gap circulation); therefore, the ratio of natural to forced heat transfer inside a gap is defined by the Richardson number, $Ri = Gr/Re^2$. Ri is a dimensionless number that expresses the ratio of the buoyancy term to the flow shear term. When this ratio is much less than unity, the effect of natural convection (buoyancy) is unimportant, and only forced heat transfer is considered. When Ri is greater than unity, the natural convection dominates. Figure 4.1 shows the influence of the circulation rate on Richardson number, for different feed temperatures from 60°C to 90°C. In general, Ri_{gap} equals unity and above for gap circulating rates below 1.0 to 1.5 l/min (depending on the hot water temperature). Thus, natural convection is important and must be considered in the heat transfer evaluation if the gap circulation rate is below 1.0 l/min. Above a gap circulating rate of 1.5 l/min, the value of Ri_{gap} is significantly less than one which indicates dominating mode of forced convection heat transfer.

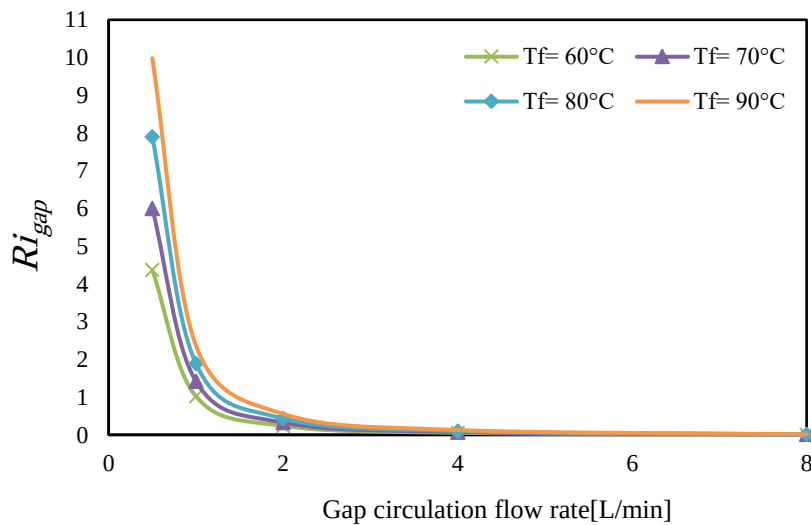
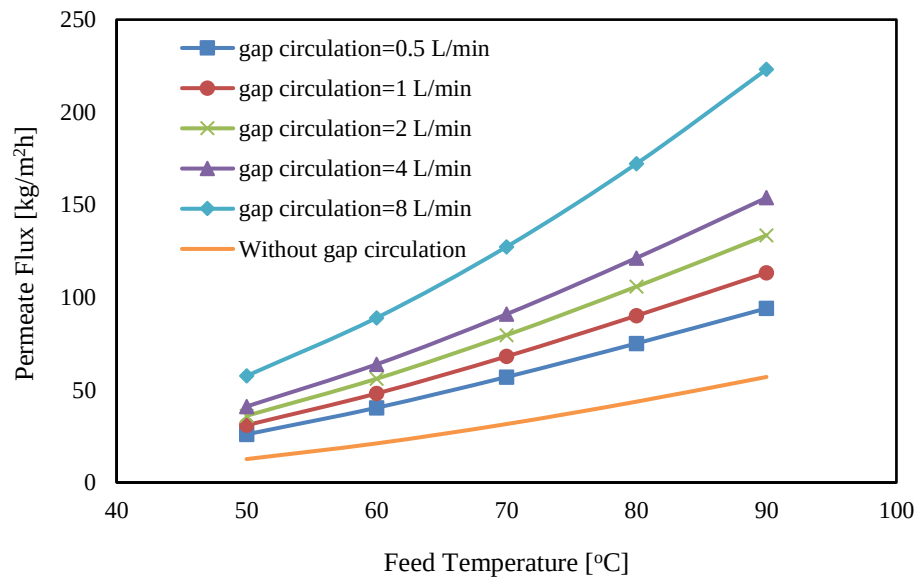


Figure 4.1 Richardson number as a function of the gap circulation rate, at different feed temperatures

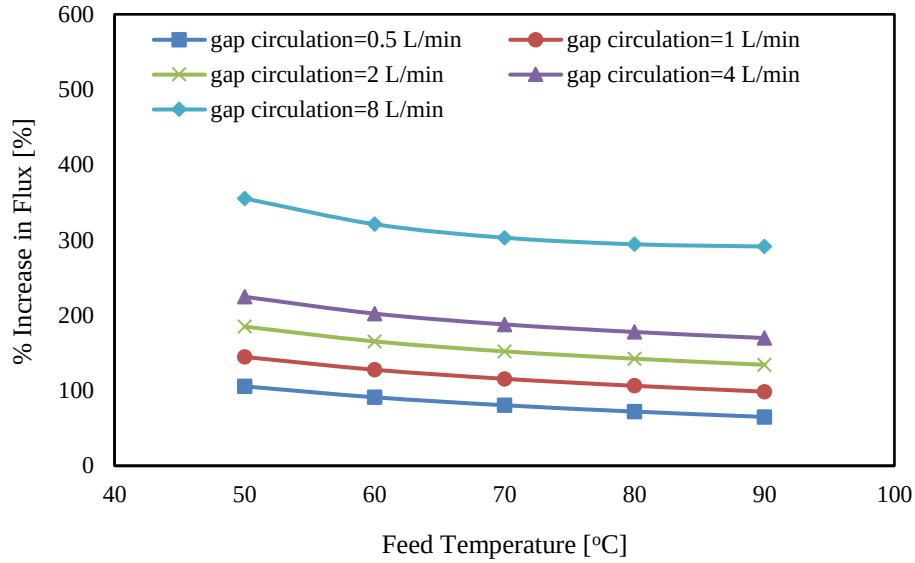
4.1.1. Effects of Operating Parameters on the Permeate Flux

The permeate flux is a commonly used parameter to evaluate the productivity of the MD process. The permeate flux represents the amount of freshwater obtained from the MD system per unit area of the membrane over one hour. The temperature variation between the cold and hot streams is the driving force for vapor permeation across the membrane. As a result, it is critical to investigate the effects of coolant and feed temperatures on the water production of the WGMD system with circulated gap. Figure 4.2 (a) compares the water gap MD system with circulated gap to the conventional WGMD system (water gap without gap circulation) in terms of permeate flux at various hot water temperatures, varying from 50 to 90°C and different circulation flow rates. Other operating parameters are kept constant, including the coolant temperature of 15°C, the feed and coolant flow rates of 3 L/min, and gap size of 8 mm. As illustrated in Fig. 4.2 (a), permeate flux increases as feed temperature increases, for both systems, due to significant growth in the net vapor pressure across the membrane, resulting in a higher distillate production rate. When the hot water temperature changes from 50 to 90°C, the output flux improves by an average of approximately 270%; however, the system without circulation exhibits a nearly 350% increase in permeate flux under the same conditions. It is easy to see from these high percentages how essential is the feed temperature in the membrane distillation process. It is also discovered that boosting the gap flow rate improves output flux due to the improvement of the gap's heat transfer coefficient associated with the water movement, resulting in an increased condensation rate. At a circulation flow rate of 8 L/min the flow characteristic inside the gap changes from laminar to turbulent for all feed temperatures (all values of Reynolds number within the gap are greater than 2300), this explains the

noticeable growth in the output flux at 8 L/min circulation rate compared to other circulation flow rates. Figure 4.2 (b) depicts the percentage increase in the permeate flux when the gap water is circulated compared to the permeate flux obtained with the conventional water gap system to visualize the flux enhancement. It is found that when the circulation is used, the output flux increases by a maximum of about 355% at 50°C hot water temperature and 8 L/min gap flow rate. It can be noticed that at lower feed temperatures, the improvement in the permeate flux is more significant as a percentage. Therefore, adopting the gap circulation may be advantageous when the maximum hot water temperature is limited. At 90°C hot water temperature and 8 L/min circulation rate, flux increased by order of magnitude to roughly 220 kg/m²h.



(a)



(b)

Figure 4.2 Influence of gap circulation on output flux at various feed temperatures
Conditions: 15°C coolant temperature, 3 L/min cold and hot water flow rates, and 8 mm gap thickness

Another critical operating parameter of the MD system is the coolant temperature, as the permeation rate is directly related to the temperature variation between the heated and cooled membrane surfaces. In addition, the cooling stream is necessary to cool the gap water and maintain it at the required temperature. For the WGMD system, the thermophysical characteristics of the gap water are critical when modeling the overall mass and heat transfer. Therefore, it is believed that the gap water movement will improve heat transfer between the cold and hot streams, resulting in permeate flux enhancement.

The change in the flux with the temperature of the cold water is investigated, with and without gap circulation, as displayed in Figure 4.3. The temperature of the cold stream is varied between 5 and 30°C in 5°C increments. The feed temperature is 70°C for all coolant temperatures, the cold and hot water flow rates are 3 L/min, and the gap size is 8 mm.

Results indicate that reduced coolant temperature positively affects output flux for both systems due to the increased temperature variation through the membrane. When the cold

water temperature is lowered from 30 to 5°C, under the above-mentioned operating conditions, flux increases by a maximum of around 50% with a gap circulation flow rate of 0.5 L/min. However, within the same coolant temperature range, a roughly 55% improvement in permeate flux is observed for systems without circulation. This demonstrates that the coolant stream is more effective at cooling stagnant water than the circulating stream. It can be concluded that gap circulation increases the flux by a maximum of about 350% compared to a stagnant water gap, which is achieved at a coolant temperature of 30°C and a circulation flow rate of 8 L/min. It can be observed that at 8 L/min circulation flow rate the change in the flux associated with changing the coolant temperature is low compared to other flow values. Because at 8 L/min the flow rate inside the water gap is turbulent for all values of the cold-water temperature, which makes the output flux less dependent on the variation of the coolant temperature.

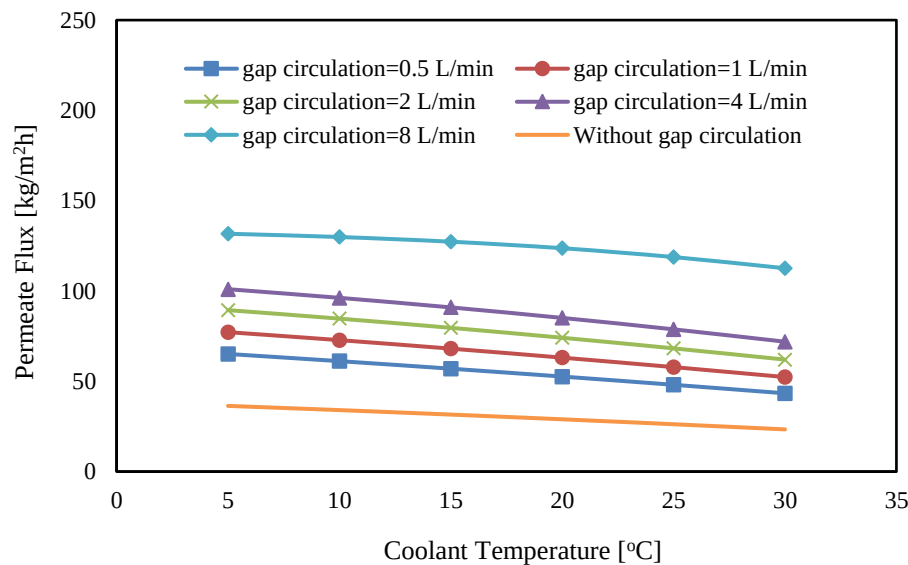


Figure 4.3 Influence of circulated gap on output flux at various cold-water temperature
Conditions: 70°C feed temperature, 3 L/min cold and hot water flow rates, and 8 mm gap size

Figure 4.4 illustrates the influence of adjusting the hot water flow rate on the flux for both systems. With a gap thickness of 8 mm, hot and cold water temperatures of 70 °C and 15

°C, respectively, and a coolant flow rate of 3 L/min, the hot water flow rate is changed between 1 and 10 L/min to investigate the effect of hot water flow rate on output flux. In general, the output flux improves with rising the flow rate of the feed stream due to the enhancement of the turbulence level in the hot water flow channels resulting in improved mixing in the boundary layer and improved characteristics of heat transfer on the feed side of the membrane. Looking at Figure 4.4, it can be noticed that when the hot water flow rate exceeds 3 L/min, the output flux abruptly rises due to the flow transformation from laminar to turbulent for Reynolds values more than 2300. When the hot water flow rate is changed from 1 to 10 L/min, a considerable rise in permeate flux occurs, which is about 40 to 90% for the circulation flow rates of 0.5 to 4 L/min; however, the enhancement in flux with hot water flow rate is about 270% at a circulation flow rate of 8 L/min with a Reynold Number of 2700. In other words, the 270% increase of flux at a hot water flow rate of 4 l/min and circulation rate of 8 l/min is the result of turbulent flow in both feed and gap channels which indicates the effect of forced heat transfer convection on the increase of the system's flux. The comparable increase in flux for a water gap without circulation is 25%, indicating that circulated gap functions better with a higher hot water flow rate, resulting in higher output flux values. From these percentages, it can be concluded that having a higher feed flow rate as well as a higher circulation flow rate is the optimum condition to obtain the maximum output flux of the water gap MD system. At 10 L/min hot water flow rate and circulation flow rate of 8 L/min, the maximum output flux is achieved, which is about 330 kg/m².hr.

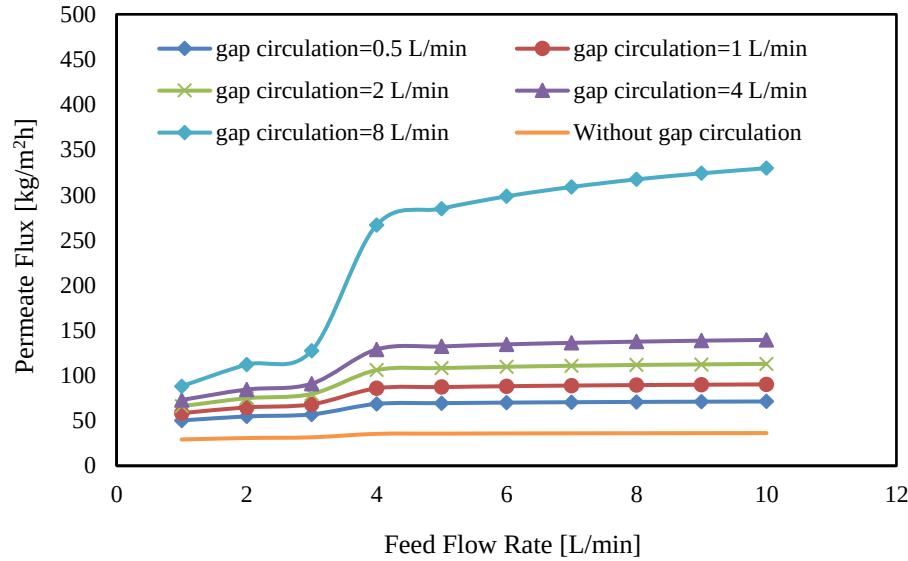


Figure 4.4 Influence of gap circulation rate on output flux at various feed flow rates.
Conditions: 70°C feed temperature, 15°C cold water temperature, 3 L/min cold water flow rate, and gap size of 8 mm

To investigate its effect on the output flux, the cold water flow rate is changed within the same range as the hot water flow rate, while the other parameters remain constant at 70°C hot water temperature, 15°C cold water temperature, 3 L/min hot water flow rate, and 8 mm gap size. Figure 4.5 illustrates the variance in output flux as a function of coolant flow rate for both systems. When the cold-water flow rate is changed from 1 to 10 L/min, the permeate flux increases by an average of 14% for the system with gap circulation; however, it improves by about 6% for the system without gap circulation. The Reynolds number through the coolant channels varies between 225 (at a low flow rate of 1 L/min) and 2250 (at a high flow rate of 10 L/min), remaining within the laminar flow region, which explains the gradual change in output flux as the cold water flow rate is increased. Along with the improvement produced by raising the cold-water flow rate, the addition of the gap circulation results in further flux enhancement. The circulated gap water enhances permeate flux by a maximum of 300% compared to a system with a stagnant water gap, which is obtained at a circulation flow rate of 8 L/min.

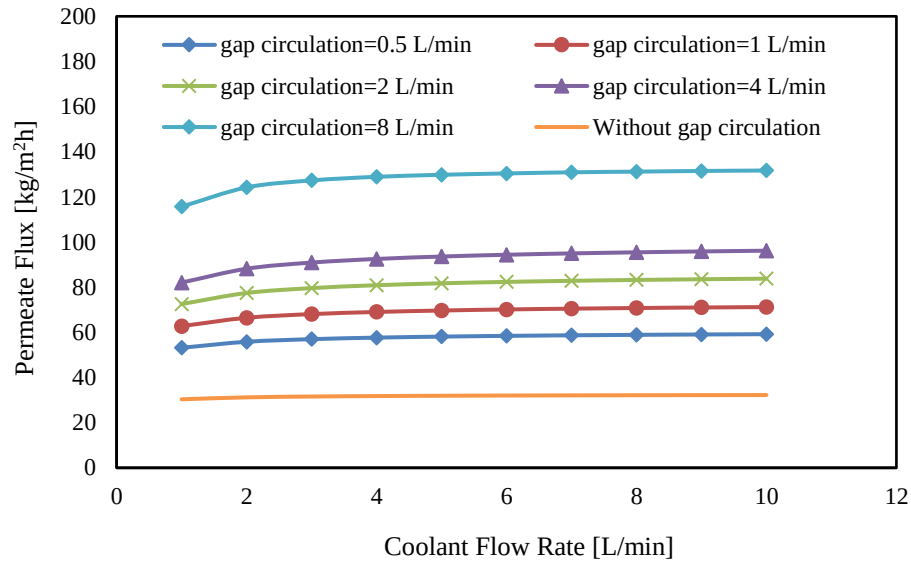
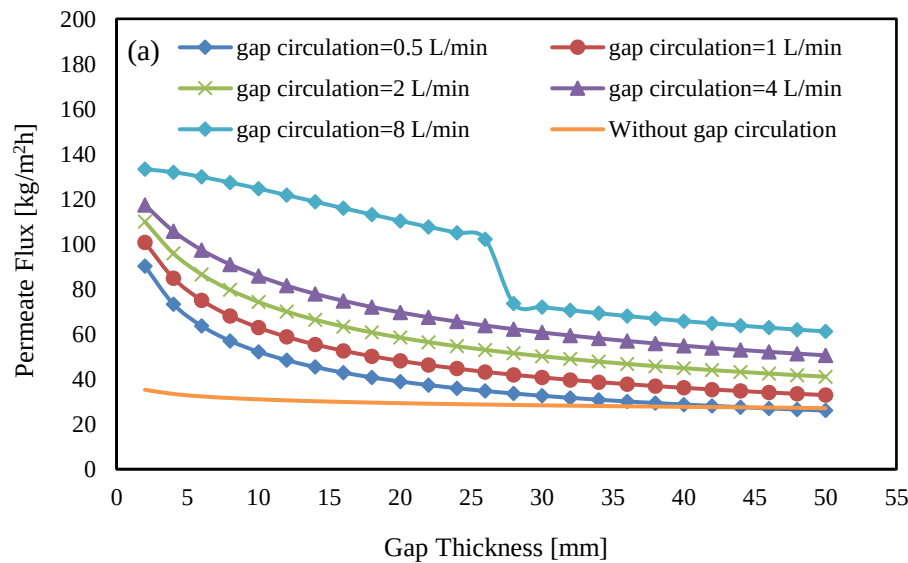


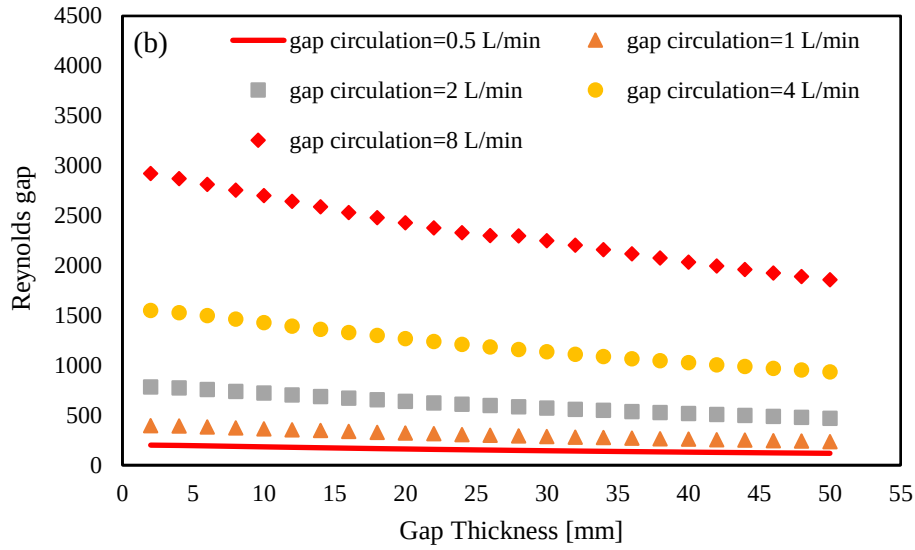
Figure 4.5 influence of gap circulation on output flux at various cold-water flow rates.
Conditions: 70°C feed temperature, 15°C coolant temperature, 3 L/min hot water flow rate, and 8 mm gap size

When analyzing the performance of the WGMD unit, one of the most critical factors to be considered is the gap size. The MD process uses the gap to reduce heat loss between the cold and hot streams and thus the process's energy consumption. As a result, the gap size directly affects both the energy consumed and the output flux. The effect of gap circulation at various gap sizes is demonstrated in Figure 4.6. In this case, the gap size is varied between 2 and 50 mm. The analysis is conducted at constant operating temperatures of 70 °C and 15 °C for the feed and coolant, respectively, at flow rates of 3 L/min for the coolant and feed. As shown in Figure 4.6, the output flux decreases noticeably as the gap size increases. changing the gap size from 2 to 50 mm reduces the output flux by a maximum of 71% at a circulation flow rate of 0.5 L/min. On the other hand, there is only a 20% reduction in the flux observed when the gap size is changed from 2 to 50 mm for the system without circulation. Compared to the impact of feed and coolant temperatures, these lower percentages indicate that gap size has little effect on the output flux of WGMD system. This conclusion can be extremely beneficial in the design of improved MD systems, as it

allows for a larger gap without negatively impacting system flux while providing additional space for internal design and manufacturing of circulation inlet and exit. It can be seen that at a higher circulation flow rate (8 L/min) there is a slight drop in the flux values with expanding the gap size up to 26 mm, then a significant decrease in the flux is obtained when the gap size is adjusted to 28 mm. This sudden drop is due to the transformation of the flow type inside the gap from turbulent to laminar at a gap thickness of 28 mm, which can be observed from the values of Reynolds number within the water gap as shown in Figure 4.6 (b). Figure 4.6 (b) also represents the variation for gap flow rates from 0.5 to 4 l/min showing a laminar flow for all cases. This is the main reason behind the absence of a sudden drop in the stated flow rates in Figure 4.6 (a).



(a)



(b)

Figure 4.6 Effects of gap thickness on output flux and gap Reynolds number at different gap circulation rates
Conditions: 70°C hot water temperature, 15°C cold water temperature, 3 L/min cold and hot water flow rates, circulation rate 8 L/min

4.1.2. Energy Performance

Many variables influence the STEC and GOR of the WGMD unit; however, the hot water temperature is the most significant factor. The following analysis examines the impact of gap circulation on the GOR and STEC at various hot water temperatures and circulation flow rates. This analysis is carried out under constant operating conditions of 15 °C coolant temperature, 3 L/min cold and hot water flow rates, and a gap size of 8 mm. Figure 4.7 compares STEC and GOR of the water gap system with and without circulated gap. Results reveal that the circulated gap system shows better energy performance (higher GOR and less SEC), compared to non-circulated gap system, for the circulation rates up to 4 L/min. When the circulation flow rate rises to 8 L/min, the module with gap circulation has lower energy performance (lower GOR and higher STEC) than a module with no circulation. This decrease in energy efficiency results from the increased energy loss between the cold and hot streams caused by the increased circulation rate. Compared to the enhancement in

the production rate that is obtained at a circulation rate of 8 L/min, the reduction in energy efficiency is marginal. For example, when a circulation flow rate of 8 L/min is used, the permeate flux increases by an average of about 300% compared to that for the system with no circulation; however, the GOR decreases by approximately 20%, and the SEC increases by approximately 24% on average.

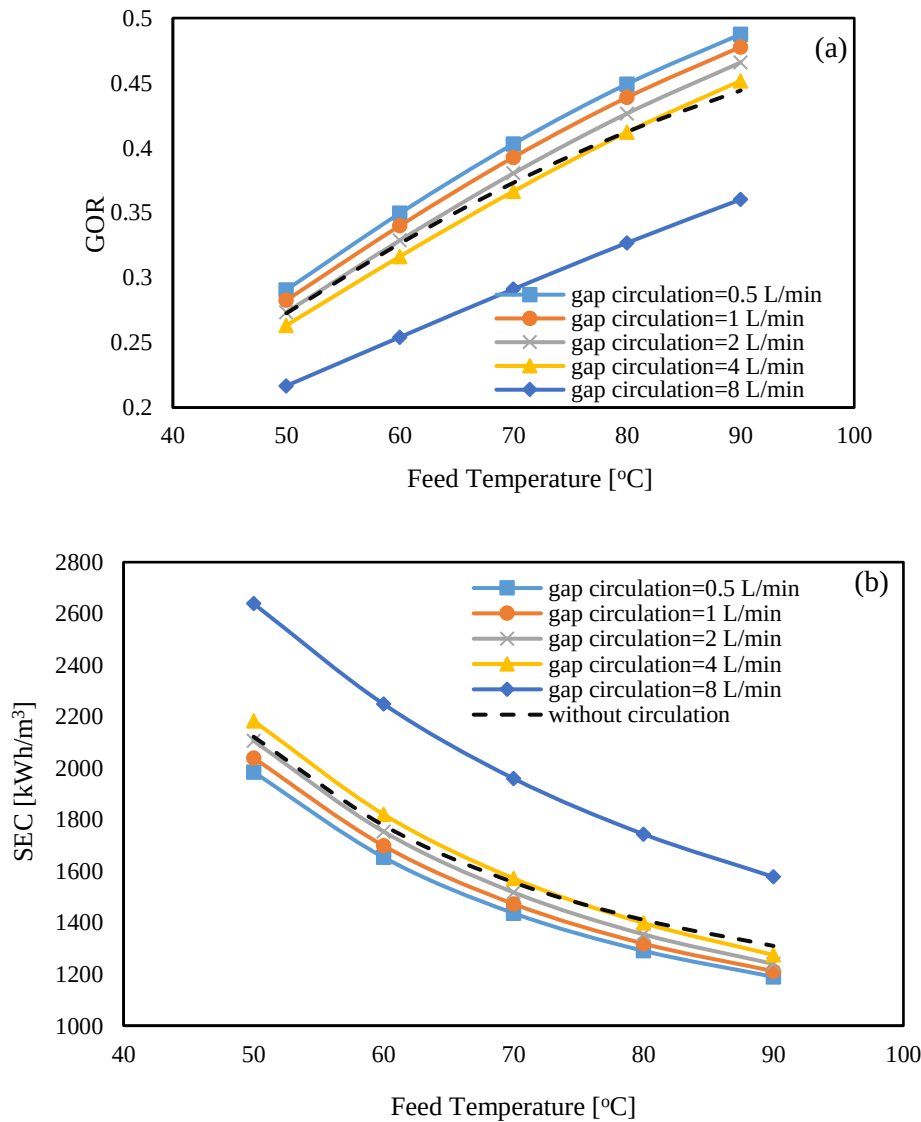


Figure 4.7 influence of hot water temperature and circulation rate on the system GOR and SEC

Conditions: 15°C cold water temperature, 3 L/min cold and hot water flow rates, 2 L/min circulation flow rate, and gap size of 8 mm

4.2. Solar-Powered Membrane Distillation

The solar intensity is different from one location to another, from one hour to another, and from one day to another. To provide a yearly estimation for the water production from the solar-assisted circulating PGMD presented in Figure 3.4 for the Dharan city, KSA, the system analysis is based on a yearly average day, with average solar intensity and average weather conditions. A day in April was chosen as an average day for the year [156], and the analysis has been performed. The daily solar intensity and the daily ambient temperature are presented in Figure 4.8.

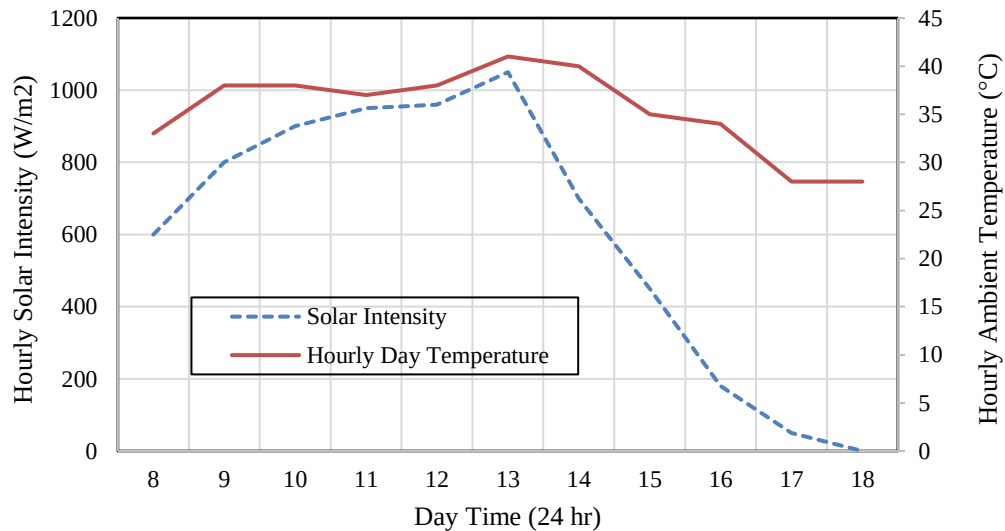


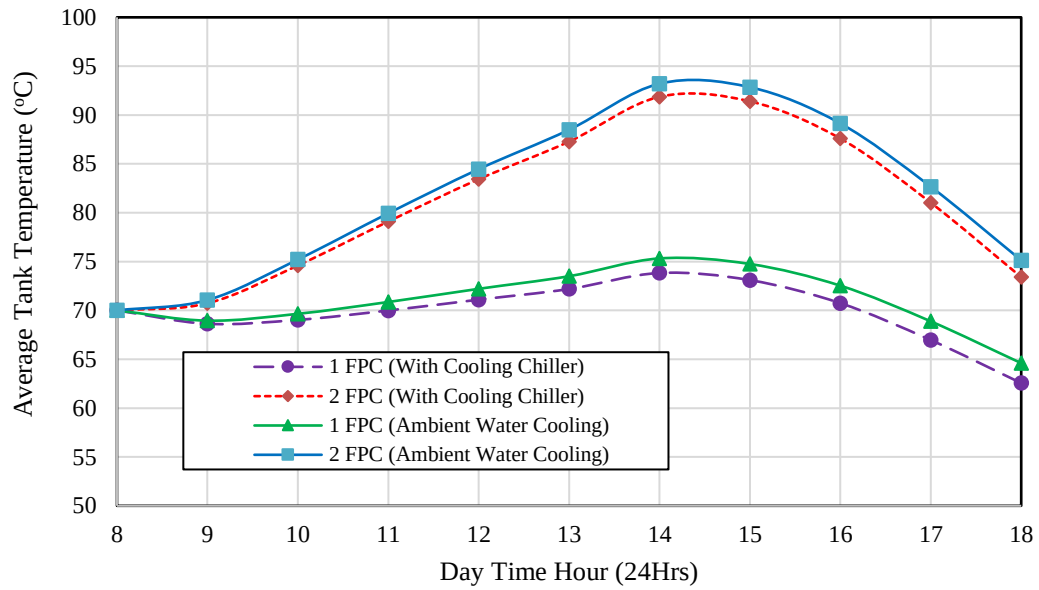
Figure 4.8 The hourly average recorded solar Intensity “G” in (W/m²) and ambient temperature in (°C) for April in Dharan [122]

Analysis for the system in Figure 3.4 is for 160 L HWT at 150 ppm constant feed concentration through a 0.0066 m² PTFE hydrophobic membrane with 0.45 μm pore size and 80% porosity. Brass plates, 1.5 mm thick, serve as support and condensation plates. The simulation was performed using SUN600 FPC from "lacaze-energies" with a 1.9 m² aperture collecting area, 95% absorption coefficient and 5% emissivity aluminum absorber

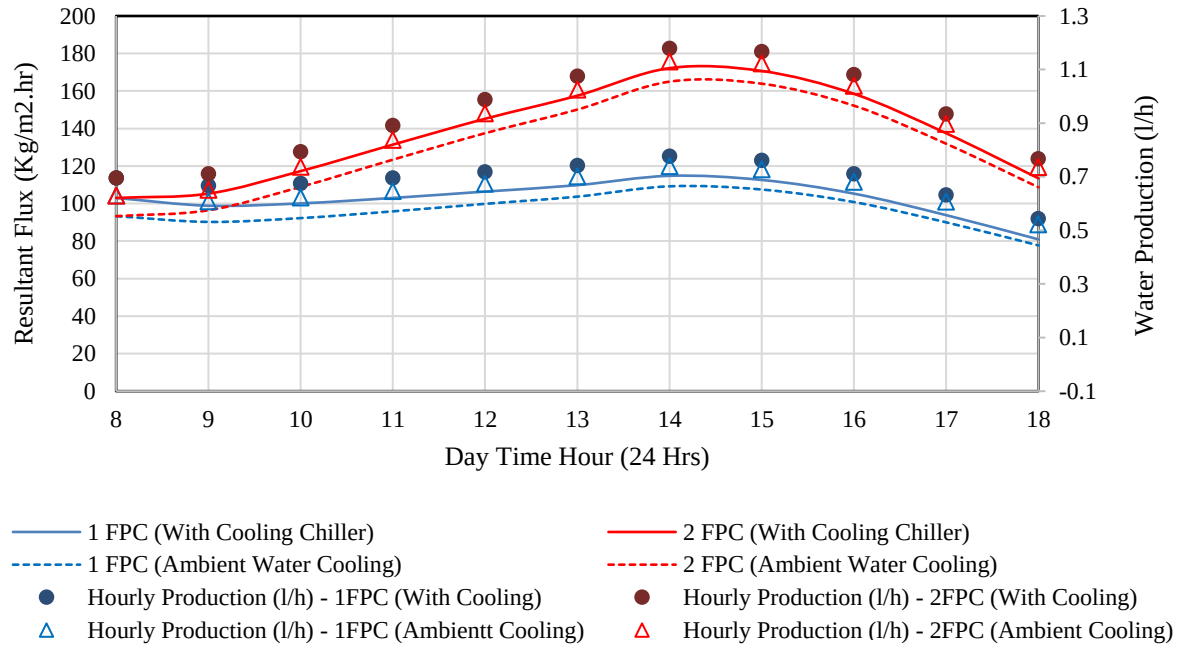
plate, and 30 mm rock wool insulation. The simulation follows the manufacturer's recommended operating conditions, with a tilt angle β equal to 35° and a circulation flow rate of 2 l/min per collector.

4.2.1. System Daily Performance

The physical processes in the integrated solar-MD system were studied to assess its effect on freshwater productivity and efficiency. The average water tank temperature was found to be the most important factor in determining system performance. The inlet temperature of the solar collector, and the feed temperature of the MD unit were assumed to be equal to the average water tank temperature. The mathematical model shows that increasing the inlet temperature decreases the thermal efficiency of the FPC but increases the GOR of the MD process. The initial tank temperature was set at 70°C and the feed and coolant flow rates, as well as the gap permeate circulation rate, were determined to be 5.3, 4.3, and 2.6 l/min, respectively. The daily performance of the water temperature, resultant flux, and hourly production are shown in Figure 4.9. These results are key to understand the performance of the integrated solar-MD system.



(a)



(b)

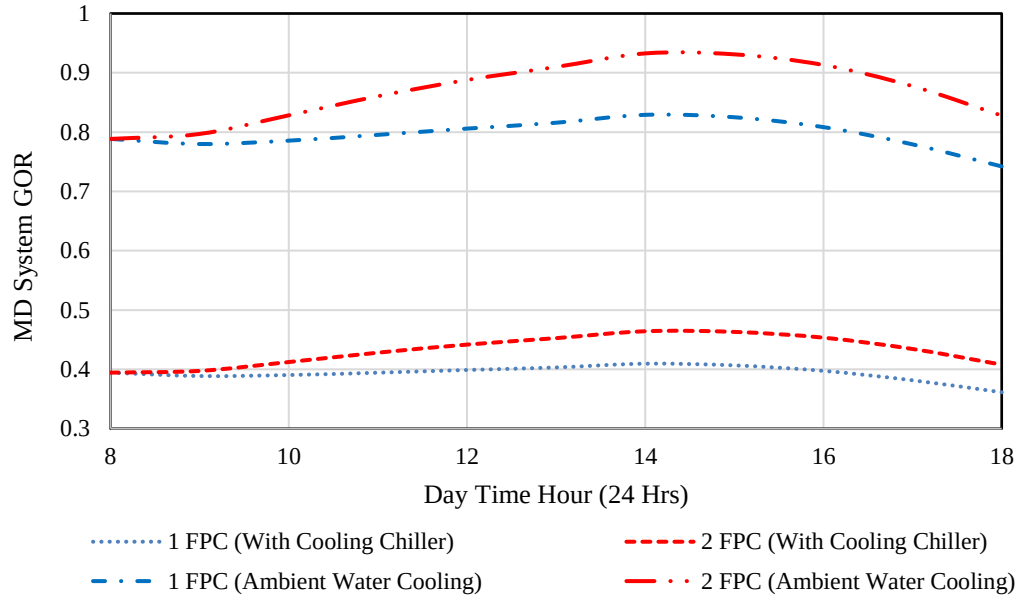
Figure 4.9 The Resultant daily performance for the integrated system representing the tank hourly temperature and the resultant flux and productivity for the studied MD system

Evaluating the daily performance for 4 scenarios impacting freshwater productivity and production cost calculation. Scenarios 1 and 2 use 1 or 2 FPCs for feed water heating with a water chiller to lower the coolant temperature to 15 °C. Scenarios 3 and 4 also use 1 or 2 FPCs but with ambient water at 22 °C as coolant. The effect of FPCs number and the cooling case on harvested solar energy and production cost are examined. Lowering coolant temperature increases productivity but the additional cost is to be justified in later sections. The number of collectors is limited to 1 or 2. This is because using 3 or more FPCs with a 160L HW storage tank raises the tank water temperature above 100°C, exceeding the MD limit. Figure 4.9 (a) shows no significant temperature difference in the storage tank temperature between using a chiller or ambient water for the same number of FPCs. Ambient water causes a slight tank temperature to increase due to a lower heat transfer coefficient. Using 1 FPC results in a slight tank temperature decrease in morning hours, increasing to a 75 °C peak at 14:00 before decreasing again. Using 2 collectors increases tank temp from 70 °C to 93 °C peak at noon. Figure 4.9 (b) shows hourly flux and production with the same trend to the tank temperature. Ambient cooling water results in a maximum flux of 109 and 165 Kg/m².hr and co-productivity of 0.74 and 1.1 l/h for 1 or 2 FPCs. Using a chiller only increases productivity by 1 or 2% for 1 or 2 FPCs.

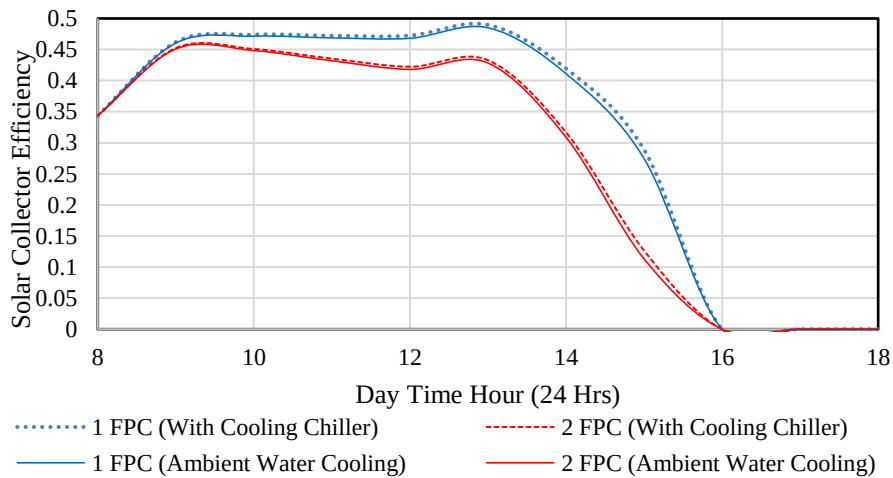
4.2.2. Energy Analysis

The energy efficiency analysis for a single stage solar powered circulating PGMD has been performed for the solar-MD subsystems separately and the results are presented in Figure 4.9. To estimate the energy performance for the integrated solar-MD system as one system, the term GOR_{STD} has been introduced and its hourly results are presented in Figure 4.12. For the four different scenarios, Figure 4.10 (a) shows that the MD system GOR follows

the same trend line as the tank's average temperature. The same trend as in Figure 4.9 (a) has been followed, a general increase in the system GOR to a peak value that coincides with the peak tank's temperature.



(a)



(b)

Figure 4.10 The resultant hourly GOR (a) and Solar Efficiency (b) for the different presented cases

Figure 4.10 (a) shows that using a water chiller for cooling reduced the resulting GOR by 50% for both 1 and 2 FPCs leading to a 50 % higher specific thermal energy consumption, as shown in Figure 4.11, with only a 5% increase in productivity as shown in Figure 4.10 (b). The results showed a maximum GOR value of 0.41 for 1 FPC with a water chiller, only increasing to 0.47 for 2 FPCs. Without a water chiller, 1 FPC had a maximum GOR of 0.82, with 2 FPCs reaching 0.92. The GOR is an efficiency indicator for thermal energy used to produce fresh water. The STEC, defined in equation (21), is another factor in energy consumption, as shown in Figure 4.11. Results showed that as the GOR decreases, the STEC increases, and vice-versa. Using 2 FPCs resulted in lower thermal energy consumption for both cooling cases, with a cooling chiller consuming twice the energy (1500 kWh/m^3) compared to ambient water (740 kWh/m^3) for 2 FPCs. As shown in Figure 4.10 (b), the solar collector's efficiency was higher at lower tank water temperatures, as higher temperatures result in more heat loss from the absorber plate to the surroundings, leading to decreased solar collector efficiency and increased MD efficiency.

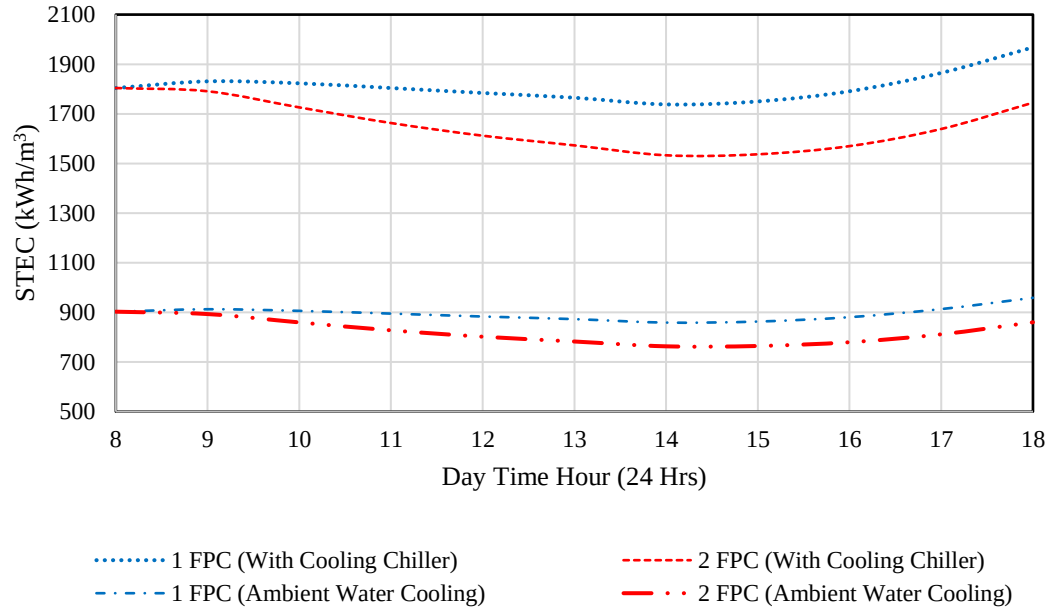


Figure 4.11 The hourly Specific Thermal Energy Consumption

Analyzing the efficiency of both systems separately could not provide a complete picture of the whole process's efficiency. Here is the role of the GOR_{STD} factor which relates the hourly used energy in the distillation process to the available solar energy. The hourly solar utilization factor is shown in Figure 4.12.

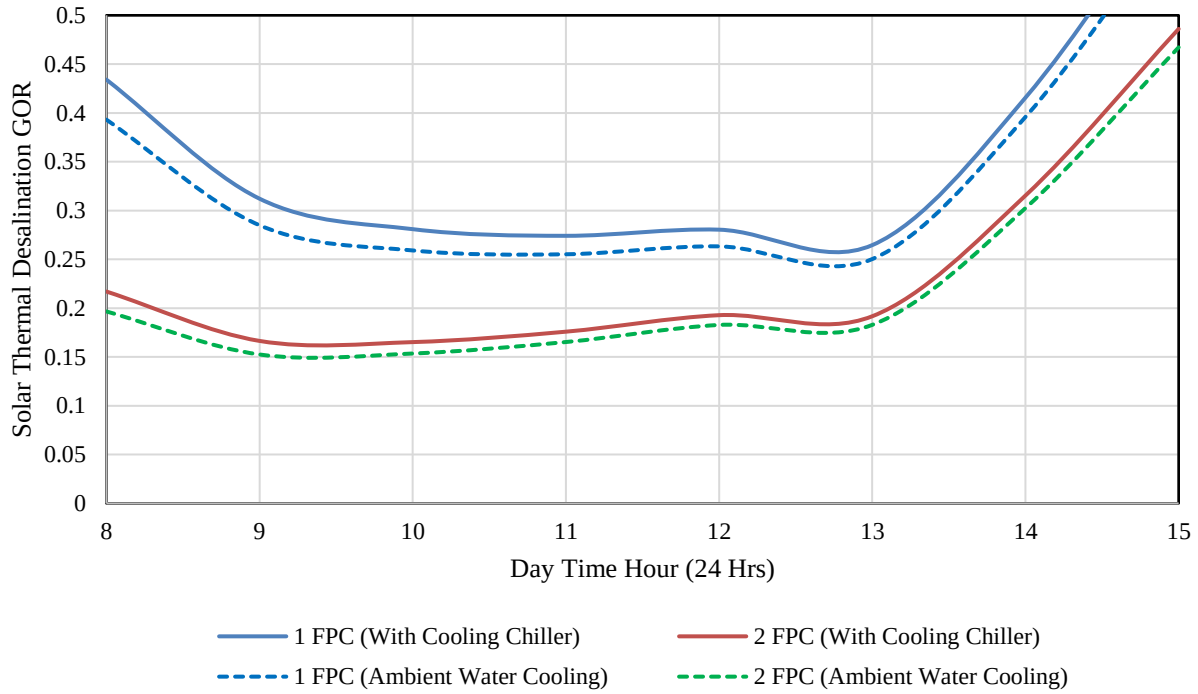


Figure 4.12 The hourly resultant GOR_{STD}

Even though the utilization of two solar FPC resulted in better performance in terms of productivity, and MD GOR, it presented lower hourly solar GOR (GOR_{STD}) because less of the collected energy is used in the instantaneous distillation process, whereas the difference between the consumed and collected energy is stored efficiently in the storage tank. This can be shown in Figure 4.10 (a and b) revealing that even though the collector presented zero efficiency from 15:00 (which means no collected energy), the temperature was maintained at adequate values above 70 °C that allowed to provide continuous operation until 19:00 which lead to a substantial increase in the GOR_{STD} . Here is the importance of storage that could compensate the intermittency of available solar energy. To avoid increasing the temperature of water in the storage tank above the boiling temperature, it could be recommended to increase the number of MD stages so that

harvested energy could be used directly in the MD process when increasing the number of FPCs.

In the previous figure, the value of the GOR_{STD} may be affected by the energy stored in the previous hours and could not give an overall picture about which system is more energy efficient. For this reason, we examined the average overall daily GOR value for the four different scenarios, and they are presented in Figure 4.13.

Our observations from the cooling chiller system show a noticeable decrease in GOR when the number of FPCs is increased from one to two - dropping from 0.1537 to 0.1286. This reduction in GOR with the addition of a second FPC could potentially signify that the additional solar energy captured by the second FPC is not effectively utilized. The reason for this could be the decreased solar FPC efficiency, Figure 4.10, when the temperature increased in the storage tank when implementing 2 FPCs. Similarly, the findings from the ambient water-cooling system also reveal a decrease in GOR as the number of FPCs is increased from one to two, diminishing from 0.2880 to 0.1959. However, it is noteworthy that despite the decrease in GOR with an additional FPC, this system, under both configurations, exhibits a higher GOR in comparison to the cooling chiller system. This implies superior performance and overall efficiency of the ambient water-cooling system in the provided operational circumstances.

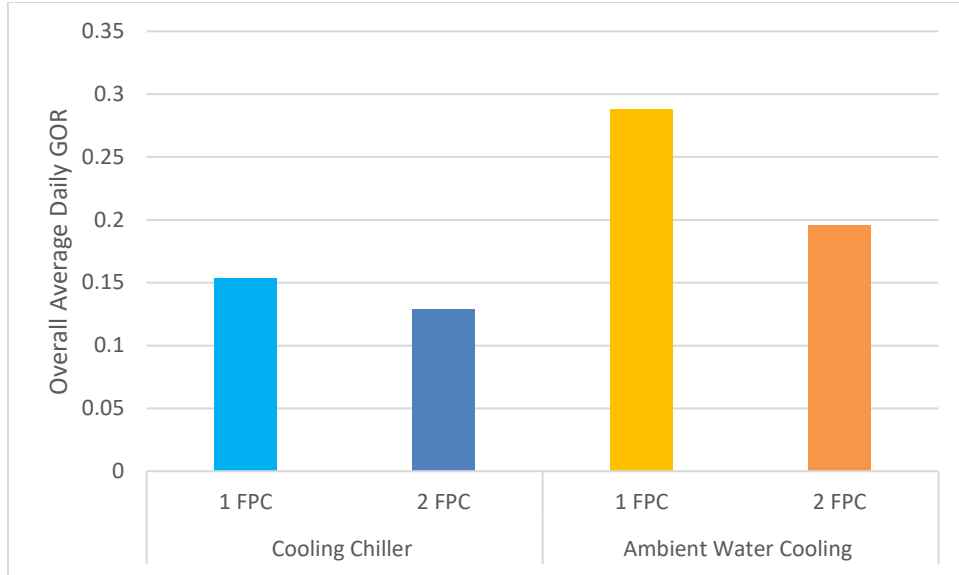


Figure 4.13 The overall average GOR for the four different scenarios

4.2.3. Cost Analysis of the Solar Powered MD

The analysis of Figure 4.14 revealed that increasing the number of collectors resulted in a decrease in the water production cost. The capital cost breakdown is presented in Table 4.1 and it is annualized for 20 years with an interest rate (i) that is assumed to be 5% [157] [158]. Other operational costs are considered as follows:

- the cost of the intake of seawater is zero [159].
- The membrane is replaced once a year.
- The chiller has a coefficient of performance (COP) of 2.5 and the electric cost is equal to 0.045 USD/kWh.
- The maintenance cost is 1.5% of the capital cost of the system [158].

Table 4.1 Capital cost breakdown for single stage circulating WGMD

Item	Cost (USD)	Justification
FPC (USD/m²)	100	[160]
Storage Tank	200	Quoted from Local Market
Pumping and Connections	200	Quoted from Local Market
Chiller	300	Quoted from Local Market
System Instrumentation	500	Previous Experience
MD-Module	100	Manufactured at our Labs
Single Membrane Cost	10	Supplied to our Labs

The cost analysis for the four different scenarios has been performed, and the results are presented in Figure 4.14. The highest cost was observed for the case of using only one FPC with a value of 0.068 US\$/liter when using a coolant chiller, however, using ambient water cooling, provided 35 % less of the production cost for a value of 0.044 US\$/liter. Increasing the number of FPC to two has consequently increased the amount of energy harvested from the sun which in turn increased the water productivity resulting in a decrease in the production cost. A maximum of 13% decrease in the cost have been achieved to obtain the minimal cost of 0.046 US\$/liter for using two FPC for using a cooling chiller and 0.038 US\$/liter for cooling with ambient water. The cost analysis claimed that omitting the usage of a water chiller has a substantial impact on decreasing the water production cost because of the high energy consumption required for vapor condensation.

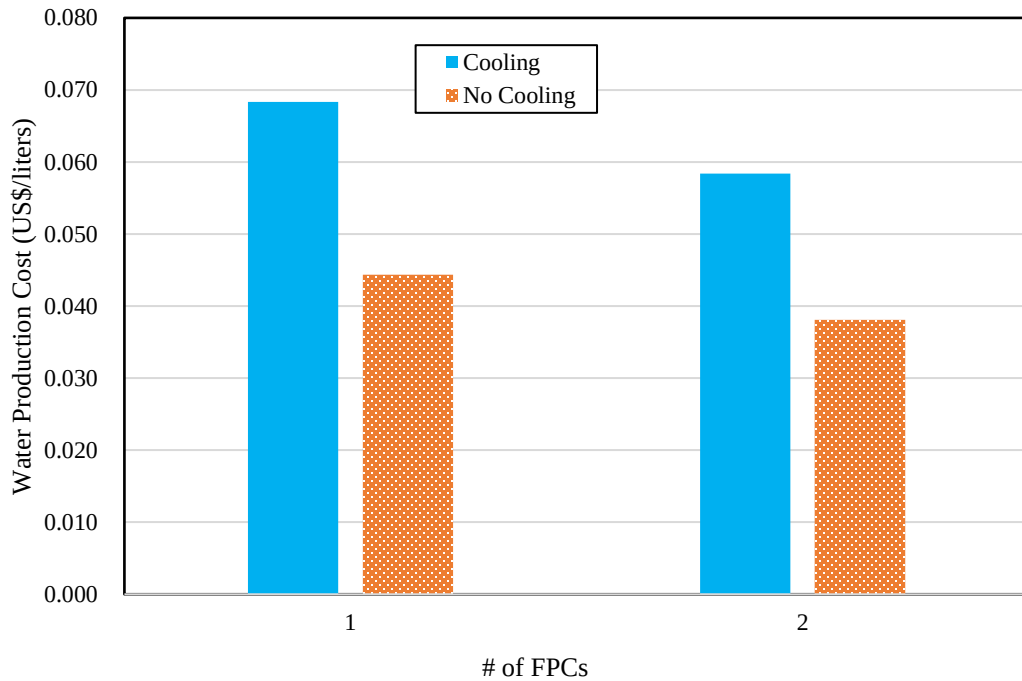
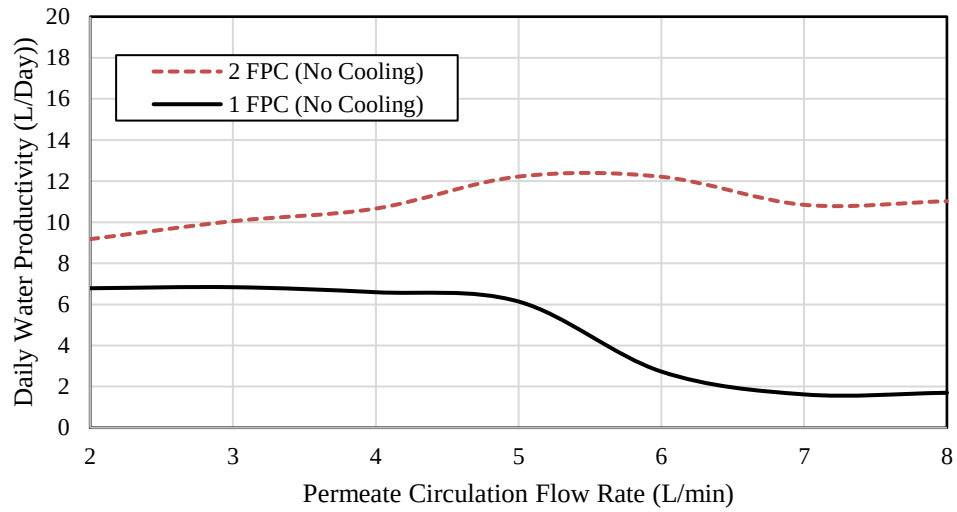


Figure 4.14 The overall Production Cost Based on the Previous Operation Parameters and Performance

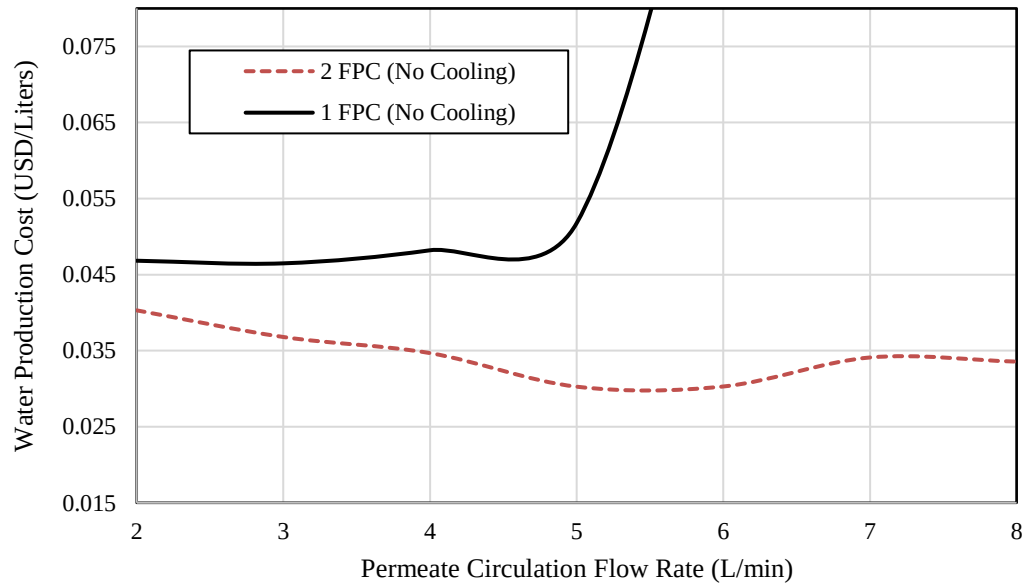
4.2.4. Sensitivity Analysis

This section analyzes the impact of various operating and design parameters on yearly water production and unit product cost. The prior section analyzed the different operating conditions such as feed and coolant flow rates, and gap permeate circulation rates. The results compared the impact of system design parameters like the number of FPCs and cooling medium on daily performance, yearly production, and overall cost. The tank temperature (and therefore the feed temperature) was found to be an uncontrolled parameter influenced by design, solar radiation, and weather, unlike a typical MD process where feed temperature can be controlled. Figures 4.15 and 4.16 examine the impact of different operating parameters such as feed flow rates and gap permeate circulation rates on daily productivity and average production cost, considering ambient water cooling only because using a cooling chiller resulted in a 50% rise in production cost for just a minor

improvement in water productivity, making it an unfeasible scenario for different FPC numbers.



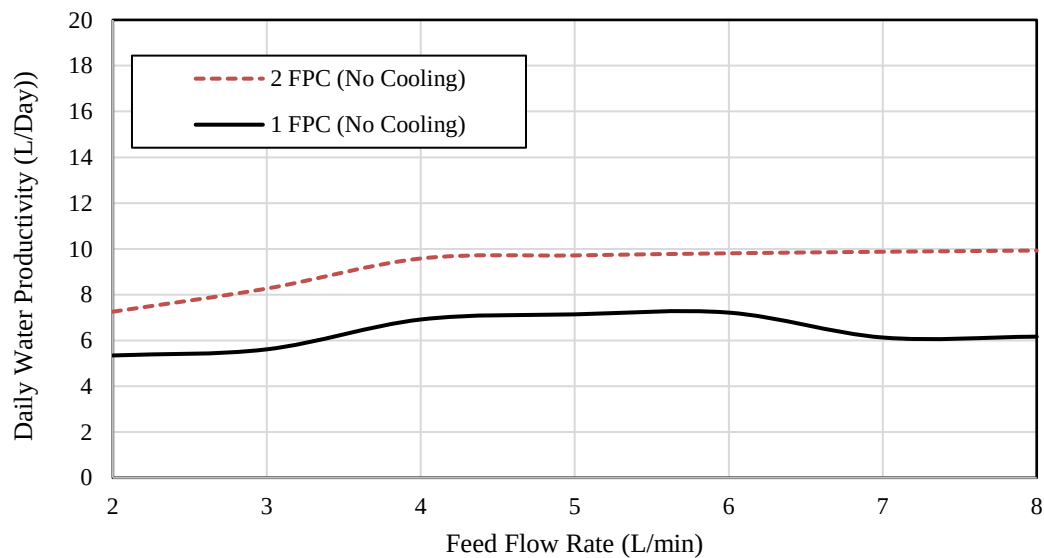
(a)



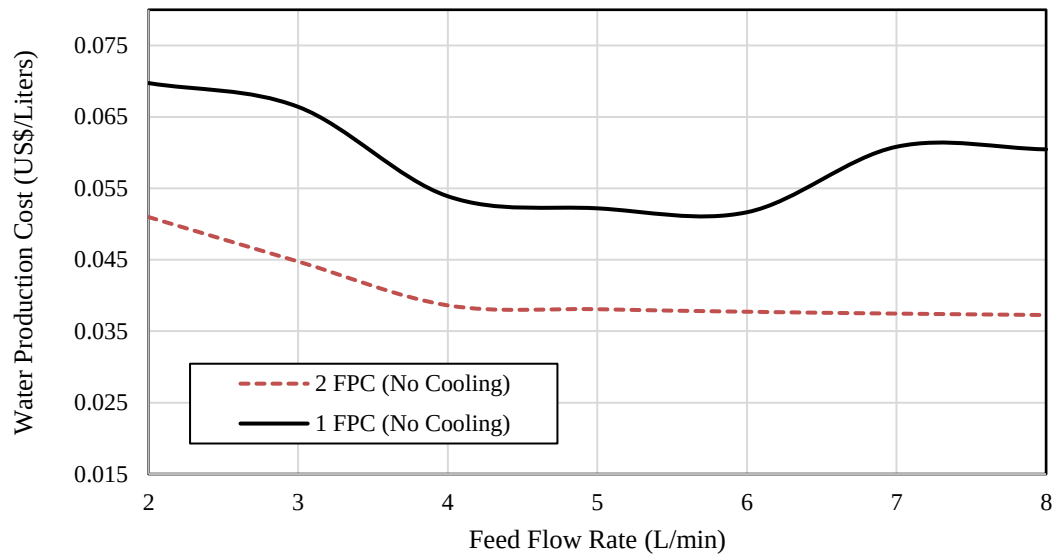
(b)

Figure 4.15 The resultant daily production (a), and the yearly water production cost (b), for the variation of the permeate circulation flow rate at feed flow rate = 5.3 l/min and coolant flow rate = 4.3 l/min

Figure 4.15 analysis compared two FPCs in an MD system (5.6 and 4.6 l/min feed/coolant flow) under similar conditions as Figure 4.9. Results showed that two FPCs improved daily production and reduced cost, with a gap permeate circulation rate of 5.5 l/min maximizing productivity (12.2 l/day) and reducing cost by 33% (Figure 4.15 (a) and (b)). Increasing the gap permeate circulation rate improved heat transfer and prevented storage tank temperature from rising above boiling, but decreased distillation efficiency. A feed flow rate of 4 l/min was optimal for two FPCs, while an optimum range of 4-6 l/min was observed for one FPC. The cost was inversely proportional to the production rate with a 25% increase for two FPCs. A comparison of Figure 4.15 and 4.16 showed that increasing the gap permeate circulation rate was more effective in water production than the feed flow rate, as it controls heat/mass transfer and results in higher heat transfer coefficients with the maximum productivity of 12.2 l/day was achieved by increasing gap permeate circulation rate.



(a)



(b)

Figure 4.16 The resultant daily production (a), and the yearly water production cost (b), for the variation of the feed flow rate at permeate gap permeate circulation rate = 2.3 l/min and coolant flow = 4.6 l/min

To study the effect of different design parameters such as the number of MD stages and the number of FPCs on the yearly water production and the water production cost, an analysis has been performed and presented in Figure 4.17. The analysis has been performed at a feed flow of 5.3 l/min, a permeate gap circulation rate of 2.3 l/min, and a coolant flow of 4.6 l/min. The analysis has been performed to maximize water production from all stages by increasing the number of FPCs. For instance, to maximize the production for 1 to 8 stages, the optimum number of FPCs was found to be 2, 3, 4, 5, 7, 8, 9, and 10 FPCs respectively. This number of FPCs was found to maximize water production without exceeding the boiling point temperature of water in the storage tank. The results in Figure 4.17 revealed that increasing the number of stages to 8 provided a linear increase in water production by 500% leading to a 35 % exponential decrease in the water production cost that decays to 0.025 US\$/L.

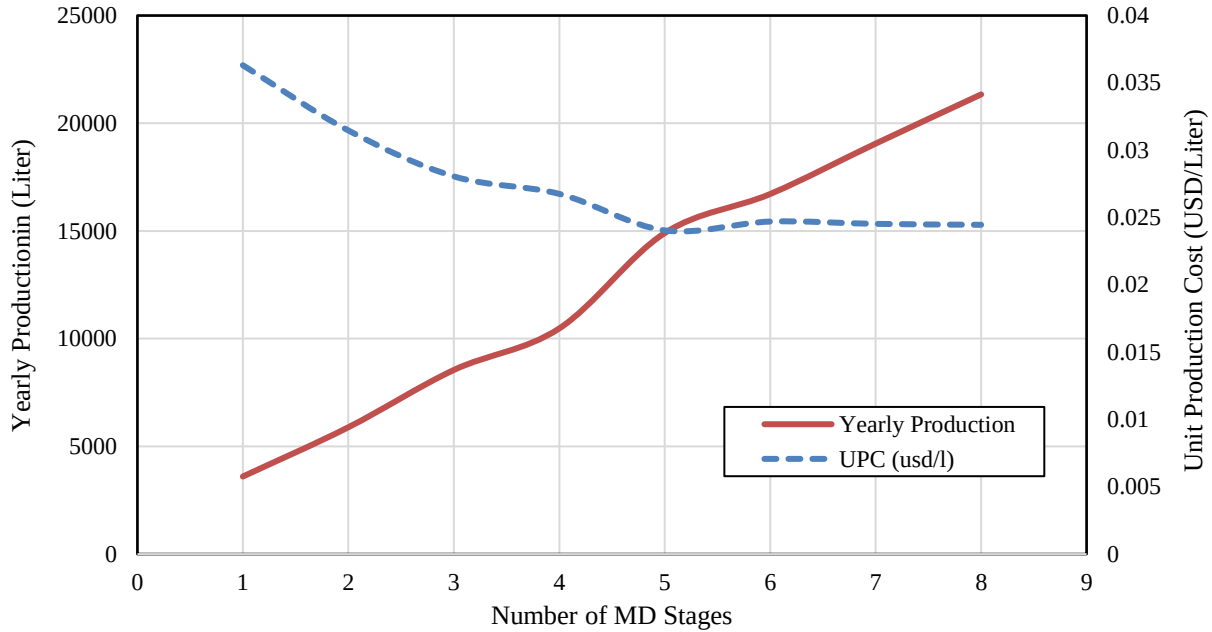


Figure 4.17 The Effect of MD Number of Stages on the Yearly Water Production and the Water Production Cost at a feed flow=5.3 l/min, a gap permeate circulation rate = 2.3 l/min, and a coolant flow = 4.6 l/min

4.2.5. Design of a Pilot MD System

This section aims to evaluate the techno-economic performance of a solar circulating PGMD installed in Dharan City, to serve a small community. The system consists of 30 MD stages and operated at 5.6, 4.6, and 2.3 l/min/stage flow rates for the feed, coolant, and gap permeate circulation respectively, with an 8 mm gap width. The storage tank is selected to have a capacity of 500 Liters.

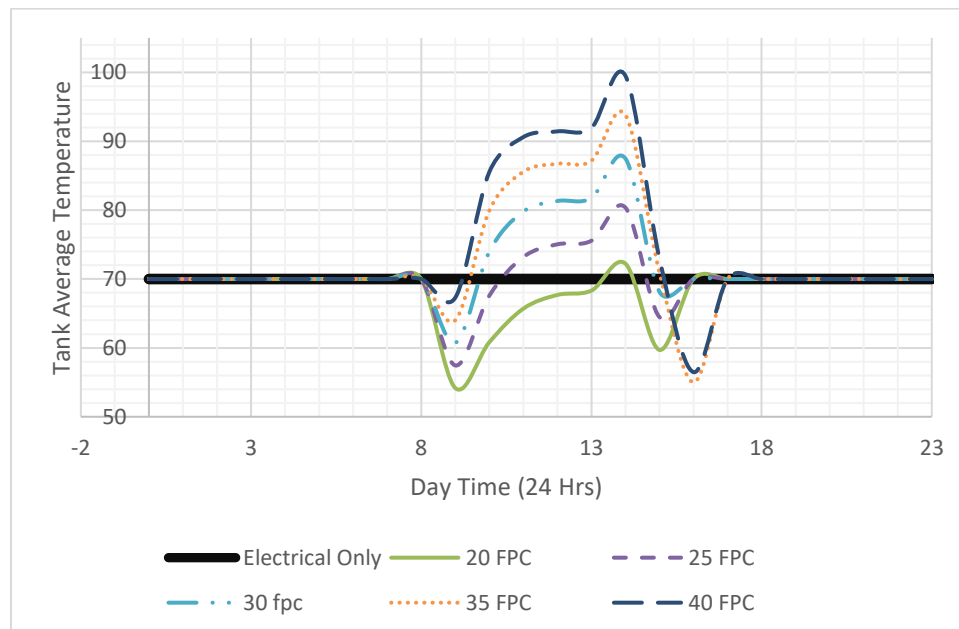
at Three scenarios were considered:

- Electric power only.
- Solar thermal only.
- Hybrid solar-electrical power.

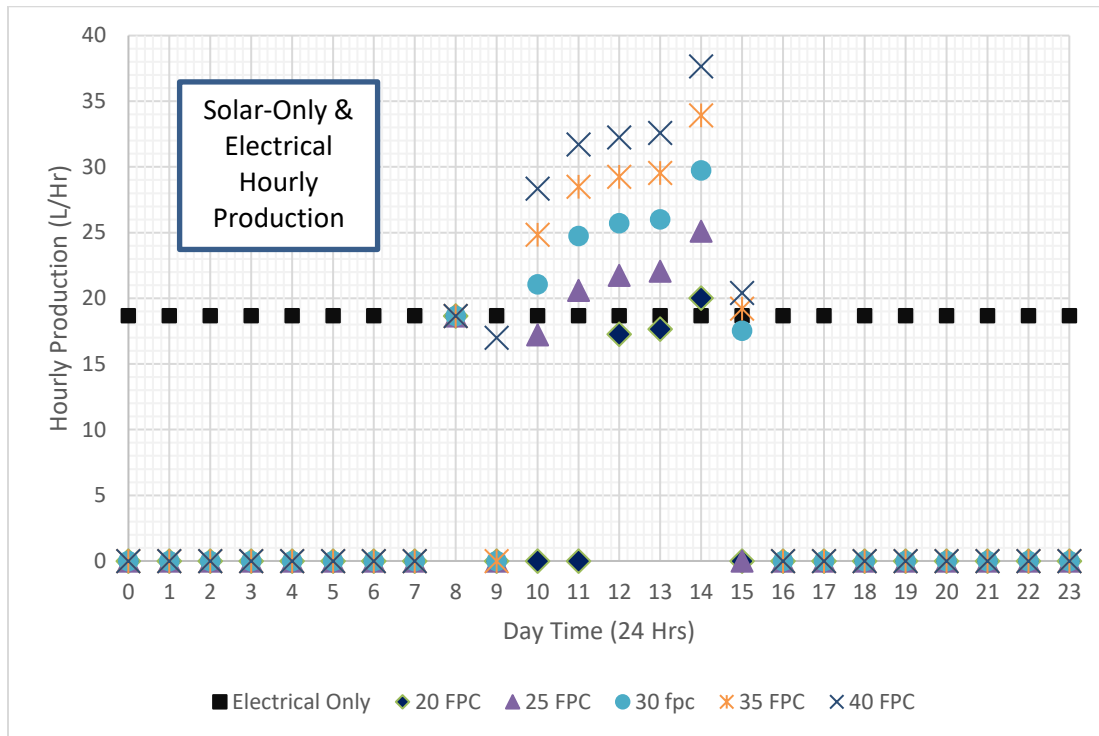
The electrical scenario is considered as the base case scenario and it is operated at a constant temperature of 70 °C (Figure 4.18, a) for the whole 24 hours with corresponding

production of 19 L/hr (Figure 4.18, b), providing 445 Liters per day as an overall daily production.

For the Solar-only scenario, the system starts production at 8:00 with a starting temperature of 70 °C and a corresponding hourly production of 19 L/hrs. In this scenario, there is no assistance from the electrical heater, and therefore, the average tank temperature is only affected by the energy balance across the tank control volume. To illustrate, the tank water temperature decreases when additional energy is supplied to the tank than that lost or transferred to provide heat for the MD-System. This explains the observation of Figure 4.18 (a) presenting a drop in the temperature between 8:00-9:00 morning because of the low available solar intensity. In this scenario, the solar production is only considered when the tank temperature is around 70 °C or higher as shown in Figure 4.18 (a) and its hourly production reflection in Figure 4.18 (b). It is shown that for a system of 30 stages it is required at least 25 solar FPCs to keep the average temperature above 70 °C in daytime to provide adequate water production rates.



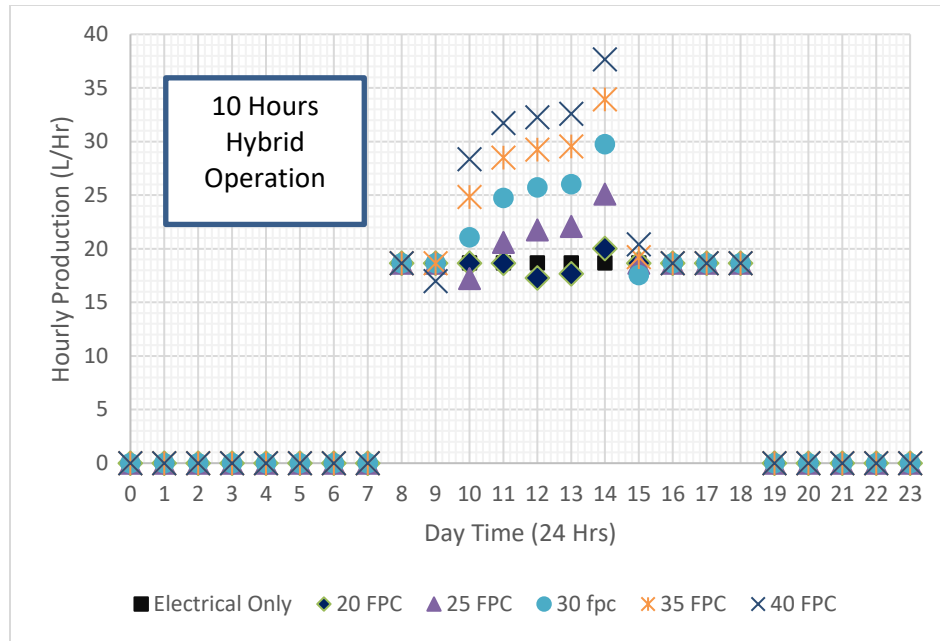
(a)



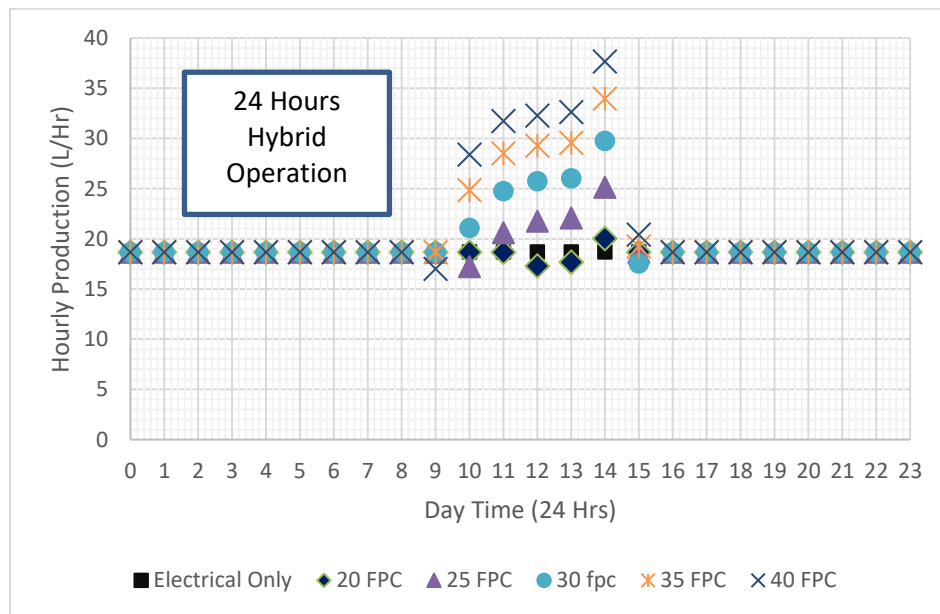
(b)

Figure 4.18 The average tank temperature (a) and the corresponding productivity (b) for the electrical only and solar only scenarios

In the third hybrid solar-electrical scenario, an electric water heater will assist the water production once the tank temperature drops below 70 °C. For this scenario two cases would be considered for the total production, the 24 Hours production, and the 10 hours production from 8:00 to 18:00. Their hourly production is presented in Figure 58 (a) for 10 hours operation and (b) for 24 hours operation.



(a)

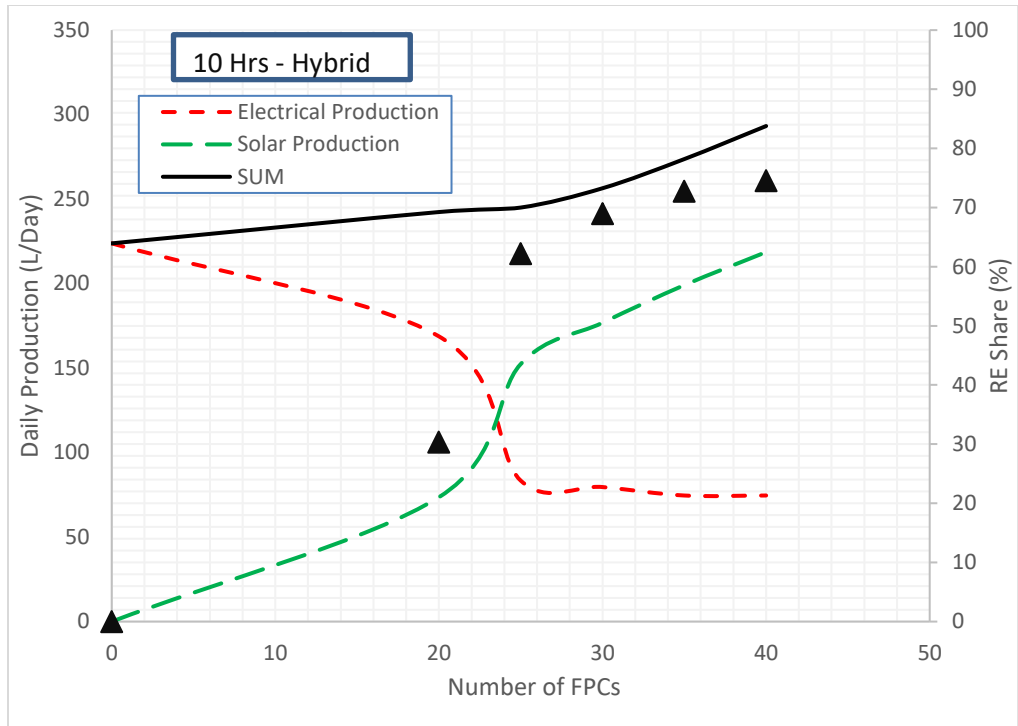


(b)

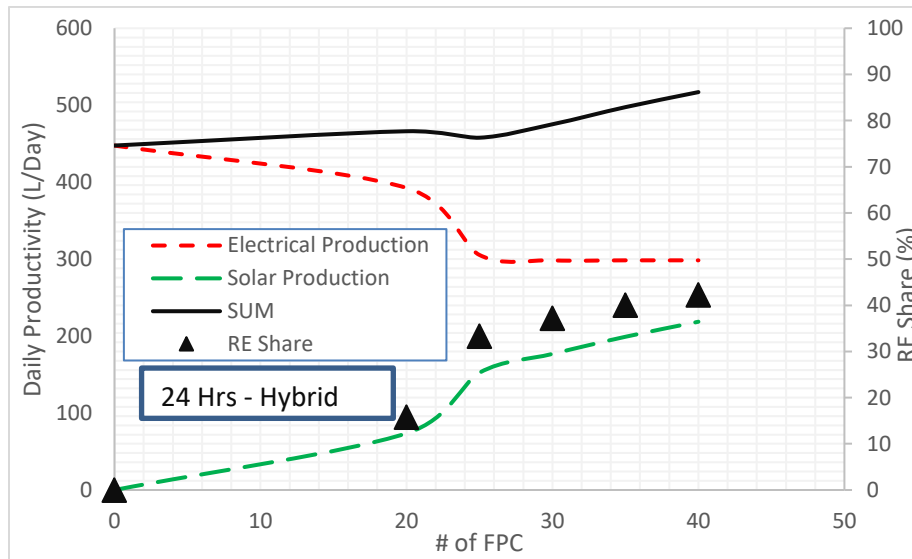
Figure 4.19 The hourly production profile for hybrid operation based on 10 hours operation (a), and 24 hours operation (b)

In Figure 4.20, the solar daily production is presented as a sum of the production in different hours for different numbers of FPCs, from the Figure 4.18. It is found that using 40 FPCs with a 500 L hot water storage tank and 30 PGMD stages, the system produced a maximum of 220 L/day from solar only. It is also found that a sharp increase in the solar productivity has been found when the number of FPCs has been increased from 20 to 25 which provided an average tank temperature above 70 °C for a longer period in day time compared to 20 FPCs. From 20 collectors and less the production is intermitted as shown in Figure 4.18. To solve this intermittency, the hybrid operation has been proposed. In this scenario, an electric water heater assists the solar production to provide continuous production for two different cases: continuous production for 10 hours duration (8:00-18:00) and another case for 24 hours production (0:00 to 23:00). The daily performance hybrid electrical- solar presented by the daily productivity is presented in Figure 4.20 for 10 hours operation (8:00 – 18:00) in Figure 4.20 (a) and for 24 hours operations in Figure 4.20 (b). The productivity of a hybrid solar-electrical system was found to be influenced significantly by the number of FPCs used, with data showing a positive correlation between FPC count and solar energy production. During the 10-hour operation, the system yielded only electrical energy production (223.8 L/day) with no FPCs, whereas using 40 FPCs maximized solar production to 218.64 L/day, resulting in a total energy production of 293.24 L/day. Intriguingly, a marked increase in solar production was observed when the number of FPCs was augmented from 20 to 25, maintaining an average tank temperature above 70 °C for an extended period, also when increasing the number of FPCs to 40 a rise in renewable energy (RE) share from to 74.56% has been observed. In contrast, the 24-hour operation increased the total energy production to a peak of 517.04 L/day with 40 FPCs but resulted

in a lower RE share, peaking at 42.29%. Thus, while the total energy production grew in the 24-hour operation, the proportion of renewable energy was less than that in the 10-hour operation. This indicates that the hybrid solar-electrical approach offers a solution to the intermittency problem seen at lower FPC counts and underscores the necessity to consider operating hours while designing these systems to optimize the RE share. For those different scenarios, the cost analyses have been studied to determine the most economic case. The results are provided in Figure 4.21.



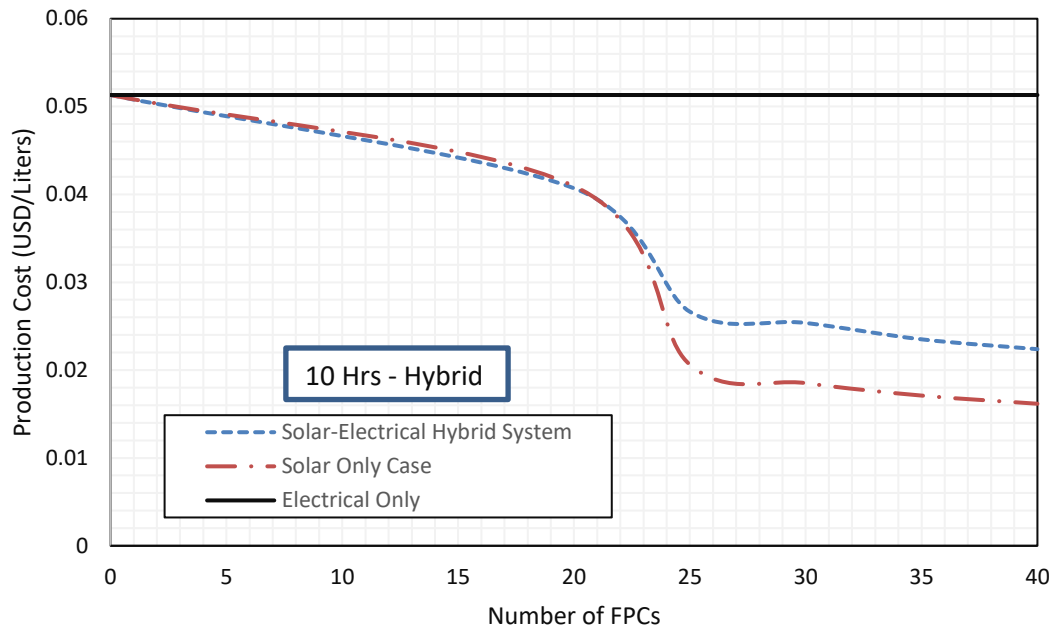
(a)



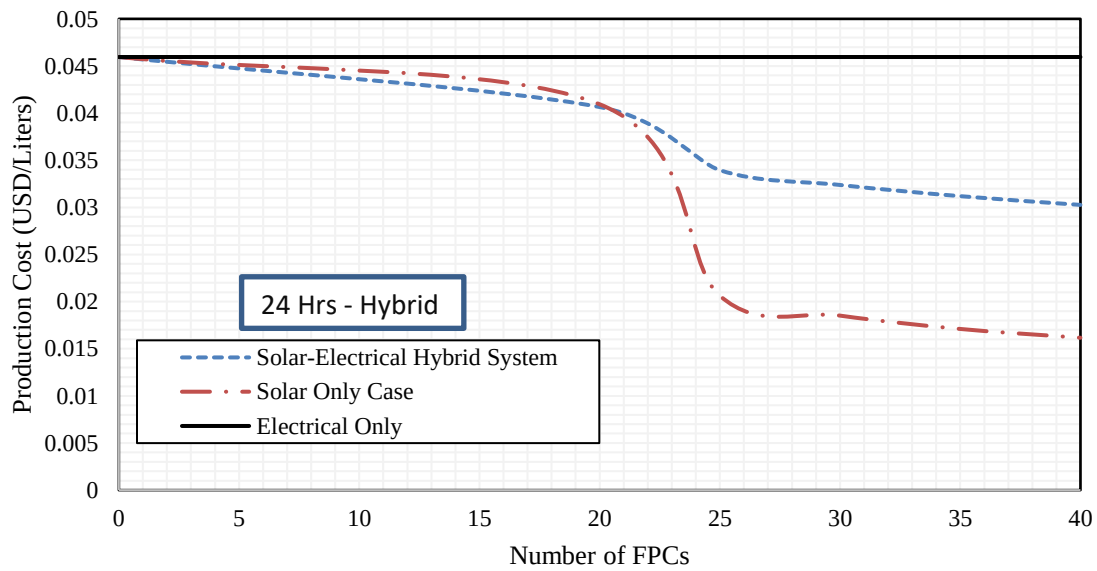
(b)

Figure 4.20 The Daily Electric, Solar, and Net Total Production for Different Numbers of FPCs, for the case of solar and hybrid solar-electrical powering for 12 hours (a) and 24 hours (b) production

The results of the economic assessment for all three cases are presented in Figure 4.20. The analyses have been performed with the same economic assumptions and estimations stated in section 4.3.3 except that for the solar system, the capital cost for the storage tank, and the pumping connections has been upgraded to 600, and 2500 USD respectively. The economic results have been presented in Figure 4.21. The results revealed the solar-only scenario presented the lower cost for the highest number of FPCs with a value of 0.016 US\$/Liters when using 40 FPCs. In the 10-hour operation of the hybrid solar-electrical system, the cost per liter decreased from US\$0.0513/L with no FPCs to US\$0.0224/L with 40 FPCs. The hybrid system became more cost-effective than the solely electrical system (constant at US\$0.0513/L) when more than 20 FPCs were implemented. Similarly, for the 24-hour operation, the cost per liter for the hybrid system also showed a reduction from US\$0.0459/L to US\$0.0303/L with an increase in FPCs, again outperforming the solely electrical system (US\$0.0459/L) when more than 20 FPCs were employed. Therefore, a hybrid system employing a higher number of FPCs, such as 40, proves to be the most cost-efficient, yielding the lowest cost per liter in both 10-hour and 24-hour operations. However, the operation period should be contingent upon specific needs, with the 10-hour operation being more cost-effective and the 24-hour operation yielding higher quantity of water product.



(a)



(b)

Figure 4.21 The average Water Production Cost for electric, Hybrid, and Solar-only Pilot Scale MD Systems for 10 hours (a) and 24 hours (b) operations

CHAPTER 5

EXPERIMENTAL SETUP & ANALYSIS

This study aims to experimentally investigate the production and energy performance of a circulating PGMD. In this section the system is technically described from the design to manufacturing, and installation process. The experimental procedures are well presented with all the control instrumentation equipment, as well as the experimental error.

5.1. System Description for the Single Stage Circulating PGMD

The unit under investigation, presented in Figure 5.1, is a single-stage permeate water gap membrane distillation (MD) system. This intricate system is designed to compare the performance parameters such as water flux, GOR, STEC for both stagnant and circulating water permeate gap. The system primarily comprises three channels: feed, permeate gap, and coolant. Each of these channels fulfills a unique purpose in the MD process.

In the experimental unit, the feed salty water, with 150 PPM salinity, is heated via an external heater and consistently fed into the feed channel by a circulation pump. This hot feedwater is passed over the hydrophobic membrane allowing only the vapor cross to the permeate gap channel, the core of the system, to be condensed into distilled water.

The distilled water, now referred to as the permeate, is collected outside the gap directly when no circulation provided. However, in case of circulation, a small tank connected to the permeate gap channel with a circulation pump and the excess of water is connected outside the tank as an overflow.

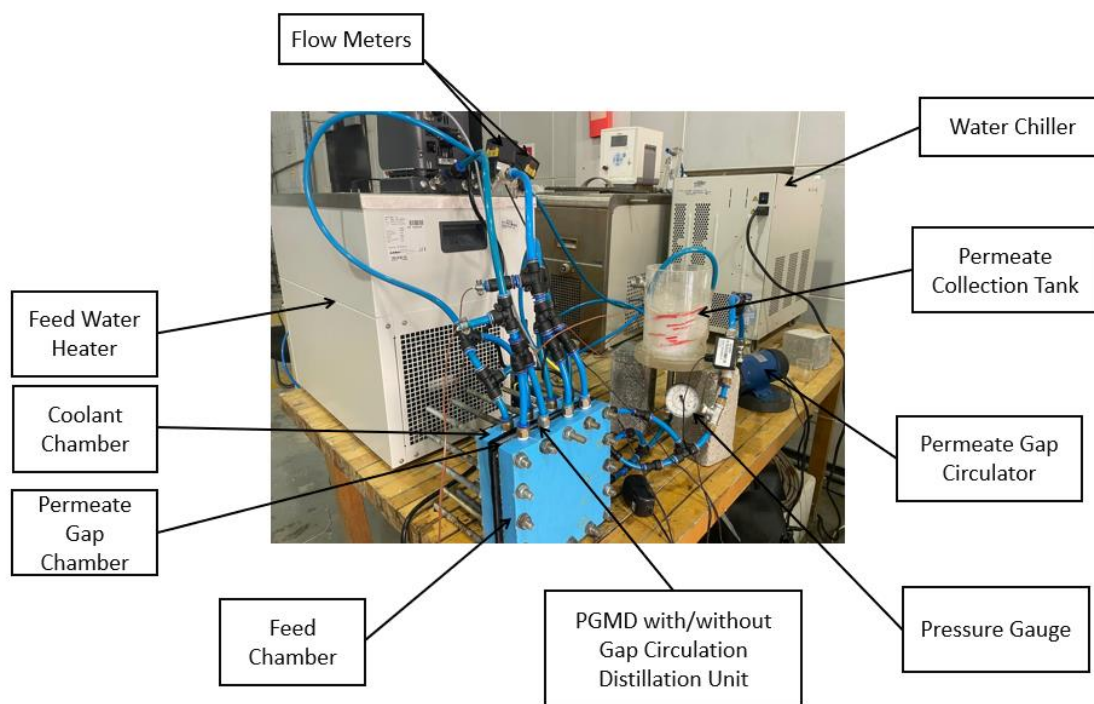
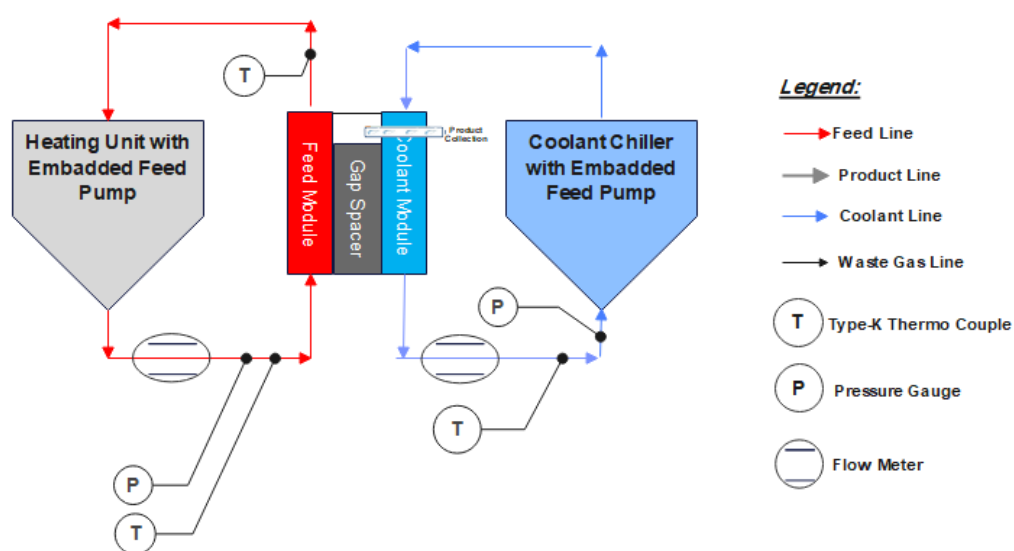
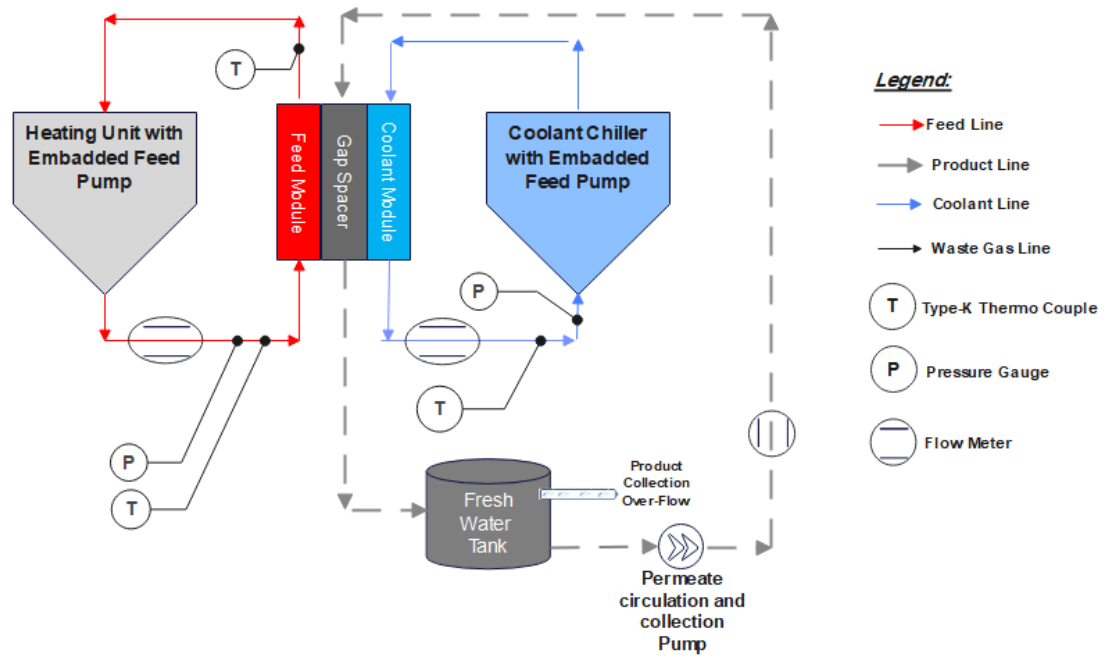


Figure 5.1 Experimental set-up of a single stage PGMD

The experimental schematic diagrams are presented in Figure 5.2 (a) for the system without permeate circulation and in Figure 5.2 (b) for the system with circulation.



(a)



(b)

Figure 5.2 Schematic diagram for the PGMD with (a) and without (b) gap water circulation

The coolant channel's purpose is to maintain the cold side of the membrane, thereby driving the thermal gradient necessary for distillation. The coolant is cooled using a water chiller and a pump ensures its continuous circulation.

To evaluate the energy performance of the MD unit, Type-K thermocouples are installed at the inlet and exit of the feed and coolant modules for calculating the STEC and the GOR of the unit using equation 19, 20, and 21. Where the $\dot{Q}_{Heating,MD}$, and $\dot{Q}_{Cooling,MD}$ are calculated through the well equation $\dot{Q} = \dot{m} * C_p * \Delta T$. Where $\dot{m}$ represents the feed and coolant flow rates with ΔT is the temperature difference of the water stream between the inlet and exit of each module.

Furthermore, digital flow meters and pressure gauges are integrated into the system to control and monitor the pressure on the membrane. This pressure control is vital for the stable operation of the MD system, ensuring the right balance between maximizing the distillate output and avoiding membrane damage due to excessive pressure.

5.2. MD Unit Design

The MD distillation module consists of feed, coolant, and permeate chambers that have the same design and are presented in Figure 5.3. They are made of high-density polystyrene (HDPE), which is well-known for its low thermal conductivity that reduces the heat losses to the surrounding and high melting point [161]. They all have the same simple design that allows water to be injected from one channel side and to exit from the other channel side. However, the feed and coolant modules have a thickness of 15 mm, whereas the circulating gap spacer has a thickness of 10 mm.

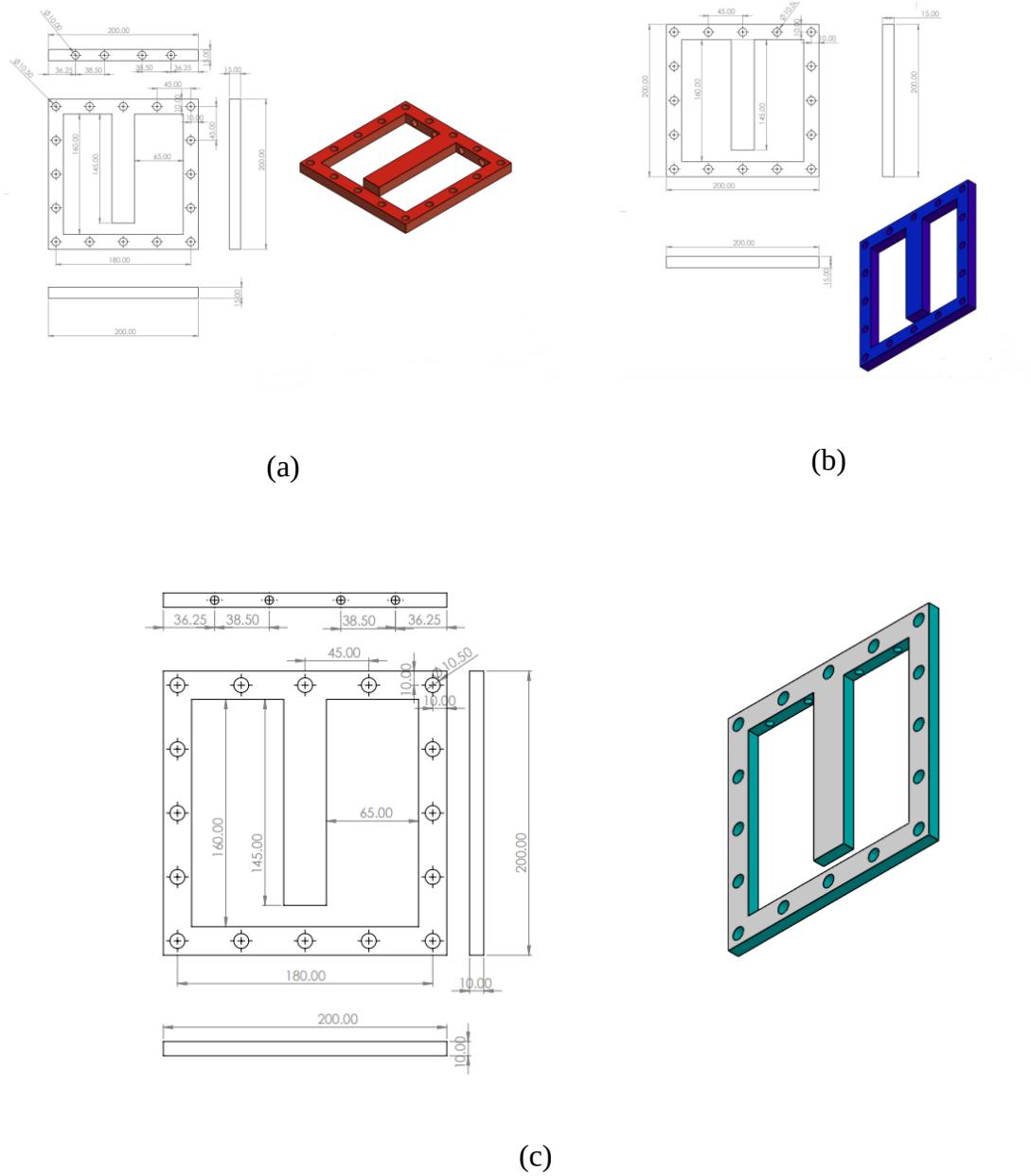


Figure 5.3 Feed (a), Coolant (b), and Gap (c) Chambers

The MD unit assembly is shown in Figure 5.4. It contains, beside the three channels, a hydrophobic membrane with an effective transport area of 0.007937 m^2 acts as a barrier to separate the feed and permeate water allowing only vapor to cross to the condensation gap side. The water gap acts as a separation layer between the feed and coolant channel to allow permeate vapor to condense. The condensation occurs on a 0.0256 m^2 solid brass

condensation plate, with a thickness of 1.5 mm. A 1.5mm brass support plate, which is perforated, provides support to the membrane, and prevents it from buckling or bending. The permeate water circulates within the gaps, allowing for maximum heat transfer and increased water vapor production. This results in an efficient and effective distillation process, ensuring high-quality purified water.

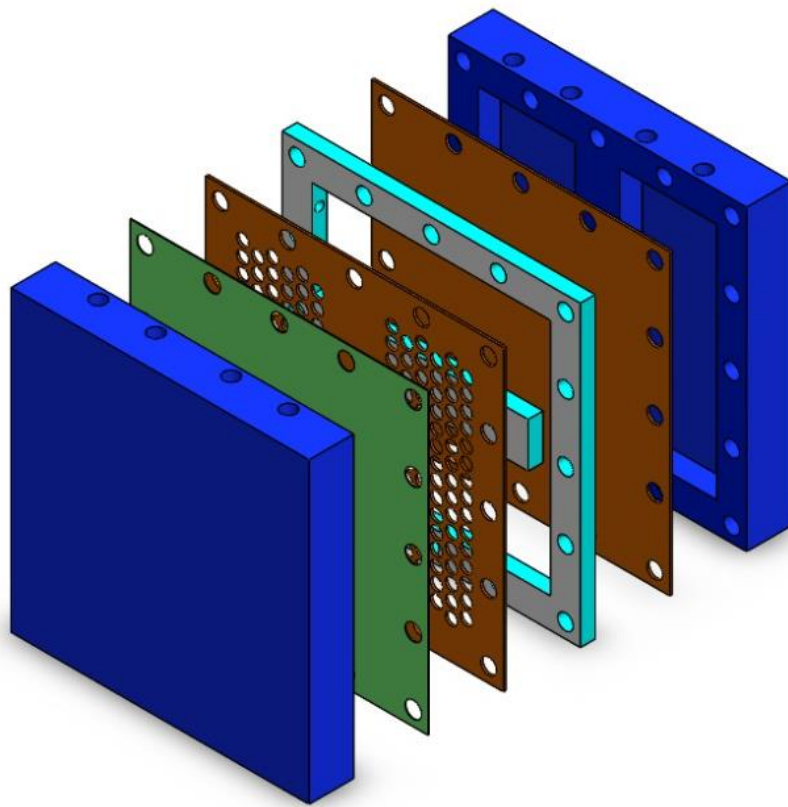


Figure 5.4 Assembly of the single stage circulating PGMD

5.3. Experimental Plan

The objective of the experiment is to evaluate the performance of a circulating water gap membrane distillation configuration using a single type of membrane. Hydrophobic membranes are designed to repel water and allow only vapor to pass through, which makes

them useful for separating and concentrating solutions in the MD process. In this experiment a PTFE membrane, Table 5.1, is used with its impressive specifications - including a pore size of 0.45 micrometer for a higher flux, a contact angle of 140 degrees to prohibit the entry of high salt water, and a porosity of 80%.

Table 5.1 Membrane characteristics

Active layer thickness (μm)	6.9 ± 2.0
Support layer thickness (μm)	141.4 ± 15.8
The thickness of the membrane (μm)	153.9 ± 13.6
D_p (μm)	0.45
ε (%)	79.7 ± 8.7
Contact Angle ($^\circ$) of the active layer	139.0 ± 2.8
Contact Angle ($^\circ$) of the support layer	119.3 ± 1.0

Saline water with 150 PPM is used as the feedwater for the experiment. The water is heated to the desired operating temperature before entering the feed module.

The experiment is conducted be conducted under a range of operating conditions, The cooling stream is fixed at a flow rate of 2 lpm and a temperature of 15 $^\circ\text{C}$ whereas the feedwater temperatures, feed flow rates, and gap circulation rates are varied for both stagnant and circulating water gap as follows:

- Feedwater temperature (from 50 $^\circ\text{C}$ to 80 $^\circ\text{C}$)
- Feedwater flowrates (from 1 lpm to 3 lpm)
- Gap water circulation rate (from 0 to 3 lpm)

The data collected are to be analyzed to determine the effects of the operating conditions on the performance of the water gap membrane distillation configuration. The analyses are to be performed to compare the performance of circulation compared to the stagnant water at different conditions.

5.4. Experimental Results and Discussions

5.4.1. Effect of Feed Water Temperature

The permeate flux is a frequently utilized measurement for assessing the effectiveness of the Membrane Distillation (MD) procedure. It signifies the quantity of freshwater derived from the MD setup per unit membrane area every hour. The disparity in temperature between the cold and hot flows acts as the main impetus for vapor transition through the membrane. Hence it is crucial to investigate the effect of different feed temperature on the resultant flux and compare it with the results discussed on the theoretical investigation in chapter 4. The effect of the feed temperature on the flux is presented in Figure 5.5.

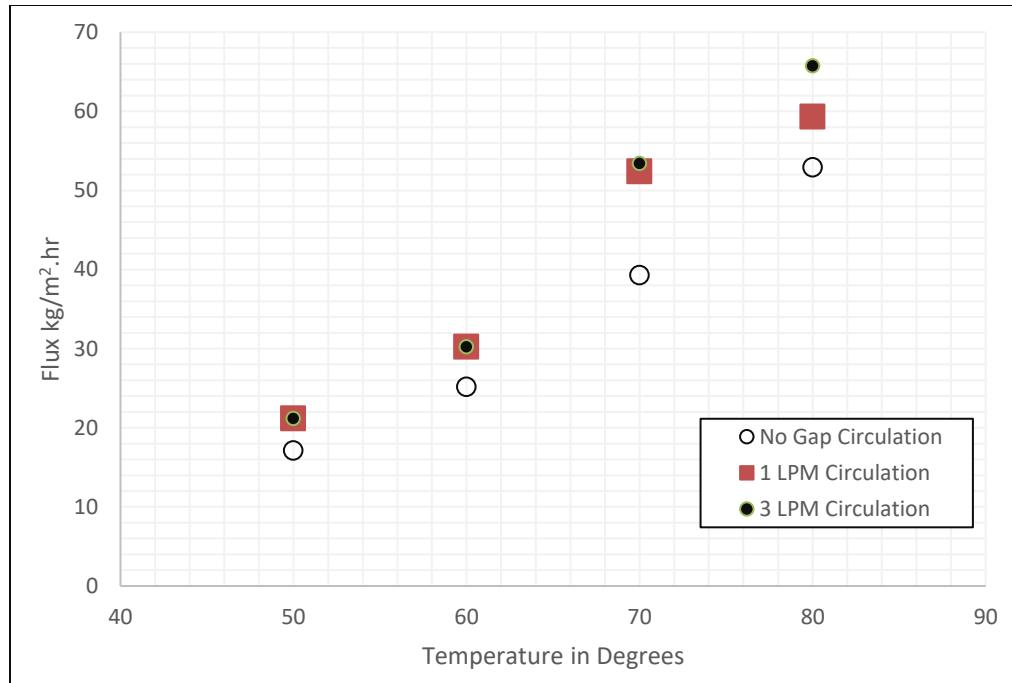
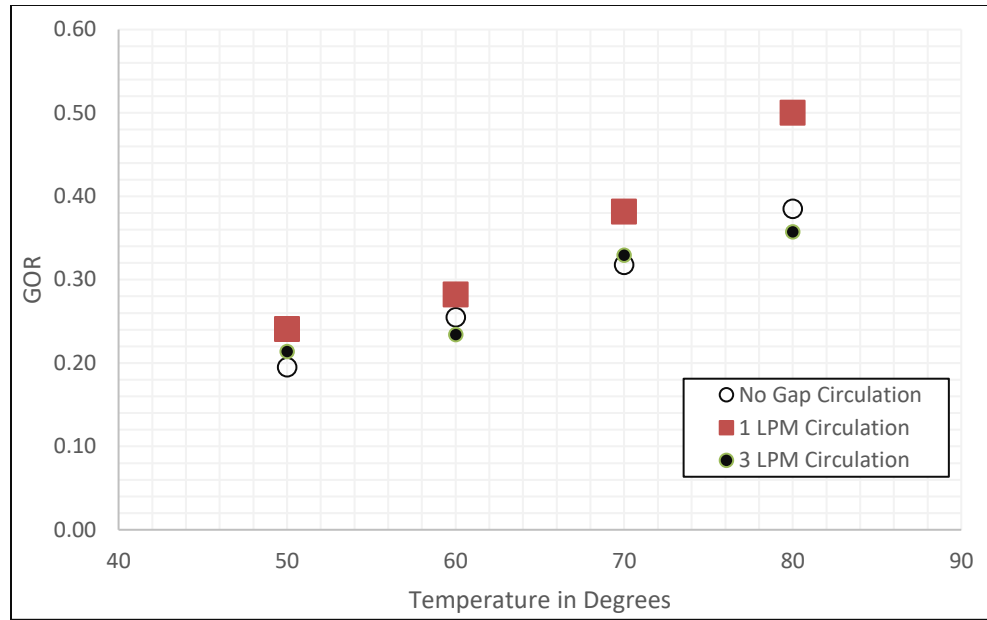


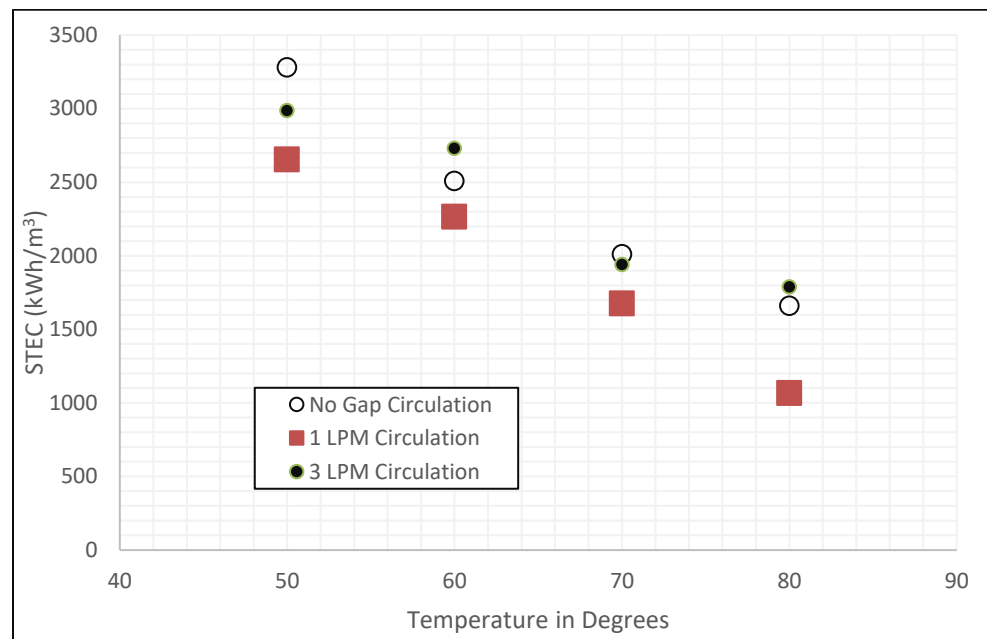
Figure 5.5 Effect of feedwater temperature on the resultant flux (in kg/m².hr) at a feed and coolant flow rates of 2 l/min and coolant temperature of 15 °C

The result in Figure 5.5 shows similar trend of the theoretical results presented in Figure 4.1. The experimental results, Figure 5.5, show that for the stagnant water gap and for different gap circulation rate an increase in the flux is observed at higher feed temperature. For the stagnant water gap, a 200% increase in the flux is observed when increasing the feed temperature from 50 °C to 80 °C with a flux of 54 kg/m².hr. At lower feed temperatures of 50 and 60 °C, providing a small gap circulation rate of such that of 1 l/min provided an increase in the flux with 25 % and 20 % for 50 and 60 °C respectively. However, any increase in the in the gap circulation rate beyond this value was shown to be ineffective without any remarkable increase in the flux, and this conforms with Figure 4.2 (b) that showed the higher percentage when using a low gap circulation rate and a slight increase when increasing the gap circulation rate. Similar to the trend of Figure 4.2 (b) at higher feed temperature less percentage increase in the flux is observed at higher feed

temperature compared to with a value of 12 % when providing a gap circulation of 1 l/min at 80 °C compared to 25% at a feed temperature of 50 °C. The analyses of the energy efficiency of the system has been performed by analyzing the GOR and STEC values and the results are presented in Figure 5.6. The hot water temperature is a key determinant in the Specific Thermal Energy Consumption (STEC) and Gain Output Ratio (GOR) of the Water Gap Membrane Distillation (WGMD) system. To illustrate, for different gap circulation rates, increasing the feedwater temperature increases the system's efficiency represented by the GOR values leading to less amount of the STEC. This is similar conclusion for the theoretical results presented in Figure 4.7. With same conclusion of the theoretical analyses in Figure 4.7, the effect of gap circulation at different temperatures and flow rates, with constant conditions of 15 °C coolant temperature, but with 2 L/min cold and hot water flow rates, and 8 mm gap size; experimental results suggest that a circulated gap system performs more efficiently (higher GOR and lower STEC) compared to a non-circulated one up to a small circulation rate of 1 L/min. Beyond this, energy performance diminishes due to increased energy loss between cold and hot streams. This drop in energy efficiency is minor compared to the increased production rate. For instance, at 3 L/min circulation rate and 80 °C feed water temperature, permeate flux rises by approximately 12% compared to a non-circulated system, despite a 10 % GOR decrease and STEC increase.



(a)



(b)

Figure 5.6 Effect of feedwater temperature on the GOR (a) and STEC (b) values at a feed and coolant flow rates of 2 l/min and coolant temperature of 15 °C

5.4.2. Effect of Feed Flow Rate

In a similar trend to the theoretical results in Figure 4.4, the effect of experimentally varying the hot water flow rate on the flux for both systems was observed and shown in Figure 5.7. The variation of the feed flow rate was performed at 80 °C and 15 °C hot and cold water temperatures, and 2 L/min coolant flow rate, flux generally improves with increasing feed stream flow rate due to better turbulence and heat transfer characteristics. The flux increase for a non-circulated water gap is only 50% when increasing the feed flow rate from 1 to 3 lpm, however better increase in the flux performance observed by the circulated gap at higher hot water flow rates with 75 % increase in the flux to achieve a value 85 kg/m².hr at 3 l/min gap circulation.

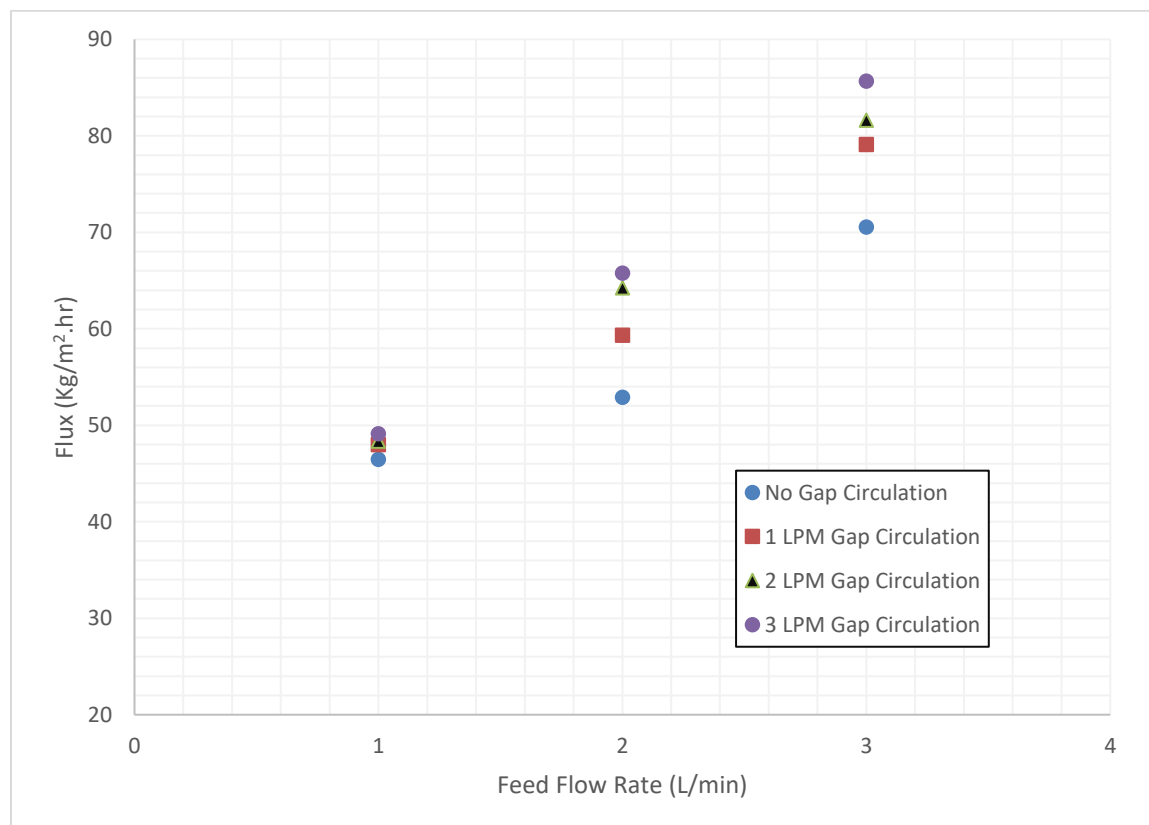
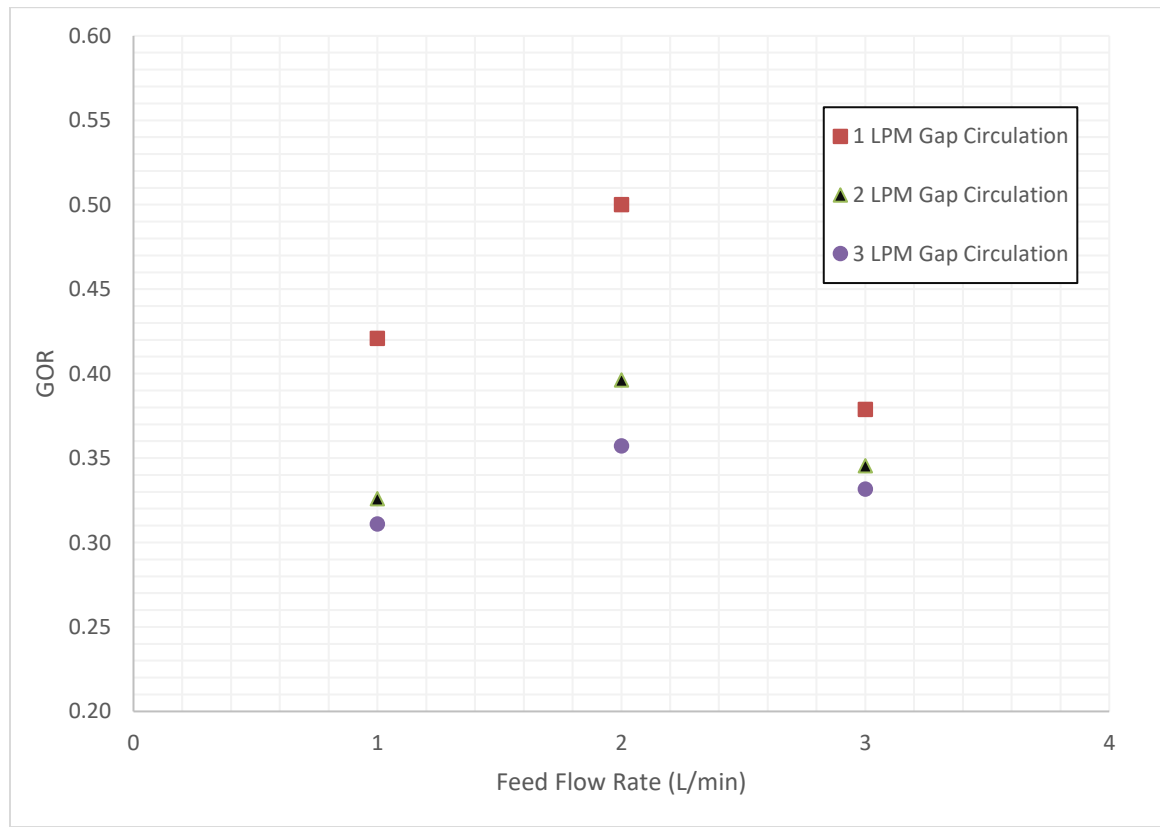
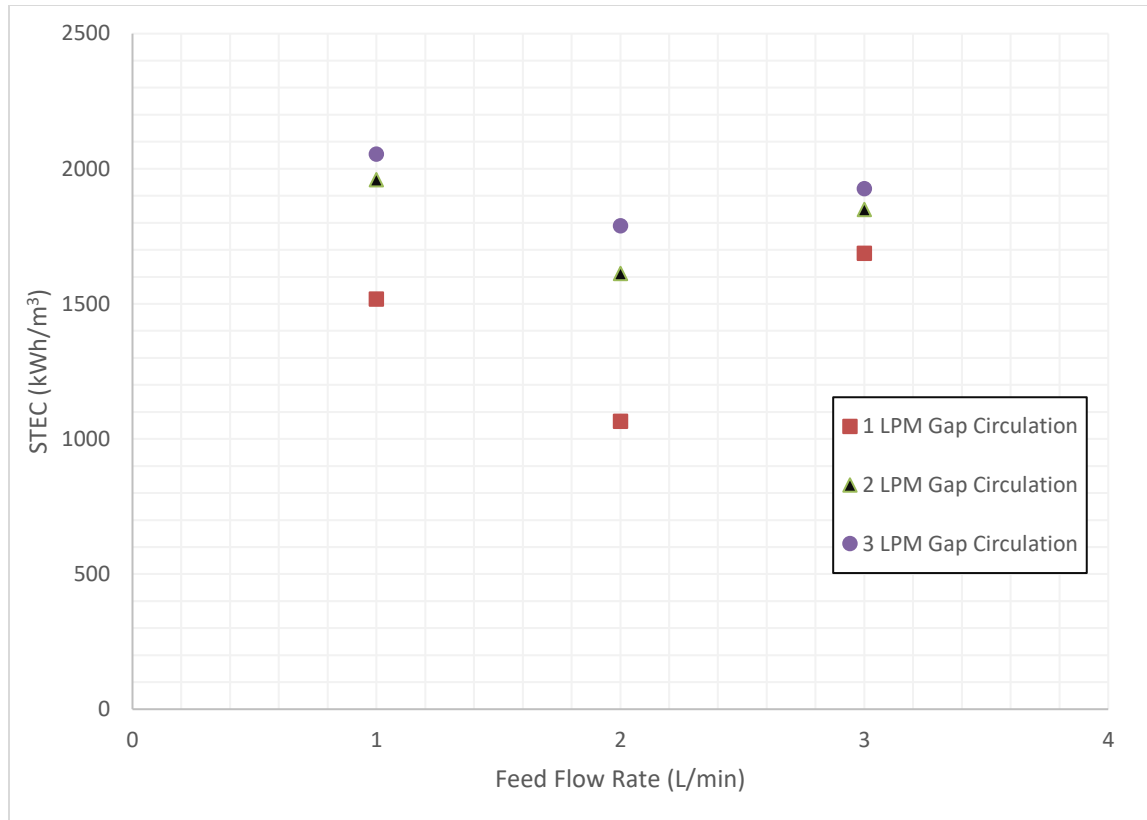


Figure 5.7 Effect of feedwater flowrate on the resultant flux (in kg/m².hr) at a feed temperature of 80 °C, coolant flow rates of 2 l/min, and coolant temperature of 15 °C

The energy efficiency analyses of the for the variation of different feed circulation rate conducted at different gap circulation rate is analyzed at fixed feed temperature of 80 °C and coolant temperature of 15 °C. The results are presented in Figure 5.8 (a) for the GOR and 68 (b) for the STEC. It is shown in Figure 5.8 (a) that increasing the feed flow rate from 1 to 2 l/min increases because of the enhanced mass transfer characteristics for different gap circulation rates. However, exceeding this value to 3 l/min lead to decrease in the system's efficiency represented by the GOR, Figure 5.8 (a), and consequently an increase in the STEC Figure 5.8 (b). The main reason for this is that at higher flow rates more conduction losses are triggered toward the cold side leading to a more heating and cooling energy consumption. It is obvious, and like Figure 5.6, increasing the gap circulation rates triggers the losses to decrease the thermal efficiency.



(a)



(b)

Figure 5.8 Effect of feedwater flowrate on the resultant GOR (a) and STEC (at a feed temperature of 80 °C, coolant flow rates of 2 l/min, and coolant temperature of 15 °C

5.5. Design, Testing, and Manufacturing for Multi-Stage Circulating WGMD

A multi-stage water gap membrane distillation unit with a circulation of gap water is a system used for the separation of fresh water from salt or brackish water through a semi-permeable membrane. The system consists of 6 stages of water gap membrane distillation modules, a 50 L hot water tank, 0.5 HP pumps for feed, coolant, and gap water, 1 inch PVC pipe headers, flow meters, thermocouples, a freshwater collecting tank, and an ambient water cooling system.

The compact circulating multi-stage water gap membrane distillation, as shown in Figure 5.9, unit is an innovative and effective solution for water treatment, offering compact design for water purification. It consists of 3 feed channels, each serving two permeate gaps and associated with a coolant channel.

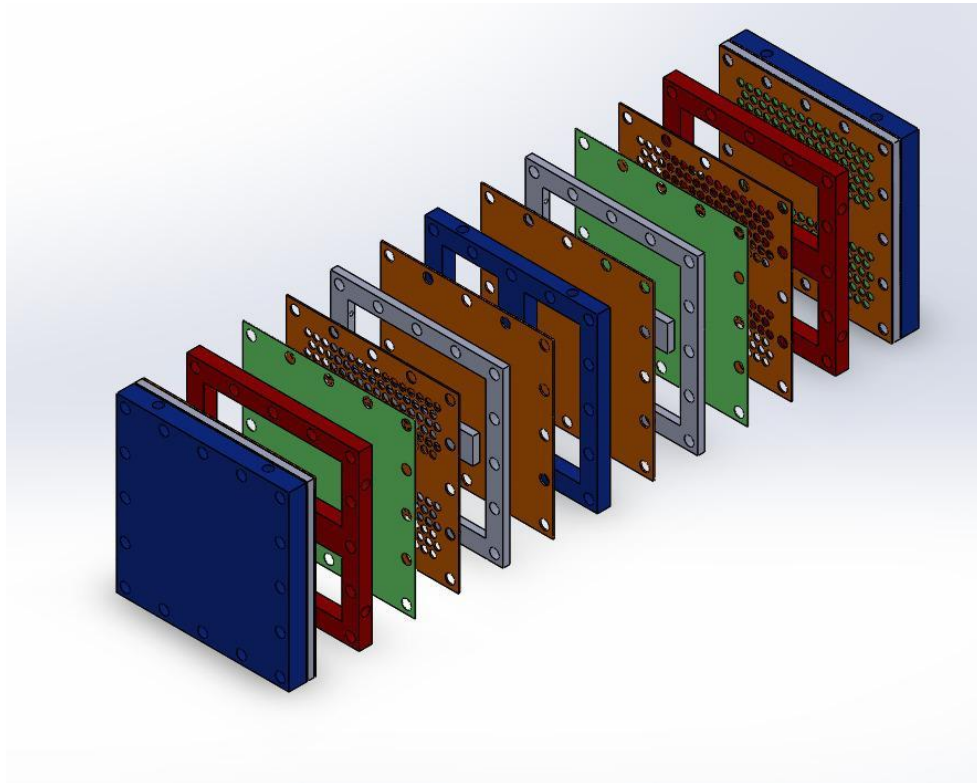


Figure 5.9 The exploded-view of the circulating water gap Multi-Stage MD desalination unit

The hot water tank pre-heats the feed water before it is pumped through the feed modules with an associated hydrophobic membrane. The ambient coolant water circulates from a nearby water source to absorb the latent energy from condensation. The ambient water-cooling helps to regulate the temperature of the system during operation.

The gap water circulates between the membranes to provide turbulence in the gap channels to enhance mass transfer of water vapor through the membrane.

By connecting the stages in parallel or series, Figure 5.10, the system can be tested under different operating conditions, such as varying the feed water temperature and flow rates. The thermocouples and flow meters monitor and measure the gap water temperature and the flow rates of the feed water, coolant water, and gap water, respectively.

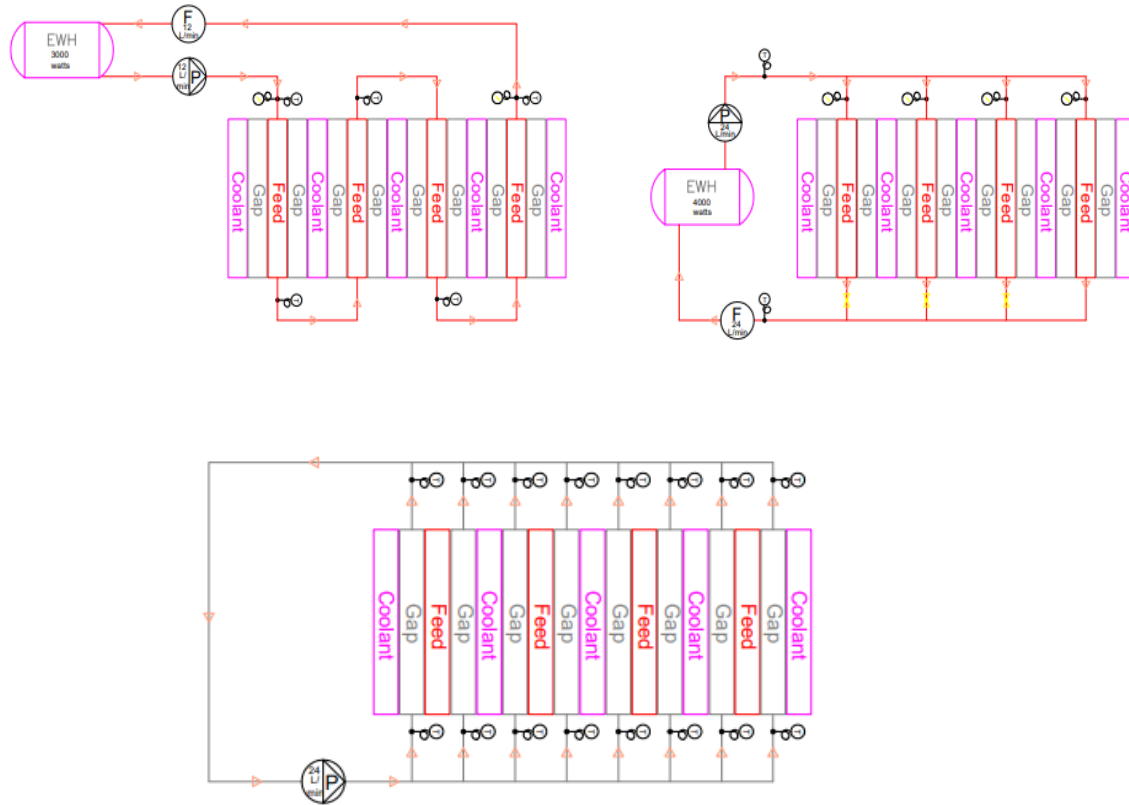


Figure 5.10 Series and parallel connection for the feed, coolant, and gap modules in the multi-stage desalination module

The fresh water produced by the distillation process is collected in the fresh water collecting tank, which is connected to the gaps, as presented in Figure 6.6.

Data such as feed water inlet and outlet temperature, permeate flow rate, permeate quality, feed water pressure, gap water temperature, gap water flow rate, and gap water circulation rate are recorded for each stage during the experiment. This data assists in calculating the

metrics performance of the multi-stage water gap membrane, such as the flux, STEC, and GOR values for different operating conditions.

The described system in this section is manufactured and tested for proper operation to be tested as future work under 24 hours continuous production for the purpose of commercialization.

CHAPTER 6

OPTIMIZATION OF THE MULTI-STAGE CIRCULATING

WGMD

This chapter utilizes an Artificial Neural network that predicts the output performance of a circulating water gap membrane distillation. This ANN will be deployed with a differential evolution algorithm to optimize the water production cost. The optimization is summarized in the flowchart in Figure 6.1. This flowchart outlines a comprehensive method for optimizing the water production cost of the proposed MD system. The methodology involved creating a mathematical model for the MD system using EES software and collecting a dataset of four inputs and two outputs. The inputs were the feed temperature, feed flow rate, gap circulation rate, and coolant flow rate, whereas the outputs were the flux and STEC. The dataset was used to train a machine learning model that established a linear input/output relationship using a Multi-Layer Perceptron Artificial Neural network (MLP-ANN), which was validated using ANOVA. A water production cost model was then developed using the flux and STEC outputs as a practical implementation for the ANN predictions. Finally, a DE optimization model in MATLAB was used to determine the optimal MD parameters based on the minimum unit production cost (US\$/m³) using the DE operators described in the optimization section. Consequently, the optimal parameters for the lowest production cost were obtained, which can be used to design and operate an efficient and cost-effective MD system.

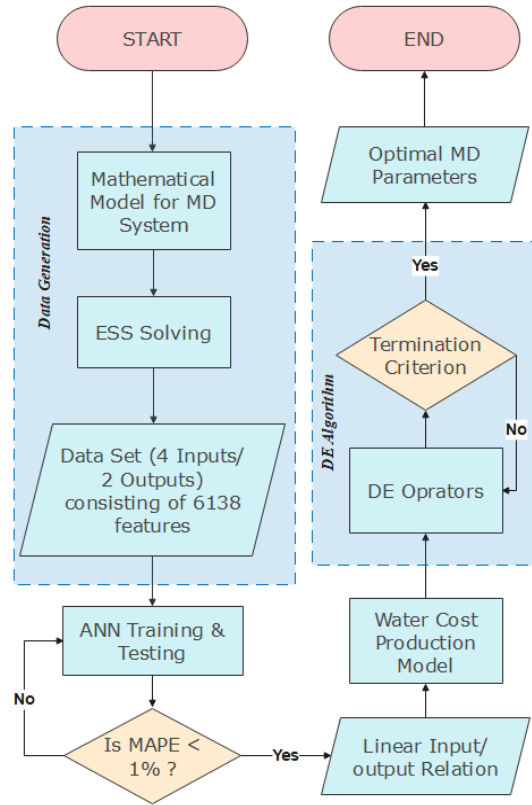


Figure 6.1 Optimization process of circulating water gap MD

6.1. ANN Modeling

The MLP-ANN was used to predict the flux and STEC in permeate-circulated water-gap membrane distillation. The MLP process utilizes several neurons in different layers to process data and predict outputs. The input layer contained four neurons for the feed flow rate, permeate (gap) circulation rate, coolant flow rate, and feed temperature, whereas the output layer included two neurons for the flux and STEC. The number of neurons in the hidden layer was optimized using a parametric study to achieve the best prediction performance. Neurons in the output and hidden layers receive weighted inputs and bias constants. The activation function used in the hidden layer is selected for the best performance between non-linear functions such as tan-sigmoid, log-sigmoid, or pure

linear, presented in equations 58, 59, and 60, and it is commonly used to capture non-linear relationships [162]. The activation function in the output layer was linear to capture the linear relationship between the inputs and outputs. The ANN architecture is illustrated in Figure 6.2.

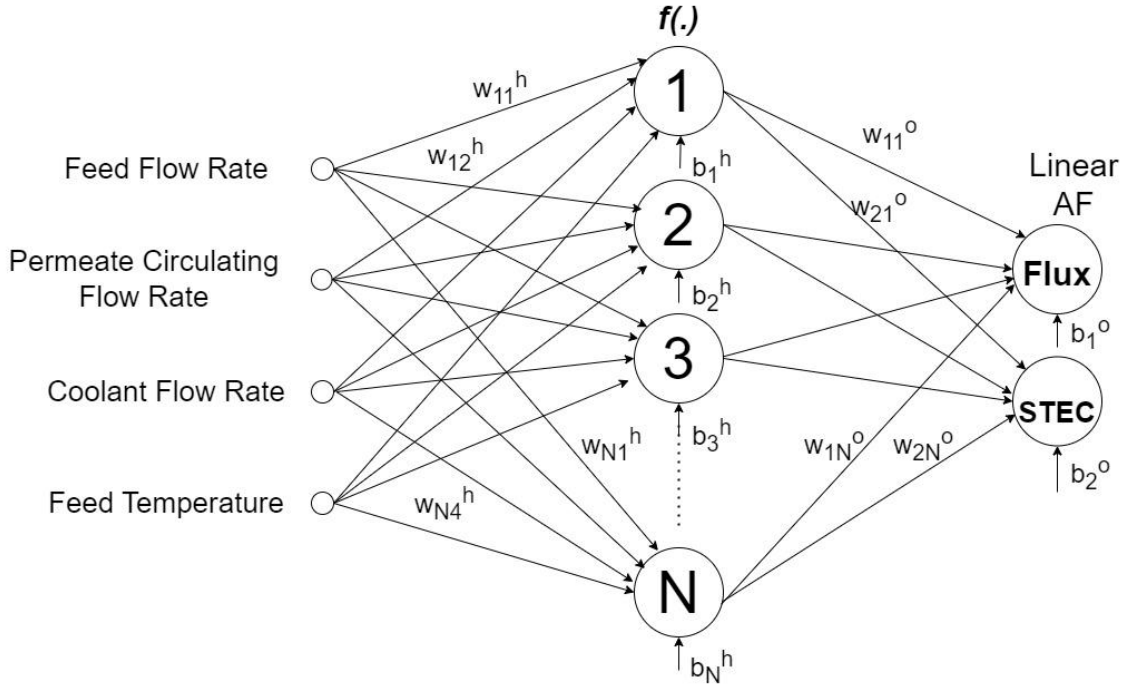


Figure 6.2 The MLP-ANN model architecture

The MLP-ANN model architecture presented in Figure 6.2 comprises an input layer with feed temperature, feed flow rate, coolant flow rate, and gap circulation flow rate as input features; a hidden layer with n number of neurons and activation functions; and an output layer with Flux and STEC as the final results. The number of neurons in the hidden layer

was determined experimentally. The model uses backpropagation to update weights and biases during training and can be used to predict flux and STEC based on new input values.

Equations 54 to 57 define the architecture of the feedforward artificial neural network (ANN) relating to the output, namely, flux and STEC with the input feed temperature, feed flow rate, coolant flow rate, and gap circulation rate.

$$STEC_n = \sum_{i=1}^N \omega_{1i}^o u_i + b_1^o \quad (54)$$

$$Flux_n = \sum_{i=1}^N \omega_{2i}^o u_i + b_2^o \quad (55)$$

$$u_i = f(v_i) \quad (56)$$

$$v_i = Q_f \omega_{i1}^h + Q_p \omega_{i2}^h + Q_c \omega_{i3}^h + T_f \omega_{i4}^h + b_i^h \quad (57)$$

Where, $F(x)$ is the activation function for the hidden layer that can be either tan-sigmoid, log-sigmoid, or pure linear [163].

$$f(\lambda) = \text{tansig}(\lambda) = \frac{1-e^{-\lambda}}{1+e^{-\lambda}} \quad (58)$$

$$f(\lambda) = \text{logsig}(\lambda) = \frac{1}{1+e^{-\lambda}} \quad (59)$$

$$f(\lambda) = \text{purlin}(\lambda) = \lambda \quad (60)$$

The weights and biases ($\omega_{1i}^o, \omega_{2i}^o, b_1^o, b_2^o, \omega_{i1}^h, \omega_{i2}^h, \omega_{i3}^h, \omega_{i4}^h$, and b_i^h) in this ANN are unknown and they can be estimated using error backpropagation algorithms. This optimization problem minimizes a cost function, typically defined as the mean squared error, using gradient descent or a similar algorithm to determine the optimum weights and biases that minimize the difference between predicted and actual outputs.

$$I = \frac{1}{2} \sum_{k=1}^2 (e_k(p))^2 = \frac{1}{2} [(e_1(p))^2 + (e_2(p))^2] \quad (61)$$

$$e_1(p) = STEC(p) - STEC_n(p) \quad (62)$$

$$e_2(p) = Flux(p) - Flux_n(p) \quad (63)$$

$STEC_n(p), Flux_n(p)$ are functions of the weights and biases that must be optimized to minimize the error between the actual and predicted values. Here, p is the pattern number, which varies from one to the maximum number of data points in the training dataset. The Iterative Search Method was used to write the iterative equations for the weights and biases [163].

$$\omega_{1i}^o(t+1) = \omega_{1i}^o(t) - \eta \cdot \frac{\partial I}{\partial \omega_{1i}^o} \quad (64)$$

$$\omega_{2i}^o(t+1) = \omega_{2i}^o(t) - \eta \cdot \frac{\partial I}{\partial \omega_{2i}^o} \quad (65)$$

$$b_1^o(t+1) = b_1^o(t) - \eta \cdot \frac{\partial I}{\partial b_1^o} \quad (66)$$

$$b_2^o(t+1) = b_2^o(t) - \eta \cdot \frac{\partial I}{\partial b_2^o} \quad (67)$$

$$\omega_{i1}^h(t+1) = \omega_{i1}^h(t) - \eta \cdot \frac{\partial I}{\partial \omega_{i1}^h} \quad (68)$$

$$\omega_{i2}^h(t+1) = \omega_{i2}^h(t) - \eta \cdot \frac{\partial I}{\partial \omega_{i2}^h} \quad (69)$$

$$\omega_{i3}^h(t+1) = \omega_{i3}^h(t) - \eta \cdot \frac{\partial I}{\partial \omega_{i3}^h} \quad (70)$$

$$\omega_{i4}^h(t+1) = \omega_{i4}^h(t) - \eta \cdot \frac{\partial I}{\partial \omega_{i4}^h} \quad (71)$$

$$b_i^h(t+1) = b_i^h(t) - \eta \cdot \frac{\partial I}{\partial b_i^h} \quad (72)$$

η is the learning rate, which is the hyperparameter and controls the speed and stability of the optimization process.

For computing the partial derivatives in above equations, the chain rule will be applied, and the weights and biases will be approximated from our training data set.

$$\frac{\partial I}{\partial \omega_{1i}^o} = -e_1(p)u_i \quad (73)$$

$$\frac{\partial I}{\partial \omega_{2i}^o} = -e_2(p)u_i \quad (74)$$

$$\frac{\partial I}{\partial b_1^o} = -e_1(p) \quad (75)$$

$$\frac{\partial I}{\partial b_2^o} = -e_2(p) \quad (76)$$

$$\frac{\partial I}{\partial \omega_{ij}^h} = -\sum_{k=1}^2 (e_k(p)\omega_{ki}^o) f'(v_i(p))X_j(p) \quad (77)$$

$$\frac{\partial I}{\partial b_i^h} = -\sum_{k=1}^2 (e_k(p)\omega_{ki}^o) f'(v_i(p)) \quad (78)$$

X_j is the j th input variable. $j=1,2,3,4$ refers to the Q_f, Q_p, Q_c & T_f respectively.

To evaluate the performance of the designed neural network, the mean absolute percentage error (MAPE) is used which is defined in equation (26) [164].

$$MAPE = \frac{\frac{|Predicted_{output} - Actual_{output}|}{Actual_{output}}}{n} \times 100 \quad (79)$$

6.2. Analysis of Variance

The mathematical relationships used to calculate the statistical estimators for ANOVA are based on the sum of squares (SS) of the differences between the predicted and the actual values. The degree of freedom represents the number of independent observations in a dataset and is used to calculate the variance and standard error of the mean. In the case of an ANN model, the degree of freedom of the residual can be calculated by subtracting the number of parameters (“a”), that is, the number of weights and biases in the model from the number of observations (“n”) in the data set. This is because the parameters in the ANN

model are not independent and are determined by the data. Therefore, they do not contribute to the degrees of freedom of the error [109].

$$df_e = n - a \quad (80)$$

$$df_m = a - 1 \quad (81)$$

$$df_t = n - 1 \quad (82)$$

The mathematical relationships used to calculate the p-value for the significance of the difference between the predicted and actual outputs based on the F distribution are obtained from [165]. If the p-value of the ANOVA test was less than the predetermined significance level (typically 0.05), the null hypothesis was rejected. This indicates that there is a significant difference between the means of the populations, and that the ANN model can accurately predict the response variables.

6.3. Differential Evolution Optimization Algorithm

Differential Evolution (DE) is a well-known population-based optimization technique that seeks to find the optimal global solution. This method is popular due to its efficiency, fast convergence rate, and ease of implementation, and has been applied in a range of engineering optimization problems [66], [166]. It operates with two key parameters, the mutation factor (F) and the crossover constant (CR). The DE algorithm consists of four fundamental steps:

1) Initialization: To begin, a set of initial vectors, with a population size (NP), are randomly generated as

$$x_{i,j} = x_j^{min} + r \times (x_j^{max} - x_j^{min}) \quad (83)$$

where $i = 1, 2, \dots, NP$; $j = 1, 2, \dots, O$; O is the number of optimized parameters r is a random number in the range of $[0,1]$, x_j^{max} and x_j^{min} are the upper and lower bounds of the optimized parameters.

2) Mutation: Each vector x_i in the population size generates a mutant vector v_i using a mutation operation [167]. The mutant vector v_i in this study was generated as:

$$v_i = x_{R1} + F \times (x_{R2} - x_{R3}) \quad (84)$$

where R1, R2, and R3 are random integer numbers within the range [1, NP].

3) Crossover: After NP mutant vectors are produced, a trial vector t_i is generated by crossing the mutant vector v_i with the individual vector x_i . The trial vector t_i is given by:

$$t_{i,j} = \begin{cases} v_{i,j} & \text{if } r \leq CR \\ x_{i,j} & \text{otherwise} \end{cases} \quad (85)$$

4) Selection: Finally, after NP trial vectors are created, NP vectors x_i^{new} are selected for the next generation using:

$$x_i^{new} = \begin{cases} t_i & \text{if } f(t_i) \leq f(x_i) \\ x_i & \text{otherwise} \end{cases} \quad (86)$$

Therefore, the optimization problem is formulated to minimize UPC (30) with subject to:

$$T_f^{min} \leq T_f \leq T_f^{max} \quad (87)$$

$$F_f^{min} \leq F_f \leq F_f^{max} \quad (88)$$

$$F_c^{min} \leq F_c \leq F_c^{max} \quad (89)$$

$$F_g^{min} \leq F_g \leq F_g^{max} \quad (90)$$

where T_f , F_f , F_c , and F_g are the feed temperature, feed flow rate, coolant rate, and gap flow rate respectively. The superscript min and max donate the lower and upper bound of the corresponding variable which are discussed in Table 6.1.

6.4. ANN Performance

To determine the most effective and optimized operating conditions, an ANN neural network was trained using a set of 6138 data points generated from a highly non-linear mathematical model, section 3.2, solved in Engineering Equation Solver (EES), a software package designed for thermal science problems. The data points were then transferred to

MATLAB for the modeling and analysis of the ANN. The data generated for the ANN include a range of values for the feed temperature, feed flow rate, gap circulation rate, and coolant flow rate, and the resulting distillate flux and Specific Thermal Energy Consumption (STEC). The range of the feed temperature is based on previous literature, while the range of the flow rates is based on the experimental design of [9]. The range of input/output is presented in Table 6.1. The data are produced at a constant feed concentration of 150 ppm and a membrane area of 0.0066 m², using a commercial Polytetrafluoroethylene (PTFE) hydrophobic membrane with a pore size of 0.45 µm, a porosity of 80%, and a thickness of 6.9 µm. The coolant temperature is fixed and assumed equal to the ambient water temperature of 22 °C.

Table 6.1 Characteristics of the generated data points

Parameter Designation	Data type	Range
Feed Temperature (°C)	Input	50-90
Feed Flow Rate (l/min)	Input	1-5
Gap Circulation Rate (l/min)	Input	1-5
Coolant Flow Rate (l/min)	Input	1-5
Distillate Flux (kg/m ² .hr)	Output	35.66 – 560
Specific Thermal Energy Consumption (kWh/m ³)	Output	592.3 - 673.4

The results in Figure 6.3 show the MAPE for different numbers of neurons in the hidden layer with three different activation functions. The tan-sigmoid activation function was shown to have the best performance for a wide range of hidden layer neurons, with a MAPE

of 1% for training and testing. The log-sigmoid showed slightly less accurate results for the same range of neurons, with a MAPE average of 1.3%. However, the pure linear activation function showed an inferior performance, with a MAPE of 6.5%.

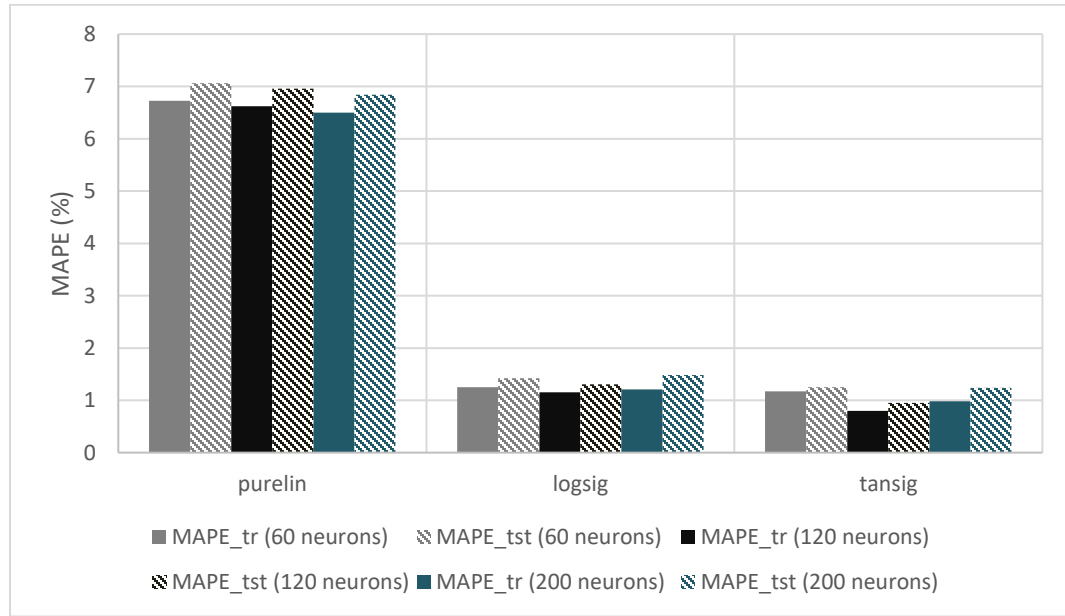


Figure 6.3 The MAPE % for different transfer functions at 60, 120, and 200 hidden layer neurons

The analysis of different numbers of hidden layer neurons used with the tan-sigmoid activation function revealed that 120 hidden layer neurons provided the lowest MAPE error percentage, with an average value of 0.75 for testing and training.

Figure 6.4 shows the performance of the ANN model based on a linear relationship between the predicted and actual outputs. The training dataset was divided into 70% for training, 15% for testing, and 15% for validation. The predicted output is in good agreement with the actual output of the training, validation, and testing data. The overall regression (correlation) coefficient for all cases is above 99%, which confirmed the accuracy of the model.

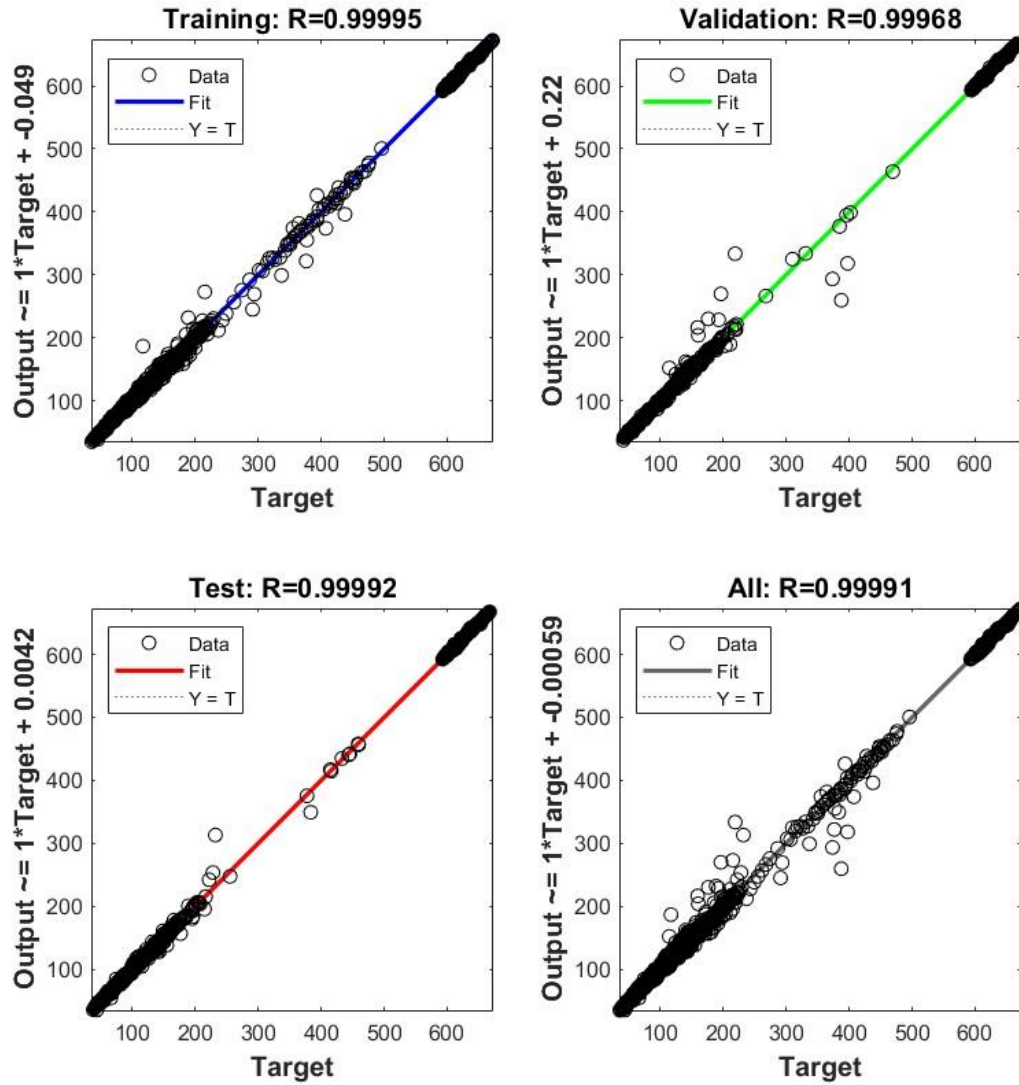


Figure 6.4 Normality analysis for training, validation, testing, and overall data

The results shown in Figure 6.5 demonstrate the effectiveness of the ANN in predicting both the flux and STEC outputs. The four parts of the figure (a, b, c, and d) compare the actual and predicted outputs and show a high degree of overlap, indicating a low error between the two sets of data. This indicates that the neural network accurately learned the relationship between the inputs and outputs. The negligible error observed in the figures supports the conclusion that the neural network effectively captured the underlying patterns

in the data. Notably, the data patterns were randomized, which presents a challenge for predictive models. However, the neural network still achieved a high level of accuracy, indicating its robustness to different types of data distributions. This is particularly evident in both the training (Figures 6.5 a & b) and testing (Figures 6.5 c & d) processes, where the actual and predicted outputs remain closely aligned.

The results of this study contribute to the previous result of 1% average MAPE, demonstrating the effectiveness of ANN in solving complex prediction problems. These results are particularly relevant for applications where high prediction accuracy is critical, such as in the current study, to determine the optimum water production cost. Overall, these results demonstrate the robustness and accuracy of ANN in predicting the flux and STEC outputs, even in the presence of randomized data patterns.

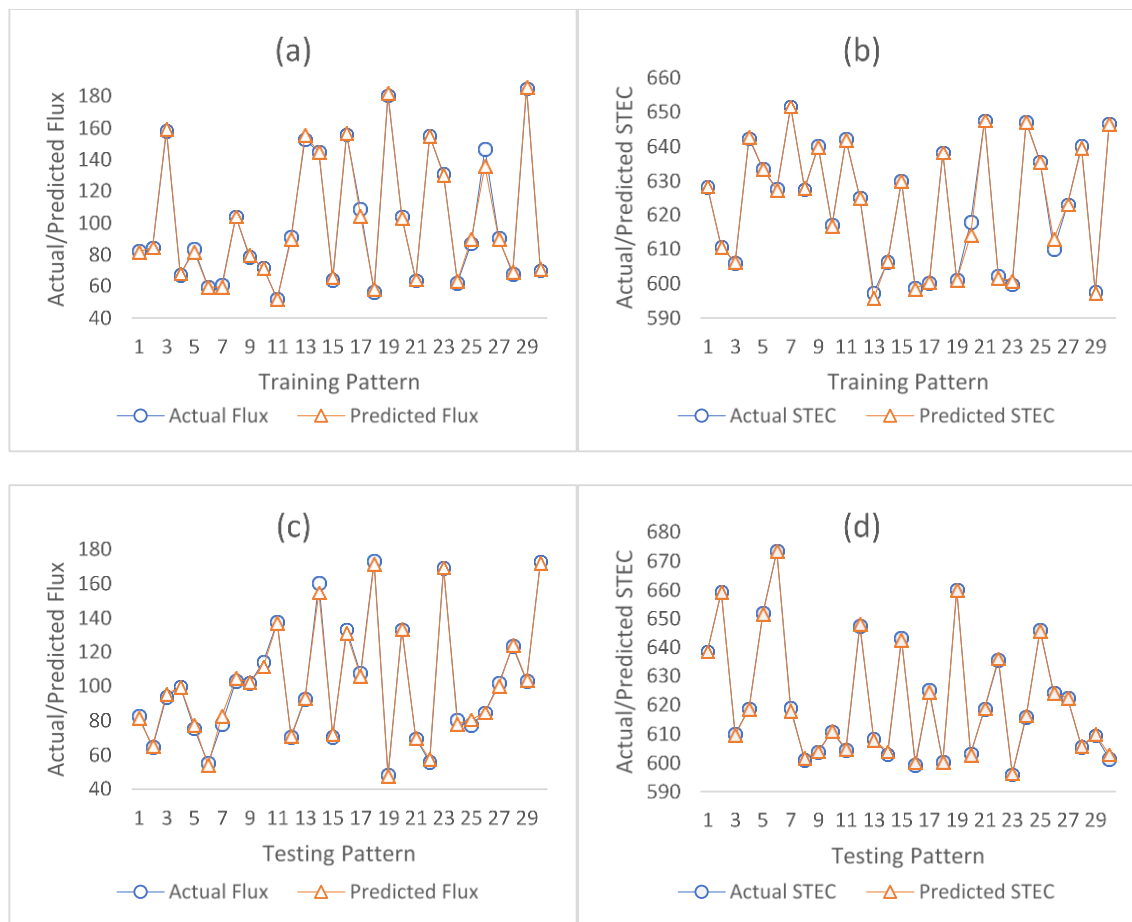


Figure 6.5 The actual vs predicted training (a & b) and testing (c & d) ANN output for random patterns

Regression analysis of variance was applied to evaluate the accuracy of an ANN model for estimating two outputs: flux and STEC. Tables 6.2 and 6.3 list the results of the ANOVA for each output, respectively.

The results for flux estimation (Table 6.2) show that the ANN model has a large sum of squares compared to the residual error and a high F-value, indicating a strong flux dependence on the four input parameters and high accuracy in explaining this relationship by the model.

Table 6.2 ANOVA table for the ANN model for the prediction of flux

Source	Degree of Freedom	Sum of Squares	Mean Sum of Squares	F-Value	P-value
Model (Flux)	604	4.0131e+05	664.42	1.14e+07	0.00
Error (Residual)	4458	0.2599	5.83e-05		
Total	5062				

Table 6.3 shows that the ANN model accurately explains the dependence of STEC on the inputs with minimal error variance, as indicated by the low mean sum of squares of residual.

Table 6.3 ANOVA table for the ANN model for the prediction of STEC

Source	Degree of Freedom	Sum of Squares	Mean Sum of Squares	F-Value	P-value
Model (STEC)	604	3.8012e+05	630.15	4.41e+07	0.00
Error (Residual)	4458	0.0637	1.43e-05		
Total	5062				

Overall, the use of regression ANOVA in this study provides evidence of the efficacy of the ANN model in estimating flux and STEC. The ANN model can handle complex and non-linear relationships, allowing it to achieve better results in this study, such as the optimization problem.

Table 6.4 compares different studies that have used ANN in conjunction with MD processes for various applications. This table provides information on the references, process descriptions, algorithms, activation functions, inputs, outputs, methods, network structures, and performance (model efficiency). This table highlights five studies, including the current study. The results in this table indicate that ANN succeeded in obtaining a correlation between inputs and outputs with a performance efficiency of more than 99%. In all the presented studies, the feed temperature and feed flow rate (feed velocity) were considered as input parameters for the investigation because they are the key parameters affecting MD performance. The difference between the presented studies lies in their MD configuration. For instance, in AGMD, the air gap thickness is studied because it affects the heat transfer coefficient in the MD process. In VMD, the vacuum pressure is another parameter that affects the MD performance. However, this is the only study that investigates PGMD-CG. Therefore, the permeate gap circulation rate is a key parameter affecting the performance of the MD system. In this study, the gap thickness was not considered because the water gap circulation provides a forced convection heat transfer, which is slightly affected by the gap width because of the variation in gap velocity that controls the Reynolds number.

Table 6.4 Comparison of different research using ANN with MD

Reference	Process Description	Inputs	Outputs	Network Structure	Performance (Efficiency of the model)
Current Work	Optimization of the Circulating Water Gap Membrane Distillation for Desalination Applications	1. Feed Temperature 2. Feed flow rate 3. Gap Circulation Rate 4. Coolant Flow Rate	Permeate Flux & STEC	4-120-2 Tansig	$R^2 = 99.9\%$
[33]	AGMD optimized and modeled by using ANN.	1. Feed flow rate 2. Cooling inlet temperature 3. Feed inlet temperature 4. Air gap thickness	Permeate flux	4-10-1 Traingd Logsig	$R^2 = 99.2\%$
[36]	Separate an HCl/water mixture from AGMD and model it by using ANN and Taguchi optimization	1. Feed temperature 2. Cooling water flow rate 3. Feed flow rate 4. Air gap width 5. Cooling water temperature	Permeate flux	5-9-1 Tansig	$R^2 = 99.8\%$
[37]	VMD process modeling and simulation using ANN	1. Feed salt concentration 2. Feed flow rate 3. Vacuum pressure 4. Feed inlet temperature	Permeate flux	4-3-1 Tansig Logsig	$R^2 = 99.4874\%$

[31]	Utilizing a DCMD-ANN to predict water flux from pretreated palm oil mill effluent	1. Feed inlet temperature 2. Feed velocity 3. Types of membranes	Water flux	3-13-1 Tansig	$R^2 = 98.59\%$
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6.5. Cost Analysis and Optimization

The ANN described in the previous section aims to predict the output flux and the specific thermal energy consumption for a gap circulating PGMD. This prediction could be implemented to determine the Unit Production Cost (UPC) of water in US\$/m³ of a multistage system as presented in Figure 6.6. The system under investigation is comprised of two distinct subsystems, which are separated by a heat exchanger. The first subsystem, located on the left-hand side of the heat exchanger as illustrated in Figure 6.6, is dedicated solely to supplying heat to the desalination system either a waste heat or electrical power source. For the waste heat case, it is considered a steam heat exchanger from a steam waste with temperature higher than 100 °C such as that utilized in pharmaceutical applications [168], oil and gas refineries [169], industrial processes [170], and also steam power plants [171].

The desalination system, on the right side of the heat exchanger, is composed of a multi-stage circulating permeate water gap MD unit, which encompasses three separate loops. The first loop serves as the feed water heater and provides hot salt water to the feed channels. The permeate (product) loop comprises a circulating pump connected to a

freshwater product tank, which serves to improve the heat transfer coefficient inside the gap and facilitate the transport of fresh water to the collection tank. The final loop, dedicated to cooling, is characterized by a coolant pump that draws water from the sea and injects it into the coolant channels for freshwater condensation. All loops are connected to the multi-stage systems in parallel connections.

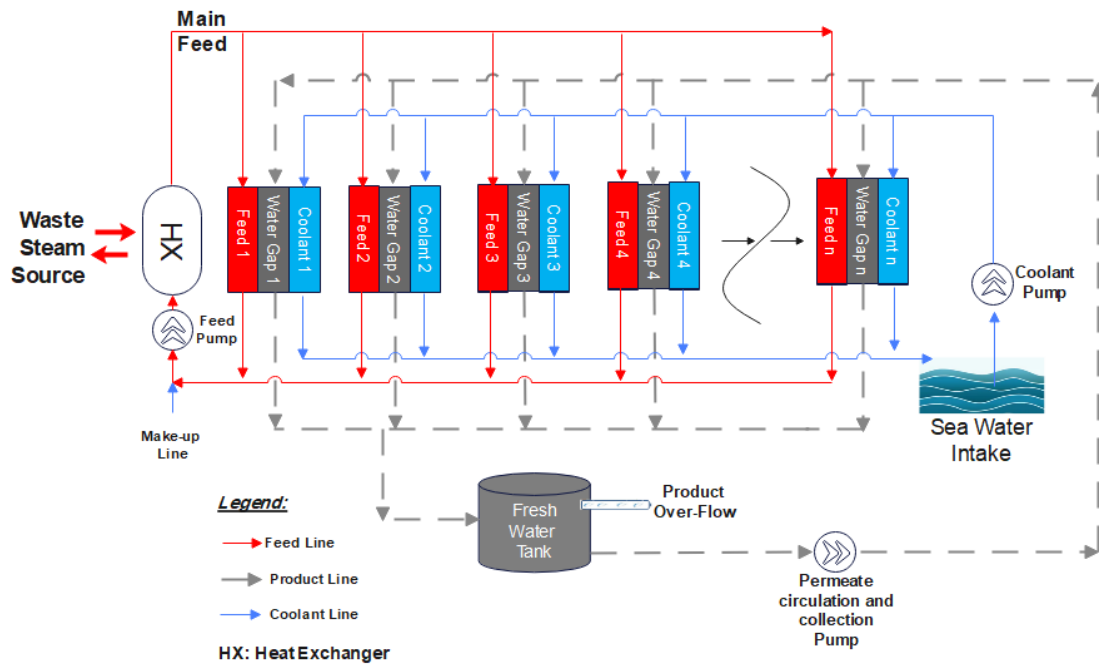


Figure 6.6 System configuration for the parallel multi-stage circulating permeate water gap MD powered by steam waste heat

The direct capital cost summary is presented in Table 6.5, and the indirect capital cost required for the initial installation is considered to be 10 % of the direct capital cost. Hence the total capital cost is the sum of direct and indirect costs.

Table 6.5 Capital cost summary

Item	Estimated Cost (USD)
Distillate Tank	100
Pumping and Connections	35 (0 – 15 l/min) 50 (16 – 25 l/min) 70 (26 – 50 l/min) 90 (51 – 90 l/min) 220 (90 – 120 l/min)
Heat Exchanger (\$/m²)	2000
Installation of single MD-Module	100
Single Membrane Cost	10

The specific cost of civil work and the cost of intake and pretreatment are neglected [157].

The study by Al-Obaidani et al. [172] serves as the basis for creating a cost model and economic assumptions for freshwater production using the proposed system. The cost model considers capital expenditure (CAPEX) and operational expenditure (OPEX). The following operational assumptions were considered: a lifetime of 20 years the MD system, with an average availability of 90% ($f=0.9$), an interest rate of 5% ($i=0.05$), specific spare parts cost of US\$0.033/m³, specific labor cost of US\$0.03/m³, specific chemical cost of US\$0.018/m³, and specific brine disposal cost of US\$0.0015/m³. The membrane was assumed to be replaced once per year and had the same replacement cost as its initial capital cost. The specific cost of electricity (COE) required to drive the pumps or provide heat to

the feed stream is US\$0.045/kWh [173], whereas heat is assumed to be provided by a diesel engine heat source.

The pumping power for the feed, gap, and coolant pumps was calculated as:

$$Pumping\ Power = \frac{Flow\ Rate * Pressure\ Drop}{\varepsilon_{Pump}} \quad (91)$$

The Flow rate is the total flow supplying all stages of each of the MD sections with a pressure drop of 1 bar (100 kPa), based on our experimental data, for the feed, gap, and coolant side. The pump efficiency ε_{Pump} is 70% (0.7).

The energy cost is:

$$\dot{C}_E = COE * (SEEC + STEC) \quad (92)$$

In the case of waste heat, STEC is neglected in energy costs.

Whereas the SEEC is the specific electrical energy consumption and is calculated as:

$$SEEC = \frac{\Sigma Pumping\ Power}{Flux * A_{Membrane}} \quad (93)$$

The heat exchanger area is calculated as:

$$A_{HX} = \frac{\dot{E}_{in}}{U * T_{av}} \quad (94)$$

U is considered the area of the heat exchanger in (W/m²), T_{av} is the average temperature difference in the heat exchanger and it is assumed to be 7°C. $\dot{E}_{in}$ is the input thermal power and it is calculated as:

The indirect capital cost required for the initial installation is considered to be 10 % of the direct capital cost. Hence the total capital cost is the sum of direct and indirect costs.

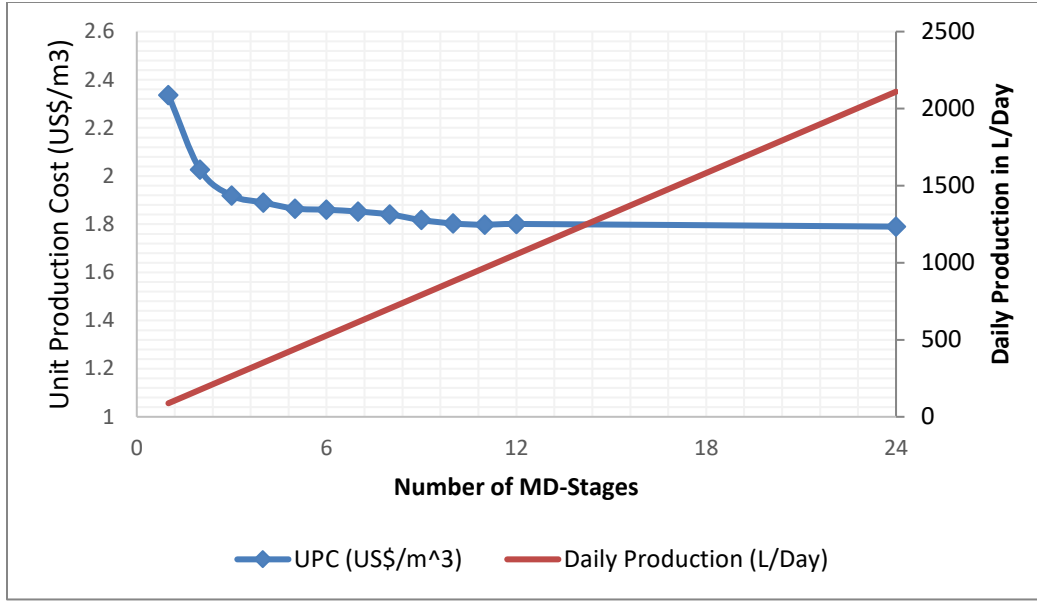
A Differential Evolution optimization MATLAB code was developed to investigate the optimal operating parameters for each MD stage. Two scenarios were studied: the first scenario considered a multi-stage WGMD-CG operating with a waste-heat source, and the second scenario considered the heat provided by an electrical source. The analyses, for both scenarios, were based on a feed concentration of 150 ppm, a commercial polytetrafluoroethylene (PTFE) hydrophobic membrane area of 0.0066 m^2 , with a pore size of $0.45 \text{ }\mu\text{m}$, a porosity of 80%, and a thickness of $6.9 \text{ }\mu\text{m}$. The seawater intake and cooling temperatures were assumed to be $22 \text{ }^\circ\text{C}$. The optimization results for both cases showed that for different numbers of MD stages, the optimal feed temperature was always $90 \text{ }^\circ\text{C}$. This temperature was the highest among the constraints for feedwater temperature. This is consistent with the literature ([174] [72] [175] [176] [177]), which shows higher water production and lower water production costs at higher feed temperatures when other parameters remain constant. Moreover, as shown in Figure 6.7, the cost of producing water decreases as the number of stages increases, because of the increase in annual water production capacities.

Figure 6.7 (a) shows that increasing the number of stages can effectively decrease the optimum water production cost for waste-heat operation. Specifically, the data show that the cost decreases by 25% from $2.35 \text{ US}\$/\text{m}^3$ for one stage to $1.8 \text{ US}\$/\text{m}^3$ for 12 stages, without any significant decrease beyond 12 stages. In contrast, the results in Figure 6.7 (b) show that the effect of increasing the number of stages on the cost of water production from electrical sources is marginal, with only a 1.7% decrease in the cost from $29 \text{ US}\$/\text{m}^3$

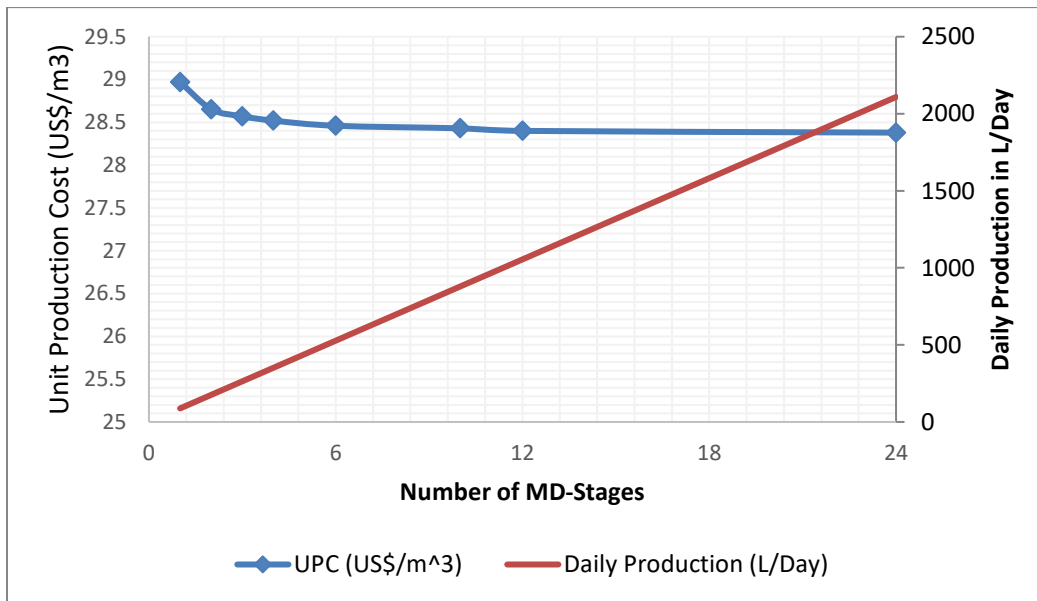
for one stage to 28.5 US\$/m³ for 12 stages. However, it is important to note that the daily productivity of this system for both heating scenarios increased linearly from less than 100 L/day for one stage to over 1000 L/day for 12 stages and more than 2000 L/day for 24 stages. This linear increase was due to the parallel connection, which created the same operating conditions across all stages.

Moreover, the data in Figure 6.8 indicate that the energy cost dominates the total production cost for providing heat from an electrical source. This is because of the high specific thermal energy consumption (STEC) of more than 590 kWh/m³ required to evaporate water, which is worth 26.5 US\$/m³. This means that although increasing the number of stages decreases the specific annualized capital cost for both operations, the impact on the overall cost is much greater for waste heat operations than for electrical sources.

In other words, increasing the number of stages can significantly increase the daily productivity and decrease the optimum water production cost for waste heat operation, whereas the effect on the cost of water production with electrical sources is much smaller. Therefore, it is important to consider the STEC for electrical operations and the initial capital cost when evaluating the cost-effectiveness of each method.



(a)



(b)

Figure 6.7 Optimum water production cost and daily productivity for different numbers of MD stages for waste heat operation (a) and electrical operation (b).

The optimization parameters are summarized in Table 6.6. For both waste heat and electrical heat provision, the number of stages was selected to be at least 12 for the least cost because a higher number of stages would not provide any decrease in the production

cost. The results also showed that the optimum flow rates for the feed, coolant, and gap circulation were in the upper bound of the optimization constraints, with values above 4.5 l/min. This implies that for higher flow rates, an enhanced heat transfer characteristic significantly increased the flux and, consequently, the water production, whereas the STEC was only marginally affected, as shown in the constraint limits in Table 6.1. These findings imply that the water production cost in membrane distillation is highly reliable for the water flux, which has a wide range over the operating conditions. In contrast, the required STEC is approximately constant without large variations over the range of operating conditions and indicates the energy required to vaporize feed water.

Table 6.6 Optimal operating conditions

Operation Scheme	Optimum Unit Production Cost (US\$/m³)	Optimum Technical Outputs	Optimum Operating Conditions
Waste Heat Operation	1.8	Flux = 555 L/m ² .hr STEC = 592 kWh/m ³	• 12 Stages • 90 °C Feed Temperature • 4.8 l/min Feed Flow Rate Per Stage • 4.8 l/min gap circulation Rate Per Stage • 4.64 l/min coolant Flow Rate Per Stage
Electrical Operation	28.4	Flux = 544 L/m ² .hr STEC = 591 kWh/m ³	• 12 Stages • 90 °C Feed Temperature • 4.6 l/min Feed Flow Rate Per Stage • 5 l/min gap circulation Rate Per Stage • 5 l/min coolant Flow Rate Per Stage

Figure 6.8 shows the breakdown structure of the cost analysis for the different heat schemes of the circulating water-gap MD process. For electrical heat provision, 93% of the water

production cost was due to energy consumption, whereas only 0.56% is for energy costs when implementing a waste heat source. This is because an intensive amount of heat is required to vaporize the water during the MD process. In the quantitative analysis, the specific annualized capital cost and other operational costs were approximately equal for both heating scenarios. However, the percentage analysis reveals that the capital cost is more significant than the total production cost of water for circulating water-gap membrane distillation when using waste heat rather than electrical heat. The energy decomposition reveals that energy cost is more significant in the case of using an electrical heat source rather than that in the waste heat source. A small amount of energy for the waste-heat scenario is only used to drive the circulating pumps of the feed, gap, and coolant streams without the need for paid energy for heat.

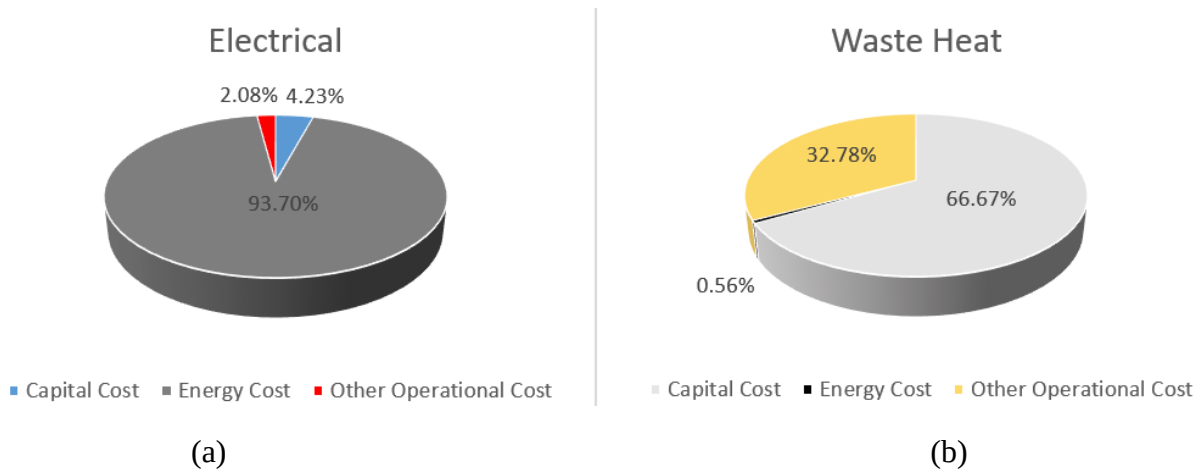


Figure 6.8 Optimum cost breakdown for the electrical and waste heat provision

This implies that heat recovery for the optimized design is required to commercialize such systems to reduce the energy and, consequently, the water production cost in the case of electrical operation. A capital cost-optimized design is required to reduce the production of a waste-heat-powered circulated water-gap MD system. These optimizations can be

implemented using a similar system of ANN integrated with a differential evolution optimization technique. The ANN was found to be a robust method to correlate input/output relations in the MD system, as demonstrated in this study and the comparison in Table 6.4. DE optimization was employed to determine the optimal operating conditions for each MD stage number, considering two different scenarios: waste heat and electric heat sources. DE optimization effectively explores the search space, converging to the global minimum cost and making it suitable for the optimization problem.

CHAPTER 7

ASSESSMENT OF HEAT RECOVERY CONFIGURATIONS

In the previous chapters, a comprehensive analysis of the circulating water gap membrane distillation (CWGMD) process was presented, highlighting its advantages over other MD configurations. One of the significant findings from this analysis was the increase in flux achieved by the CWGMD system. However, like other MD configurations, it still has a high thermal energy consumption as a drawback when it is implemented for water desalination and purification applications, which poses a challenge for its large-scale implementation and economic feasibility.

To address this issue, this chapter aims to explore the effects of internal heat recovery enhancement using the circulating water permeate gap process to enhance the energy efficiency, represented by STEC & GOR, and the overall performance. The heat recovery aims to reduce the energy consumption of the process by capturing and reusing the rejected waste heat generated during the condensation process.

By analyzing different heat recovery configurations, this chapter seeks to identify the most suitable design that can minimize energy consumption while maintaining or improving the flux and the energy metrics of the CWGMD system. The concept is influenced by the previous systems published in the literature; however, the analyses will emphasize the effect of gap water circulation on the metrics performance.

The shared concept behind different heat recovery designs is the utilization of waste heat from the distillation process to preheat the incoming feed, thereby improving the overall energy efficiency and performance of the system. Vacuum, air gap, and water gap membrane distillations, although effective, have certain drawbacks that may necessitate the use of alternative membrane distillation configurations for better heat recovery. These drawbacks include inefficient heat and mass transfer due to the presence of air or water gap. In Vacuum configuration, an increased complexity in the system design as it requires additional components such as vacuum pumps and more precise control mechanisms to maintain vacuum in the system. Moreover, proper condensation of the vapor in VMD systems can be challenging, as the vapor must travel through the vacuum gap to reach the condensation surface, which might result in reduced condensation efficiency.

7.1. Counter-Flow multi-Stage PGMD-CG

To address the previous limitations of other MD configurations, the circulated water gap membrane distillation is proposed as an alternative configuration to address these limitations, offering enhanced heat transfer and recovery by replacing the air or vacuum gap with a circulated water gap.

7.1.1. System Description

The counterflow multi-stage permeate gap membrane distillation (CFPGMD), described in Figure 7.1, is a novel approach to MD water desalination inspired from MED and MSF commercial systems, which utilizes a series of stages to optimize energy efficiency and improve product recovery. The CFPGMD process combines the advantages of both counterflow and multi-stage configurations to achieve high energy efficiency and

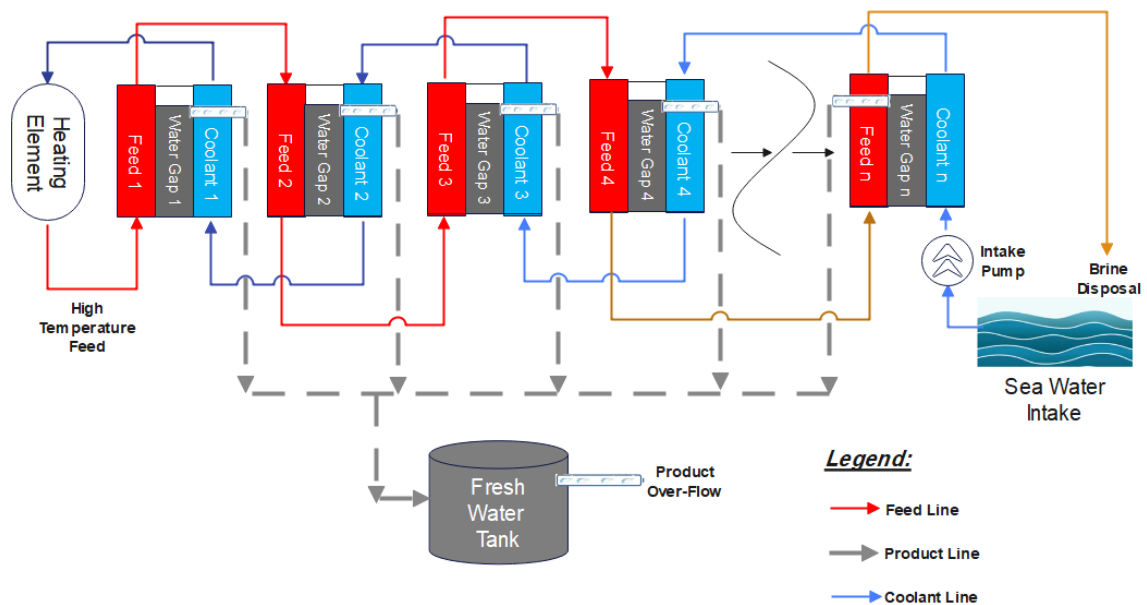
improved product recovery, making it a promising technology for water desalination and purification.

In this process, intake the water is used as a coolant for product condensation, and its temperature increases gradually after each stage, so that only a small rise in temperature at the last stage, meaning less input energy, before it is introduced back into the feed channel. The process is presented in Figure 7.1 (a) without circulation and (b) for the system with permeate gap circulation and it occurs as follows:

1. Intake water introduction: The intake water, typically containing salt, is introduced into the coolant channel of the last distillation stage. This water serves as a coolant for the product condensation process.
2. Stage-wise temperature increment: As the intake water flows through each stage, its temperature gradually increases due to the heat exchanged with the water vapor during the membrane distillation process. This increase in temperature is due to the energy absorption, recovery, from the condensation process of freshwater vapor to liquid.
3. Membrane distillation: In each stage, a hydrophobic microporous membrane separates the feed water from the permeate (or water vapor) gap. The temperature difference between the feed water and the permeate gap drives water vapor transport across the membrane. The salt remains in the feed water, while the water vapor is condensed on the cooler side of the membrane.
4. Product condensation and coolant recovery: The water vapor that has passed through the membrane is condensed in the permeate gap, the analyses are performed for a circulating water gap and compared with a stagnant water gap,

forming freshwater product. The feed water is simultaneously cooled down in the process, as it loses latent energy due to vaporization and permeation. On the other side, in the coolant channel, this latent heat is absorbed.

5. Series steps with decreasing temperature: After each stage, the injected water flows to the next stage, the feed water temperature decreases because of losing heat as a form of vapor transfer.
6. Brine disposal: Once the feed water has passed through all the stages, it becomes concentrated with salt or other impurities and is disposed of as brine.
7. Freshwater product collection: The condensed freshwater product in the permeate gap is collected and directed to a freshwater product tank, where it can be stored and used for various applications, such as irrigation, industrial processes, or treatment process for drinking water supply.



(b)

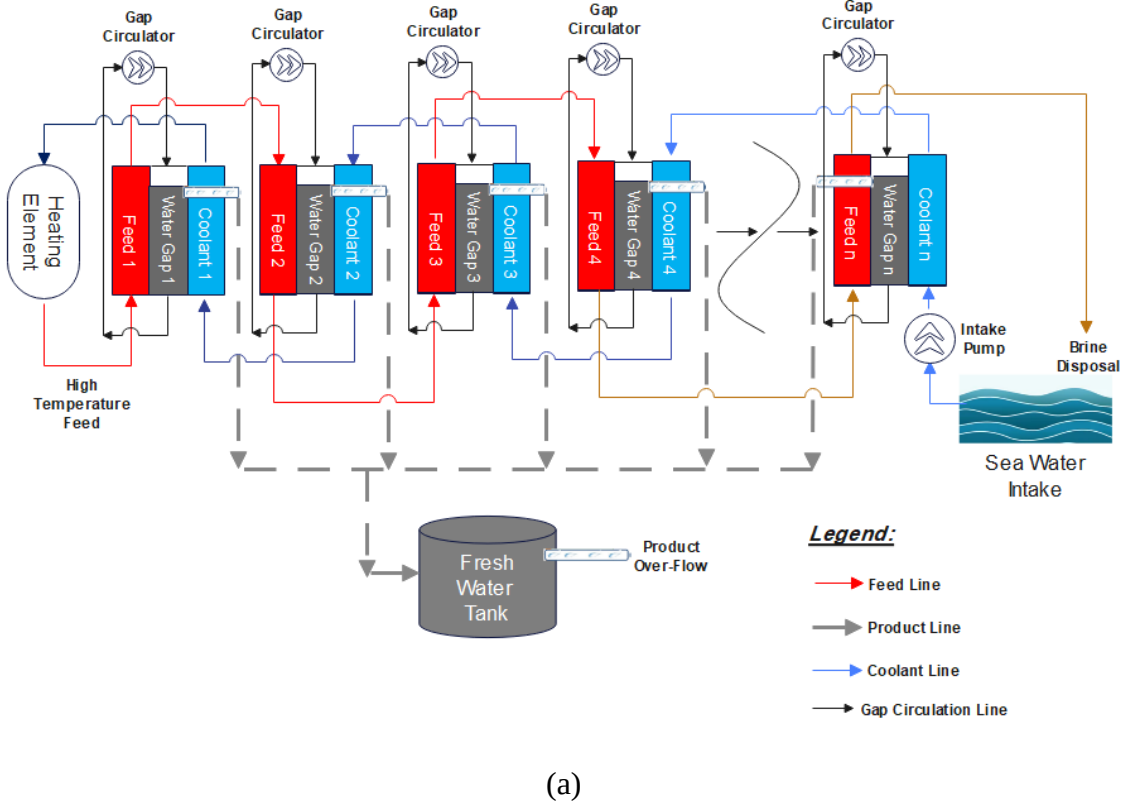


Figure 7.1 Counter-flow/multi-stage PGMD configuration for the purpose of heat recovery with stagnant gap water (a) and circulated gap water (b)

7.1.2. Mathematical Model

The heat and mass transfer modes are like those described in section 3.2, for the single stage PGMD, where the heat crosses from the feed channel to the coolant channel by convection, conduction, and vapor transfer across the membrane. The natural convection presents in the stagnant water gap while forced convection due to the circulation with a circulation pump.

For the counter-flow multi-stage system, the temperature of the cold and hot streams at the entrance of each MD unit can be estimated as follows:

$$T_{f.(i)} = T_{f.(i-1)} - \Delta T_{f.(i-1)} \quad i = 2 \text{ to } n \quad (95)$$

$$T_{c,(i-1)} = T_{c,(i)} + \Delta T_{c,(i-1)} \quad i = 2 \text{ to } n \quad (96)$$

where $T_{c,(i)}$ and $T_{f,(i)}$ are the inlet temperatures of the cold and hot stream of stage (i) , respectively. $T_{c,(i-1)}$ and $T_{f,(i-1)}$ are the inlet temperatures of the cold and hot stream of the preceding stage $(i - 1)$, $\Delta T_{c,(i-1)}$ and $\Delta T_{f,(i-1)}$ are the temperature variation between the stages in the cold and hot sides, respectively. (n) is the number of stages.

In this case scenario, the intake water is drawn at Q_{Intake} in liters per minute (lpm), the intake water flow passes across all coolant channels at the same rate with increasing temperature until it passes across the heating element when energy is added as $\dot{E}_{in}$ to be rise to the top temperature before it is injected at as a feed at the first stage. $\dot{E}_{in}$ is defined as:

$$\dot{E}_{in} = \left(\frac{Q_{Intake}}{60}\right) * C_p * (T_{f,(1)} - [T_{c,(1)} + \Delta T_{c,(i)}]) \quad (97)$$

For the first stage:

$$Q_{Feed (1)} = Q_{Intake} \quad (98)$$

Where, $Q_{Feed (1)}$ is the feed water flow rate for the first stage.

The feed water flow rate at each stage is calculated as:

$$Q_{Feed (i)} = Q_{Feed (1)} - \sum_1^i Q_{FreshWater,(i)} \quad i = 2 \text{ to } n \quad (99)$$

Where, $Q_{FreshWater,(i)}$ is the distillate flow rate at each stage “i”.

The feed recovery ratio, known as RR, for the system is defined as:

$$RR = \frac{\sum_1^n Q_{FreshWater,(i)}}{Q_{Intake}} * 100 \quad (100)$$

The heat recovery ratio for this system is defined as the heat recovered in the counterflow multi-stage system to the amount of heat that shall be theoretically added to raise the intake

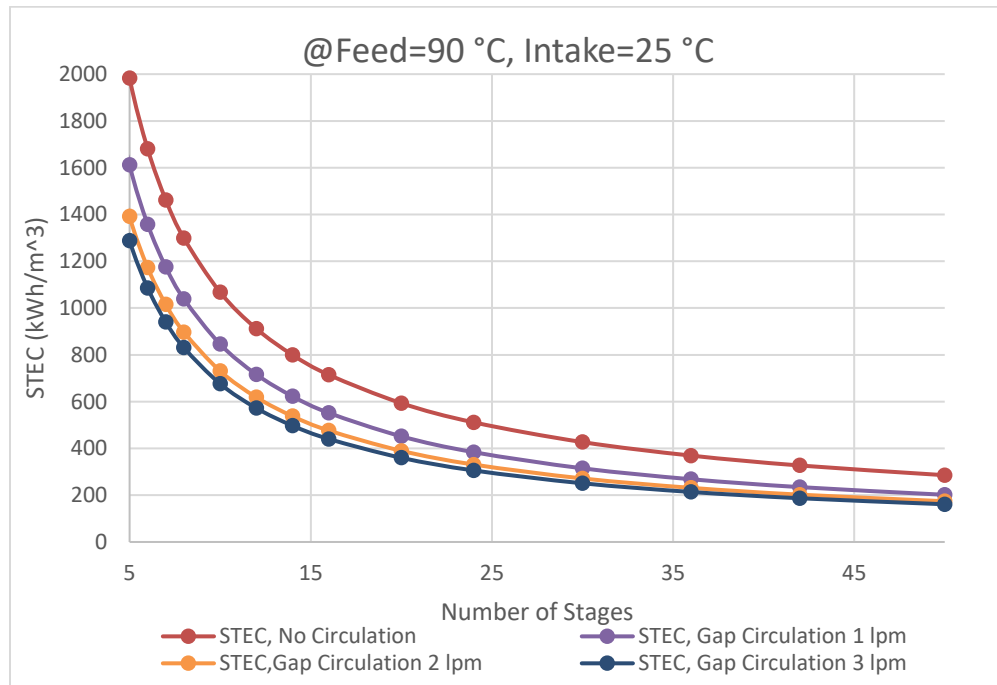
water temperature from the intake temperature to the feed injection temperature $T_{f.(1)}$ and it is defined as:

$$HRR = \frac{(T_{f.(1)} - T_{Intake}) - (T_{f.(1)} - T_{C.(1)})}{(T_{f.(1)} - T_{Intake})} \quad (101)$$

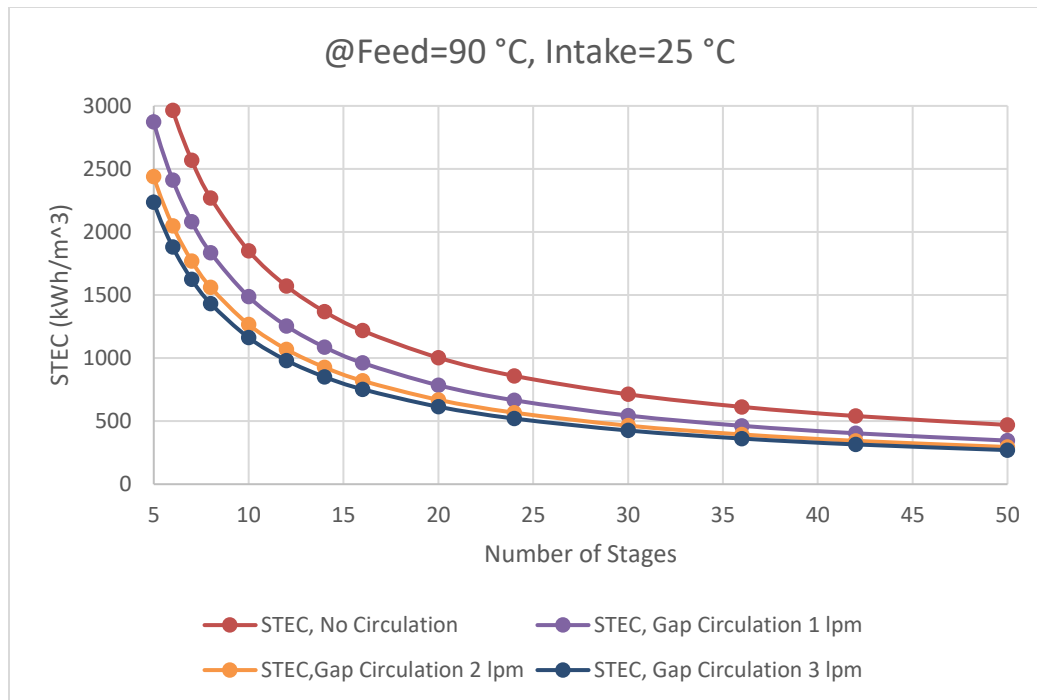
7.1.3. Results and Discussions

The analyses aim to investigate the effect of varying the number of stages on the system's metrics performance for both circulating and stagnant permeate water gap with different design and operating conditions. The first analysis has been performed to assess the effect of intake flow rate on the STEC, GOR, ratio of heat recovered, average flux, and the feed recovery ratio in Figures 7.2 to 7.6, respectively. Each Figure contains four different sections, a, b, c, and d for 1, 2, 3, and 4 l/min feed intake rate, respectively. For consistency, the similar feed, coolant, and gap dimensions with similar membrane characteristics, and similar coolant and condensation plate are used as in the analyses in chapters 4 & 6. Figures 80 to 84 are to study the effect of gap circulation rate on the system performance and they are performed at constant feed injection temperature of 90 °C, coolant intake temperature of 25 °C and 8 mm gap thickness. The analyses of Figure 7.2 reveal that increasing the number of stages in the counter flow configuration decreases the STEC required to produce 1 m³ of freshwater product. The aim of the heat recovery is to obtain a STEC the minimum optimized value of 600 kWh/m³ obtained in chapter 6 with a corresponding GOR value higher than 1. To break these values, it is required for an intake flow rate of 1 l/min 20 heat recovery stages without gap circulation with approximately half number of stages, 11 stages, to breakeven and recover enough amount of heat when using a gap circulation rate of 3 l/min. The Figure also claims that increasing the number of stages decreases the STEC

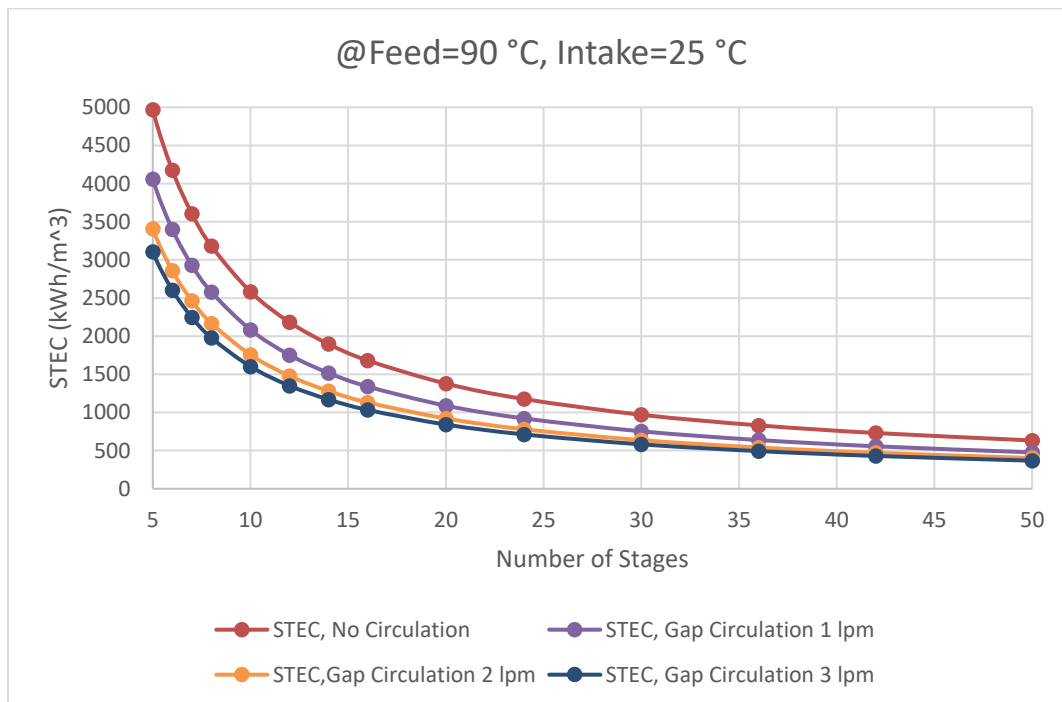
for both systems with and without permeate water gap. However, using a circulating water gap showed better STEC for a constant stage number. For example, for 50 stages, providing a gap circulation of 3 lpm leads to 43 %, 42%, 41%, and 43% decrease in the STEC respectively for 1, 2, 3, and 4 l/min intake rate, respectively, compared to stagnant water gap. The Figure also demonstrates that for lower intake flow rates, the system presents lower STEC with a minimum value of 161 kWh/m³ for 1 l/min intake increasing to 270, 370, 420 kWh/m³ when increasing the intake rate to 2, 3, and 4 l/min respectively. Those best performance parameters were observed at maximum number of stages (50) and maximum gap circulation rate (3 l/min). It is obvious that increasing the feed intake rate decreases the energy efficiency. This can be attributed to the increased energy required to raise the feed temperature at higher flow rates (mass rates) without a significant increase in flux when exceeding the turbulent feed limit, as would be explained in the detailed stage analyses.



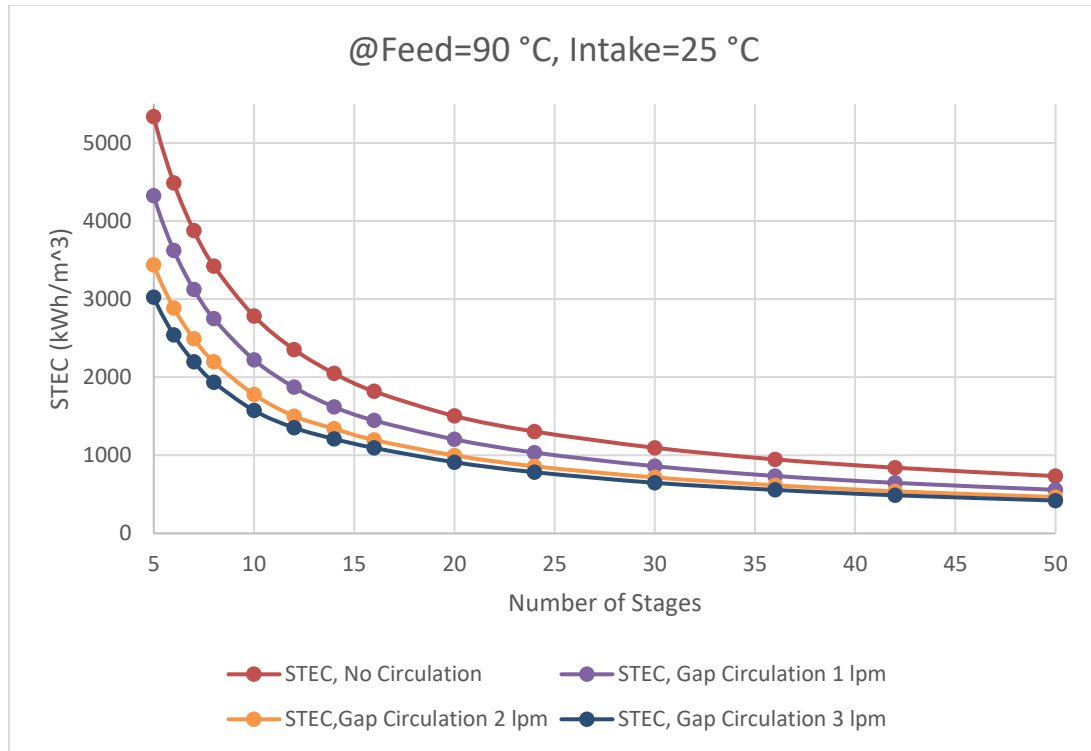
(a)



(b)



(c)



(d)

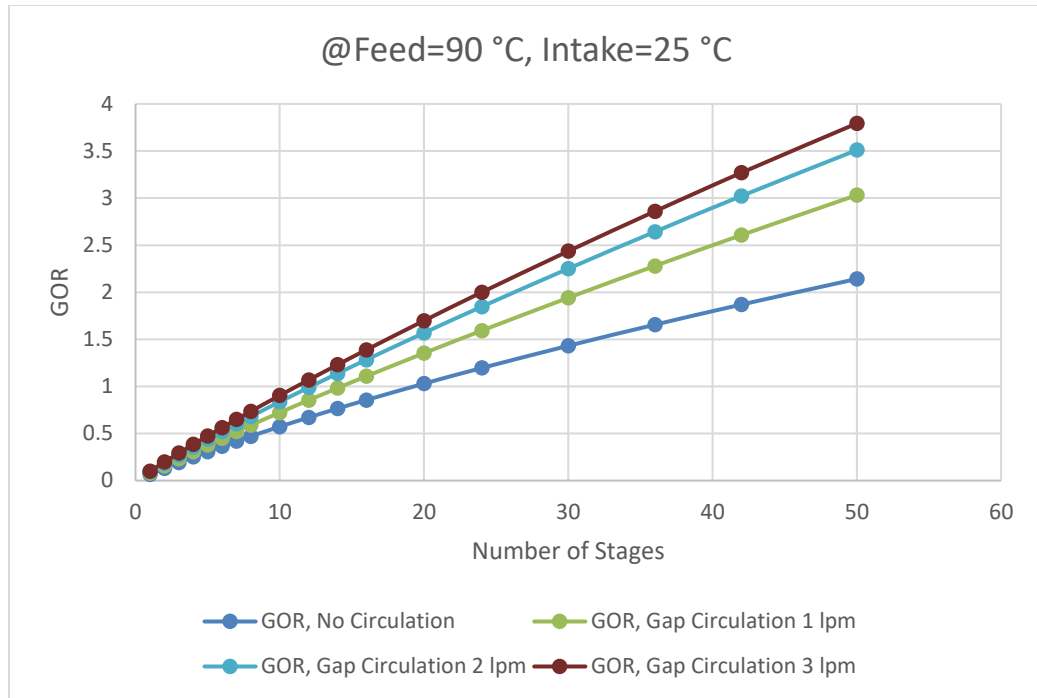
Figure 7.2 The effect of different numbers of stages on the STEC at different gap circulation rate for intake rate of 1, 2, 3, and 4 lpm in (a), (b), (c), and (d) respectively

The observation of an increasing Gained Output Ratio (GOR), in Figure 7.3, associated with a simultaneous decrease in Specific Thermal Energy Consumption (STEC) in the CFPMD system can be explained by the system's effective heat recovery and transfer between stages. As the number of stages increases in the CFPMD system, the heat recovered from the condensation process in each stage is used to preheat the incoming feed water for the subsequent stage. This heat exchange process results in a reduction of the energy required to raise the temperature of the feed water. Consequently, the STEC, which represents the thermal energy input needed for the process, decreases.

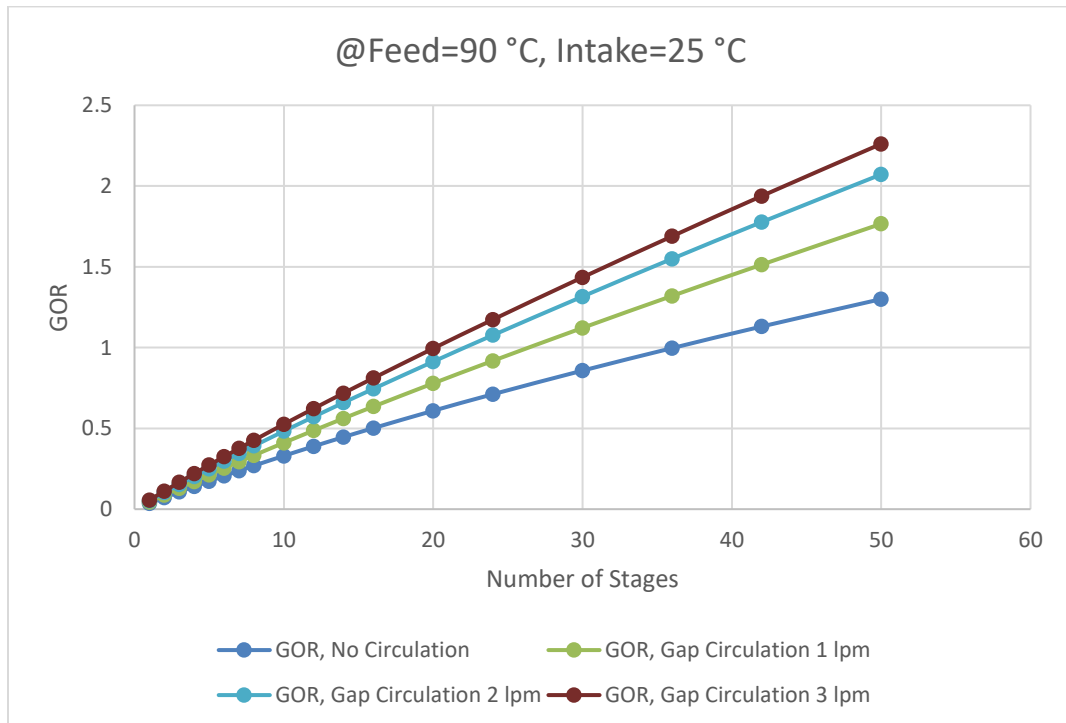
The GOR, on the other hand, is a performance metric that reflects the ratio of the heat energy utilized for distilled water production to the heat energy input required for the

process. GOR values achieved values greater than one, as shown in Figure 7.3, due to the effective utilization and recovery of latent heat of condensation within the process. In this configuration, the system recovers and reuses heat energy from freshwater condensation reducing the need for additional external heat input.

The GOR value is observed to be greater than one when increasing the number of stages, for different intake and gap circulation rates, it implies that the heat energy utilized for distilled water production is higher than the heat energy input required for the process. This can occur when the system effectively captures the condensation heat from one stage and uses it to preheat the incoming feed water in the next stage, In the case of the CFPGMD system, the efficient heat recovery between stages allows the system to leverage the waste heat generated during the condensation process to preheat the feed water in subsequent stages. As a result, the amount of external heat energy input required to raise the temperature of the feed water is reduced, leading to a decrease in the STEC value. The highest GOR was achieved at the lower intake feed rate of 1 lpm at 50 stages with a value of 3.8 for a gap circulation rate of 3 l/min which is 80 % higher than that without gap water circulation that presented a GOR value of 2.1. For different feed intake flow rate, using a gap circulator presented better performance than stagnant water gap as shown in Figure 7.3 (a), (b), (c), and (d). However, as explained for the STEC, increasing the intake flow rate presented a lower GOR value meaning that the system is less energy efficient.



(a)



(b)

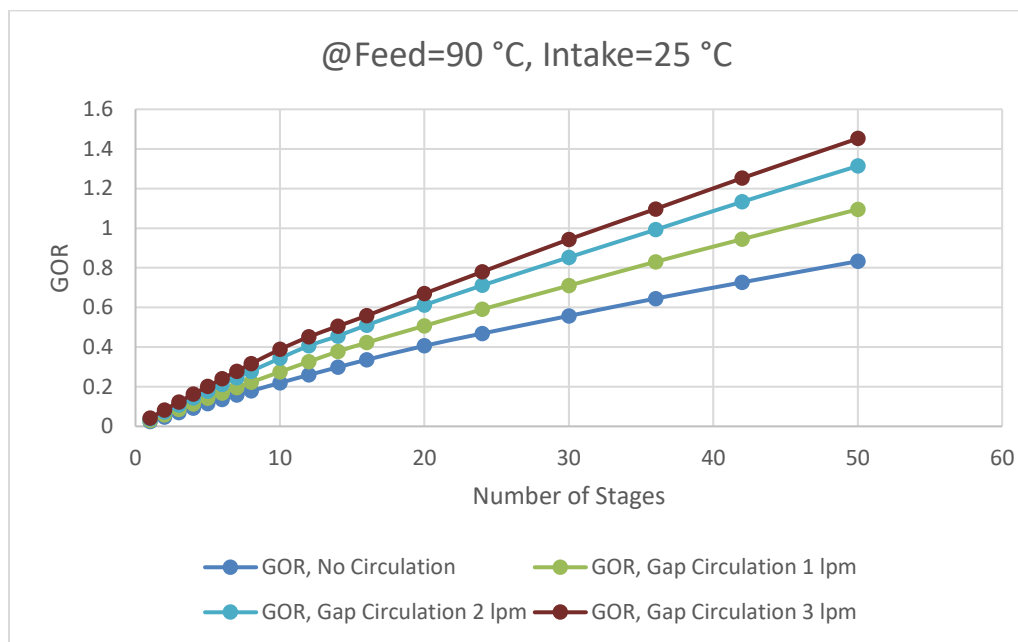
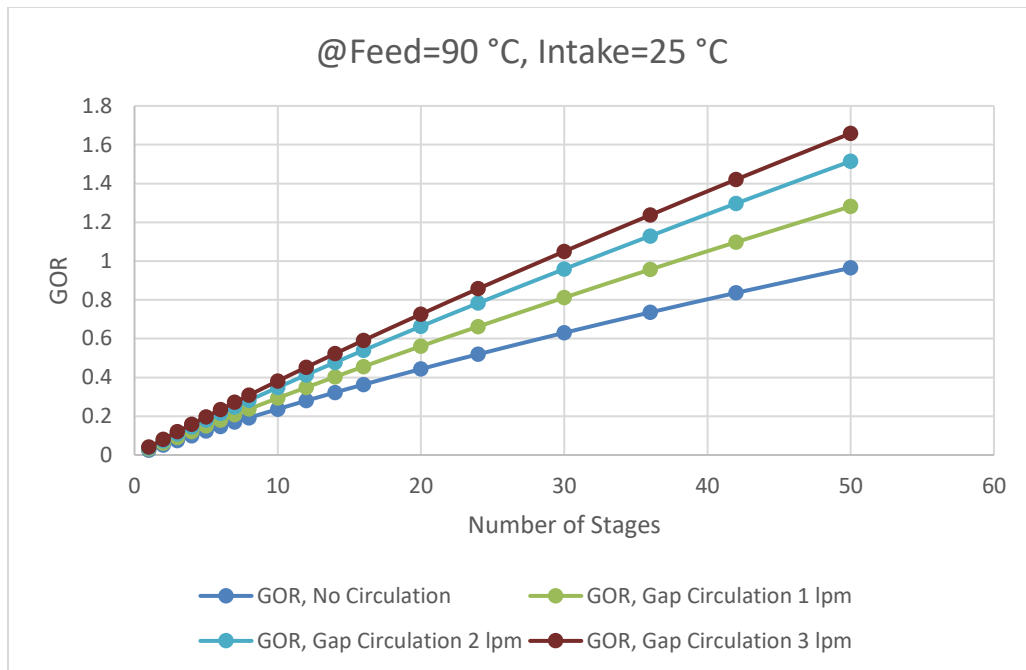


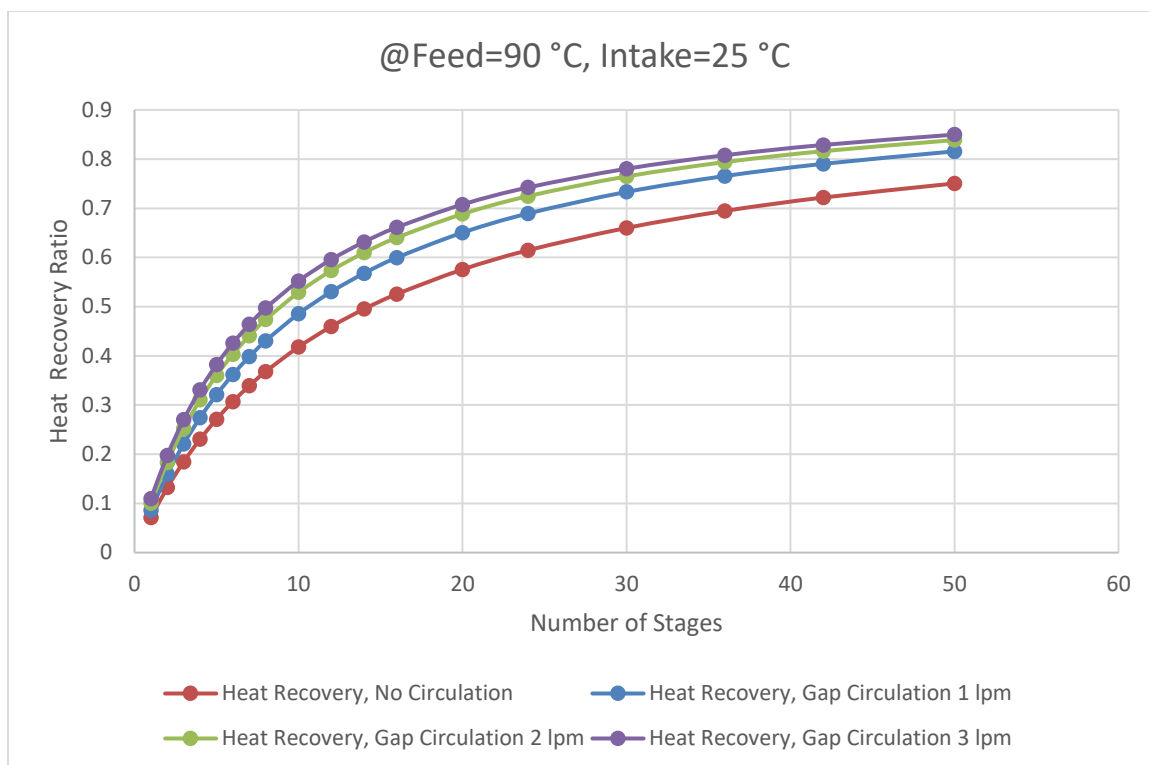
Figure 7.3 The effect of different numbers of stages on the GOR at different gap circulation rate for intake rate of 1, 2, 3, and 4 lpm in (a), (b), (c), and (d) respectively

In the counter flow CFPMD some of the heat is recovered in the coolant stream, so that a GOR value of greater than one and STEC of less than 600 kWh/m^3 are observed. The main reason is that a heat is recovered efficiently by coolant water in this configuration. The equation that defines heat recovery ratio for the CFPMD system (Equation 101) indicates that the amount of heat recovered in the system is inversely proportional to the amount of heat that needs to be added to raise the intake water temperature to the feed injection temperature. As the number of stages in the CFPMD system increases, the temperature difference between the hot and cold streams across the heating element becomes smaller due to the counter flow configuration, meaning that less input energy is required to raise the intake water temperature to the feed injection temperature.

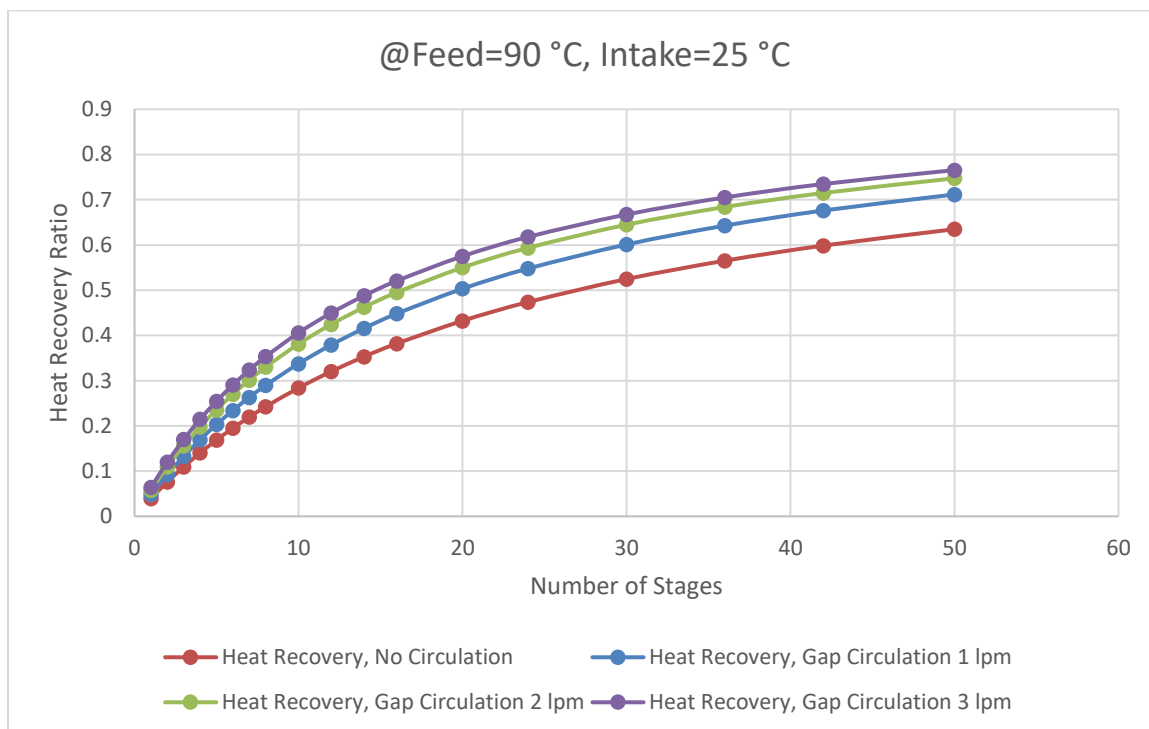
To justify, the analyses of the heat recovery ratio have been performed and presented in Figure 7.4. The results claim that more energy is recovered at lower flow rates for different stage number or feed injection temperature. The main reason is that intake water serves as a coolant for product condensation, and its temperature increases gradually after each stage as it passes through each coolant channel. At lower flow rates, the intake water spends more time in each coolant channel, allowing it to absorb more heat from the product condensation process so that it achieves higher temperature before it is introduced to the heating element. On the other hand, at higher flow rates, the intake water spends less time in each coolant channel, which limits its ability to absorb heat and increase in temperature. As a result, it enters the heating element at a lower temperature, which increases the temperature difference between the hot and cold streams and requires more input energy to raise the temperature of the cold stream to the desired feed injection temperature. This

leads to a lower heat recovery ratio and a lower energy efficiency in the system, as less of the heat generated during the membrane distillation process is recovered.

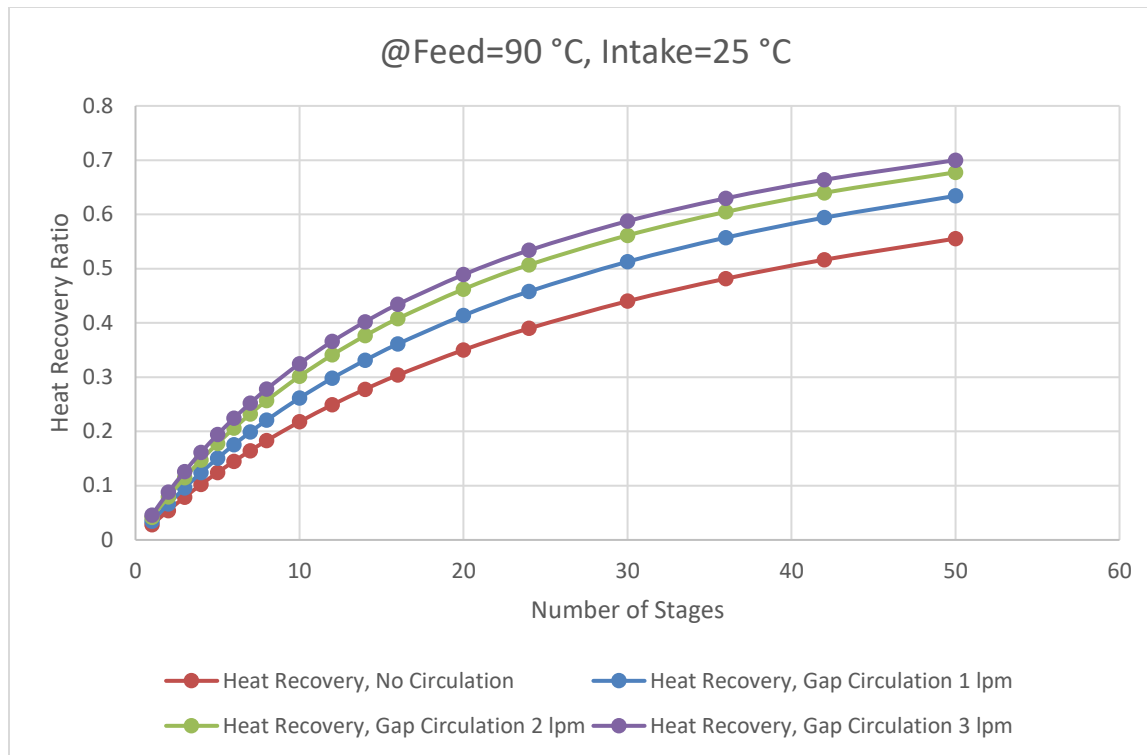
To summarize, as more stages are added to the CFPGMMD system, the hot stream is used more efficiently to transfer its heat to the cold stream, resulting in less energy needed to raise the temperature of the cold stream to the desired feed injection feed temperature. This translates to a higher heat recovery ratio and a higher energy efficiency in the system, as indicated by the higher GOR values observed in Figure 7.3 (a), (b), (c), and (d) for 50 stages compared to 1 stage. For different stage and intake flow rates, using a gap circulator of 3l/min increased the heat recovery ratio to 85%, 78% 70%, and 68% compared to 74%, 62%, 55%, and 50% for stagnant water gap at 1, 2, 3, and 4 l/min feed intake rate respectively.



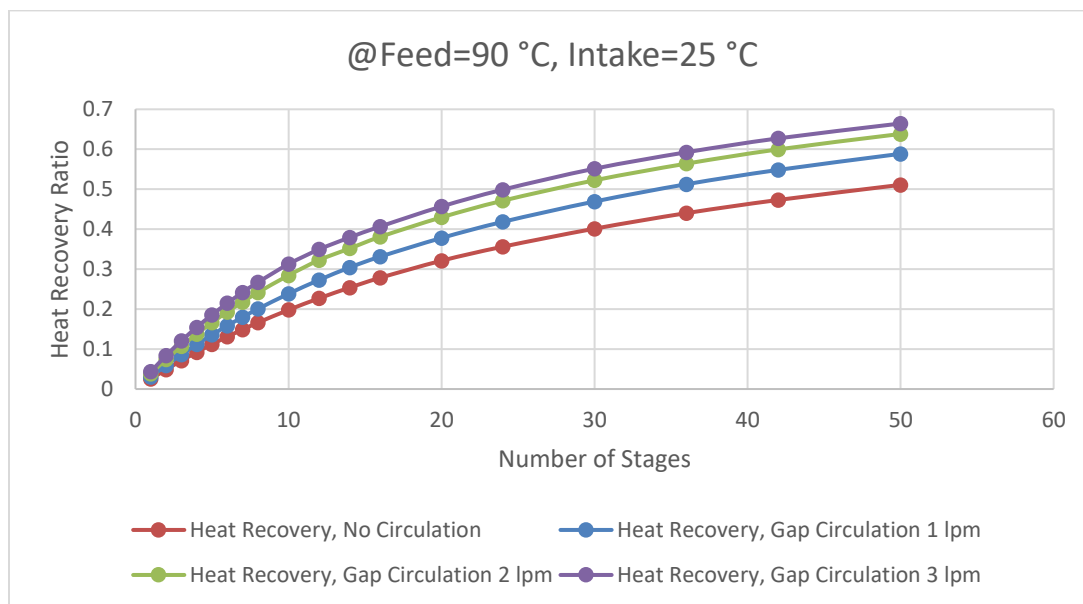
(a)



(b)



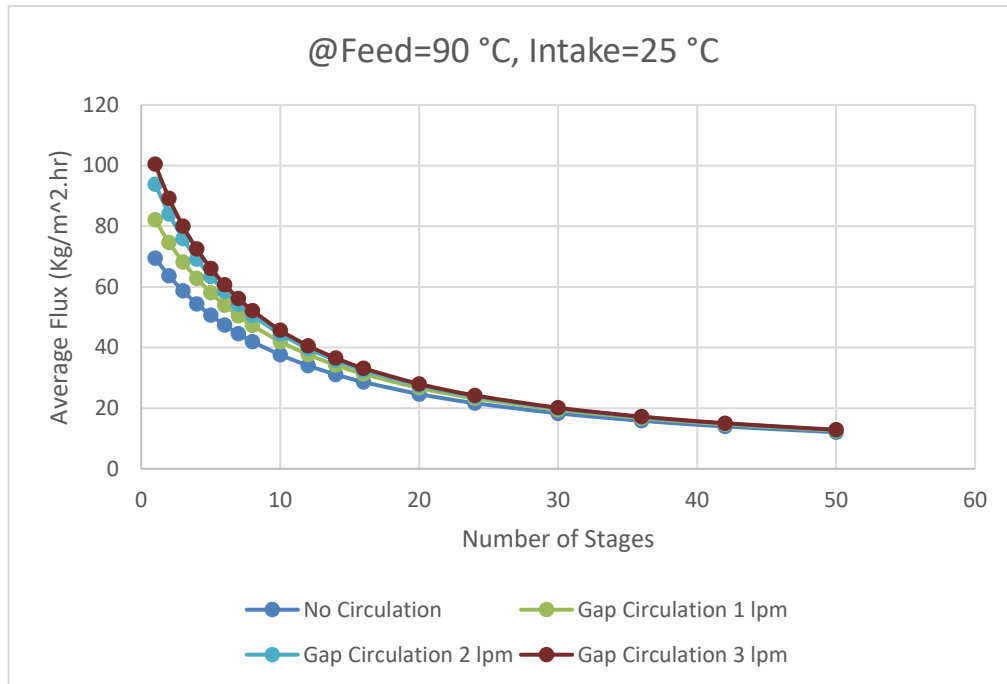
(c)



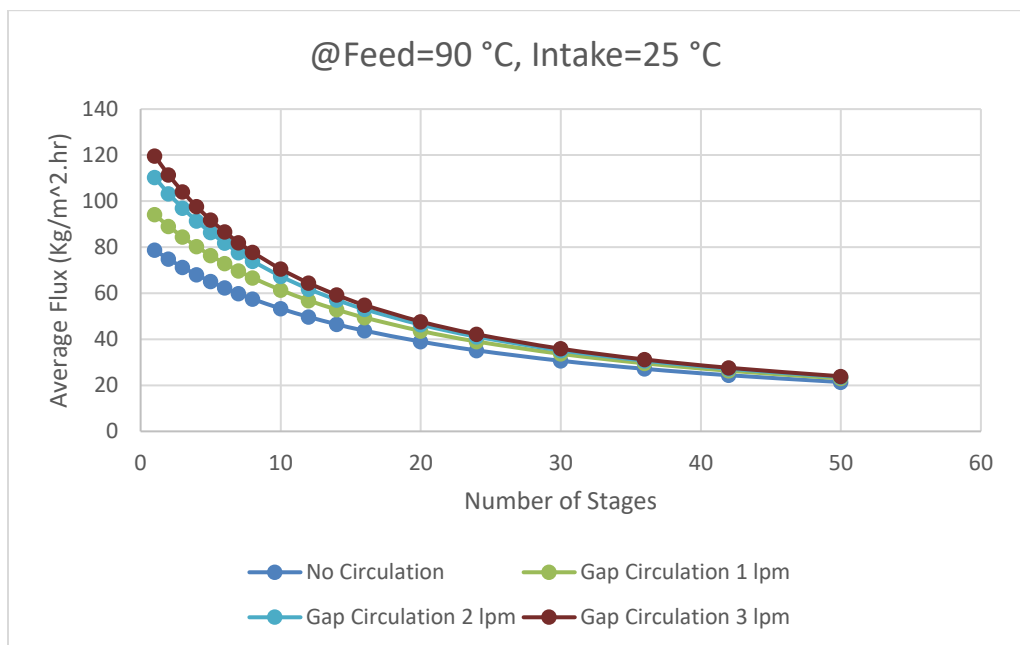
(d)

Figure 7.4 The effect of different numbers of stages on the GOR at different gap circulation rate for intake rate of 1, 2, 3, and 4 l/min in (a), (b), (c), and (d) respectively

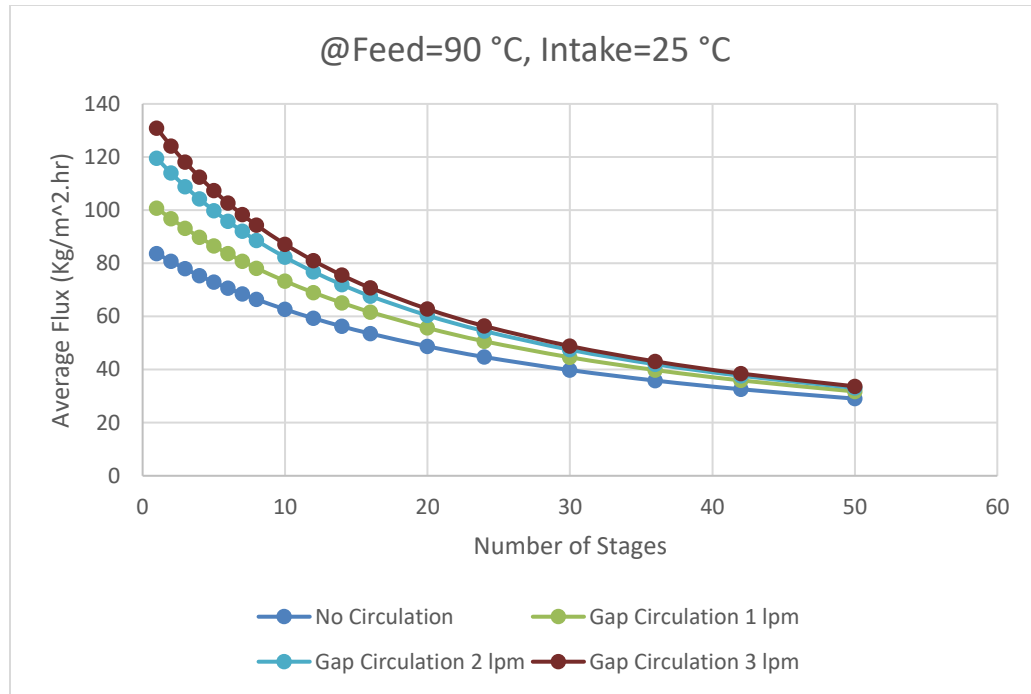
The increase of the energy consumption at higher intake flow rates is because of the increase in the mass flow rate and the decrease of temperature input to heating element, without any significant in the flux at higher number of stages. To address this issue, the analyses of Figure 7.5 are presented. The results also show that the highest flux is achieved for just one stage. This is because the temperature difference between the feed and the coolant channels is maximum, providing the highest driving force for water vapor transport across the membrane. As the intake water temperature increases in subsequent stages associated with a decrease in the feed temperature, (equations 95, and 96), the temperature difference decreases, leading to a reduction in the driving force and the average resulting flux per stage. However, for the same reasons explained in chapter 4, increasing the gap circulation rate provided an increase in the flux which is obvious for lower number of stages at different intake conditions. Indeed, more increase in the flux could be obtained for alternative design that provide more turbulence as will be shown in the following section.



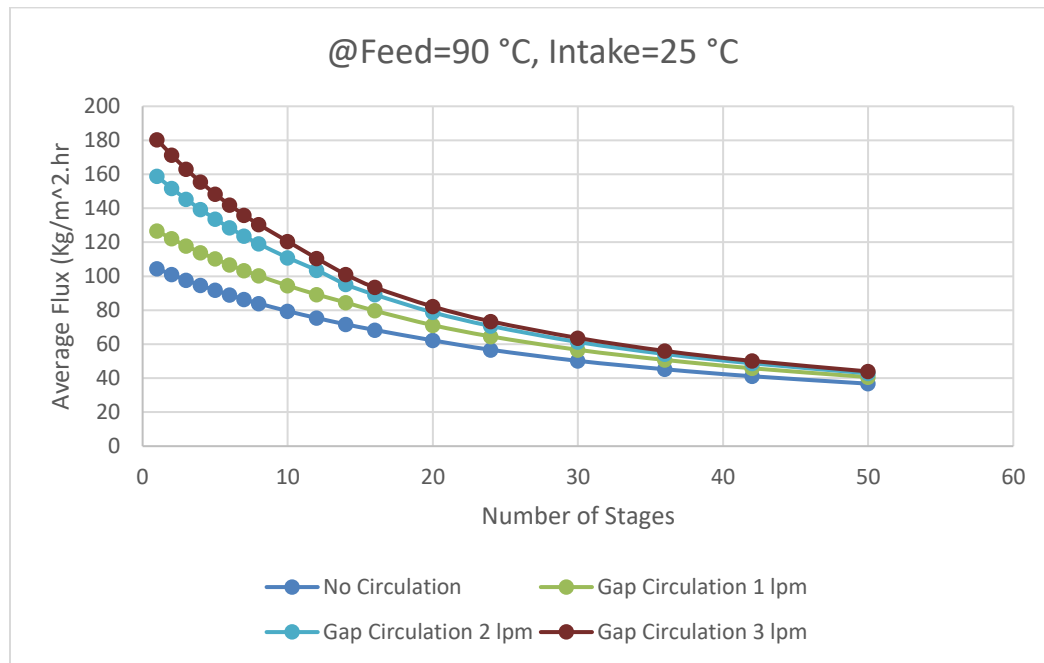
(a)



(b)



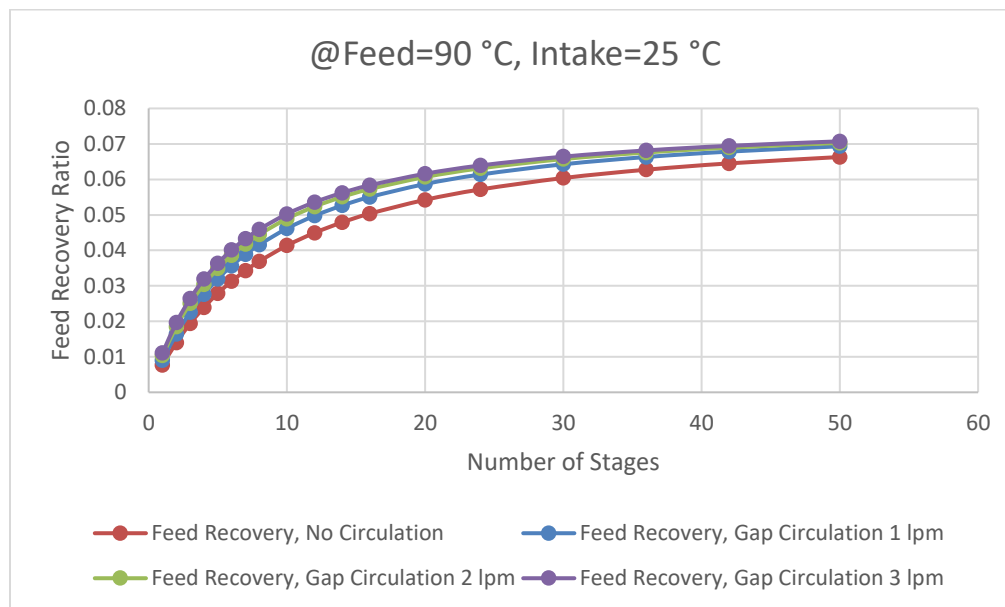
(c)



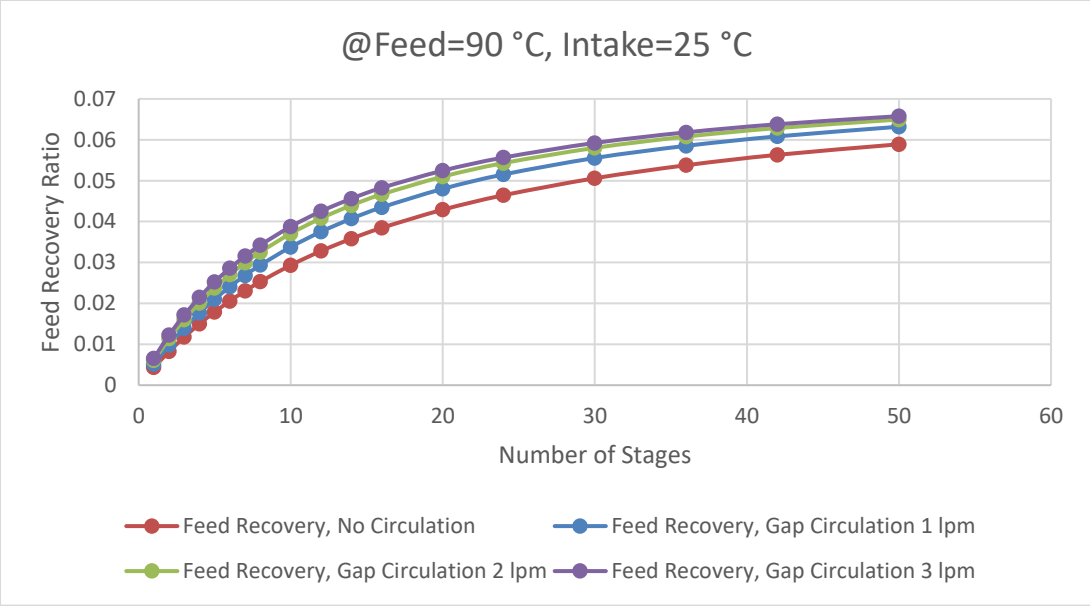
(d)

Figure 7.5 The effect of different numbers of stages on the average flux at different gap circulation rate for intake rate of 1, 2, 3, and 4 lpm in (a), (b), (c), and (d) respectively

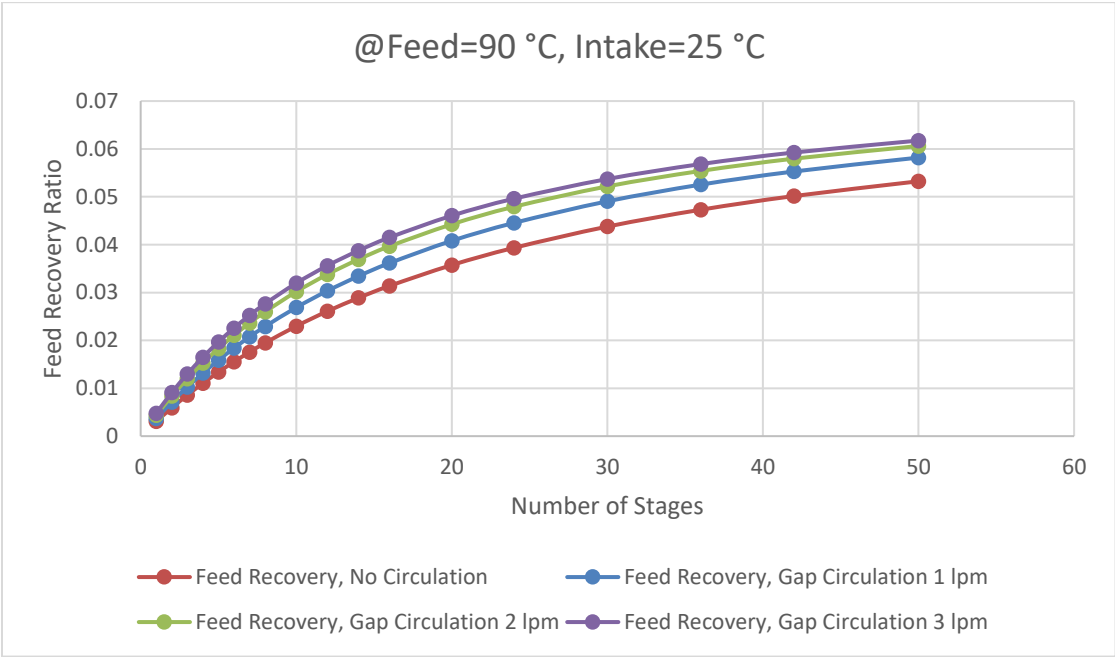
The results of Figure 7.6 represent the feed recovery ratio which is a measure of the rate of freshwater production to the rate of the intake water. The results show that as the number of stages in the CFPGM system increases, the feed recovery ratio also increases. The maximum feed recovery ratio was observed at an intake flow rate of 1 l/min for different stages number with a value of 7%, for the same reason stated before, that any increase in the intake flow would not represent an increase in the distillate proportionally. The results claim that increasing the number of stages presents a rapid increase in the feed recovery ratio for a number of stages of 15 for different conditions then only a marginal increase beyond this value. However, the impact of permeate gap water circulation is obvious in increasing the recovery ratio. A gap circulation of 3 l/min presented 7% average increase in the recovery ratio for different intake flowrates.



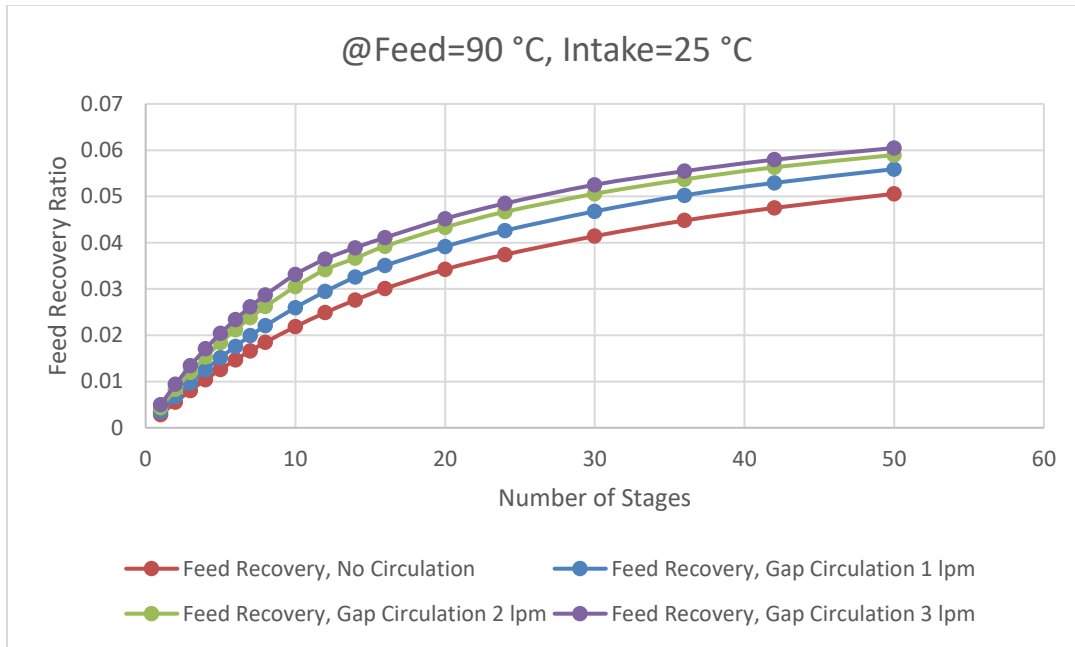
(a)



(b)



(c)



(d)

Figure 7.6 The effect of different numbers of stages on the feed recovery ratio at different gap circulation rate for intake rate of 1, 2, 3, and 4 lpm in (a), (b), (c), and (d) respectively

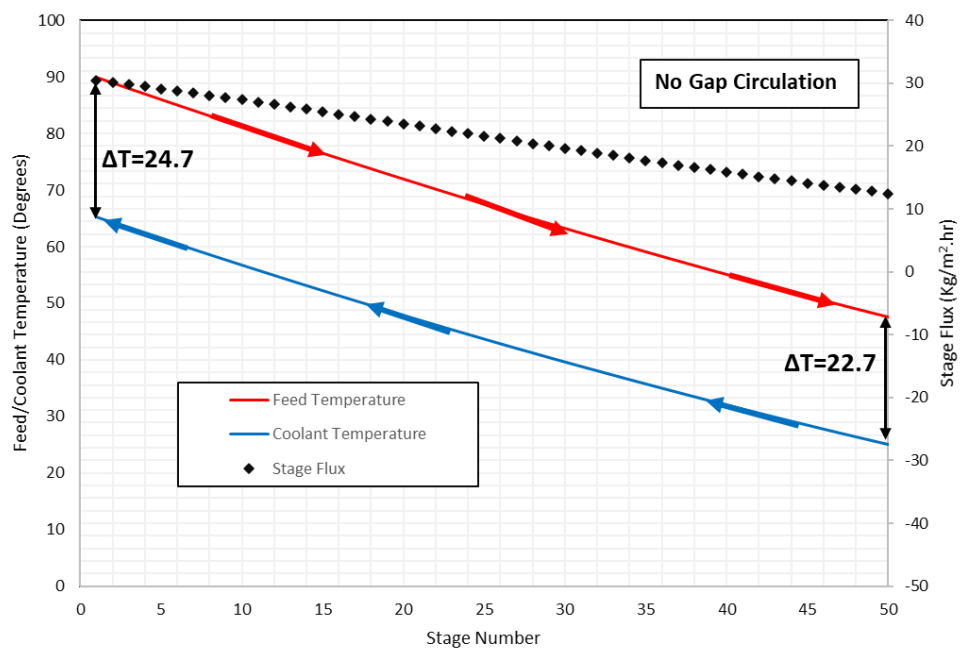
For the previous analyses, a detailed stage analysis is performed to investigate the performance of each stage separately. The detailed stage analysis is performed by the temperature difference between the feed and coolant channels of each stage and the corresponding flux. The results are presented in Figure 7.7 for the same previous conditions of intake water of 25 °C and feed injection temperature of 90 °C, however, a feed flow rate of 2 l/min is selected to be studied at different gap circulation rate.

For different gap circulation conditions, the feed temperature's decreases across the successive stage because of the ongoing heat loss due to the water vapor transfer during the membrane distillation process. Simultaneously, the temperature of the coolant, initiated at 25°C in the last stage, exhibits an increasing trend as it moves backward through the stages. This rise in the coolant temperature indicates its absorption of latent heat from the

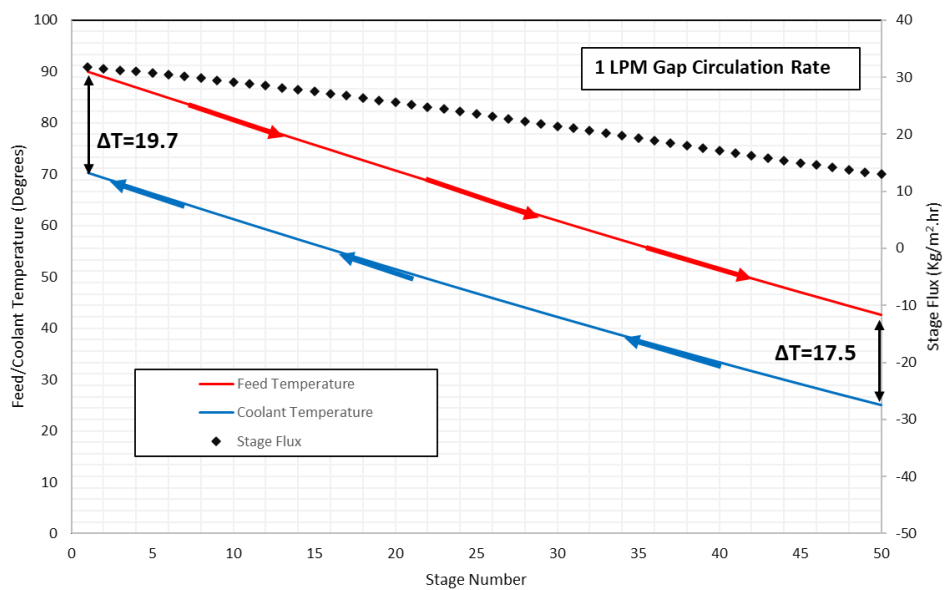
feed water, resulting in the temperature increment that continues across the distillation process.

The temperature difference between the feed water and the coolant is crucial for driving the transport of water vapor across the membrane. Typically, a larger temperature difference would mean a higher driving force for the transport of water vapor across the membrane, leading to an increased flux. However, the gap circulation rate is another key factor that can influence this relationship, affecting the heat transfer rate and consequently the temperature difference and flux. The observations of Figure 7.7 claims that as the gap circulation rate increases, the temperature difference between the feed and the coolant in each stage reduces because of the enhanced convective heat transfer resulting in a more efficient warming of the coolant and cooling of the feed. To illustrate, the average temperature difference between the hot and cold side reduced from an average of 23.7 °C when no gap circulation is presented by 36% to 15.1 °C when a gap circulation rate of 3 l/min is used. Although the overall temperature difference might decrease, the enhanced convective heat transfer could potentially result in a higher flux, due to a more efficient heat exchange and subsequent water vapor transfer, as shown in Figure 7.7. It is worth to mention that the decrease in the temperature difference between the hot and cold side did not translated to a lower flux. Instead, the flux was higher, as shown in Figure 7.7 due to the improved heat transfer efficiency, despite the lower temperature difference.

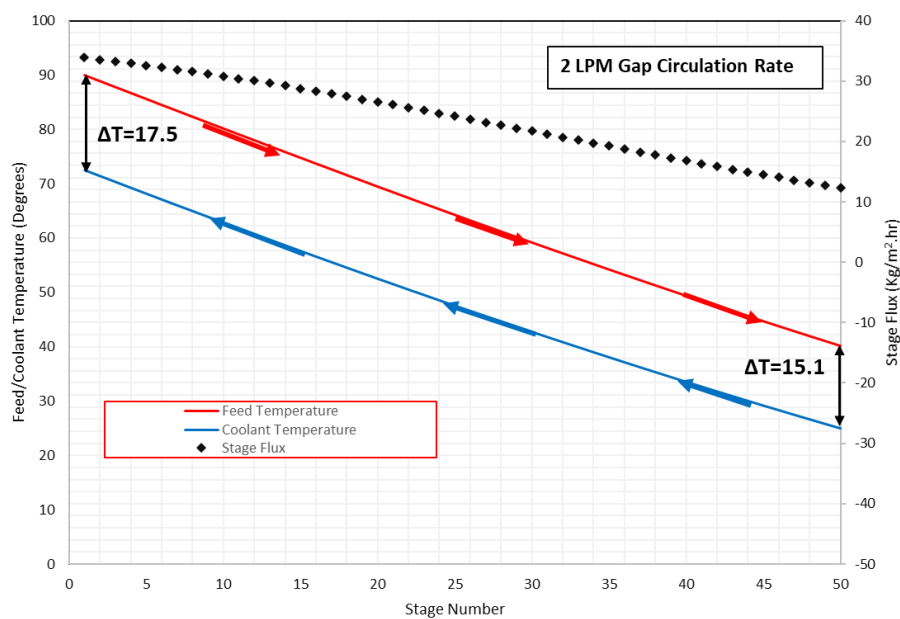
The data also illustrates a consistent decline in flux with each subsequent stage. This is anticipated as the feed temperature decreases and the coolant temperature increases with each stage resulting in a decreased vapor pressure according to Antoine's equation leading to decrease in the flux.



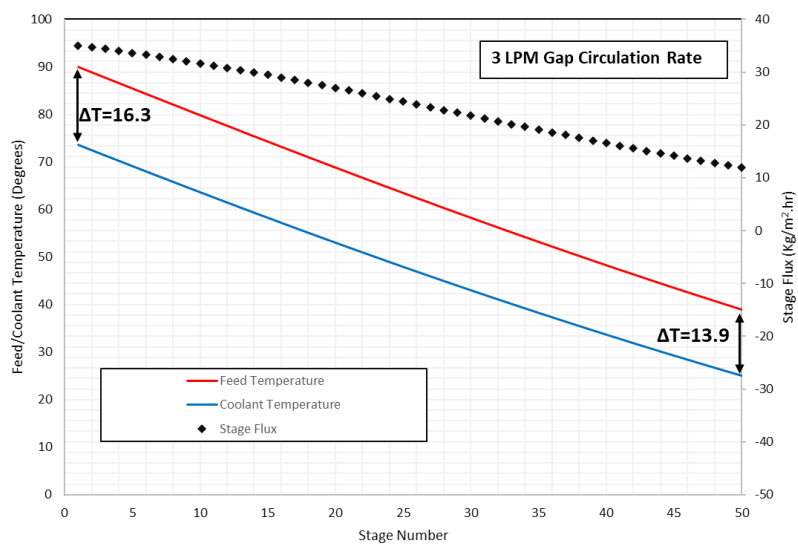
(a)



(b)



(c)



(d)

Figure 7.7 The temperature distribution of the bulk feed and coolant temperatures across each stage of the counter flow PMGD without gap circulation (a) and with 1, 2, and 3 lpm for (b), (c), and (d) respectively

The results in Figures 7.8 to 7.11 aims to investigate the effect different feed injection temperature on the same metrics parameters (STEC, GOR, average flux, and feed recovery ratio) while keeping the same operating parameters constant. For simplicity, the analyses are performed once for a feed intake rate of 2 l/min, gap circulation rate of 3 l/min and similar intake temperature for the previous analyses at 25 °C. It is shown that the for higher injection temperature the system presented lower STEC, which means the system is more efficient at higher temperature. For example, the STEC between a feed injection temperature of 90 °C is 28% and 30 % lower than using a feed injection temperature of 60°C for 5 and 50 stages, respectively. However, increasing the number of stages results in a decrease in the STEC, providing more efficient system, for different injection temperatures. This decrease breaks the value of 600 kWh/m³, meaning that the system recovers its energy for different feed injection temperature.

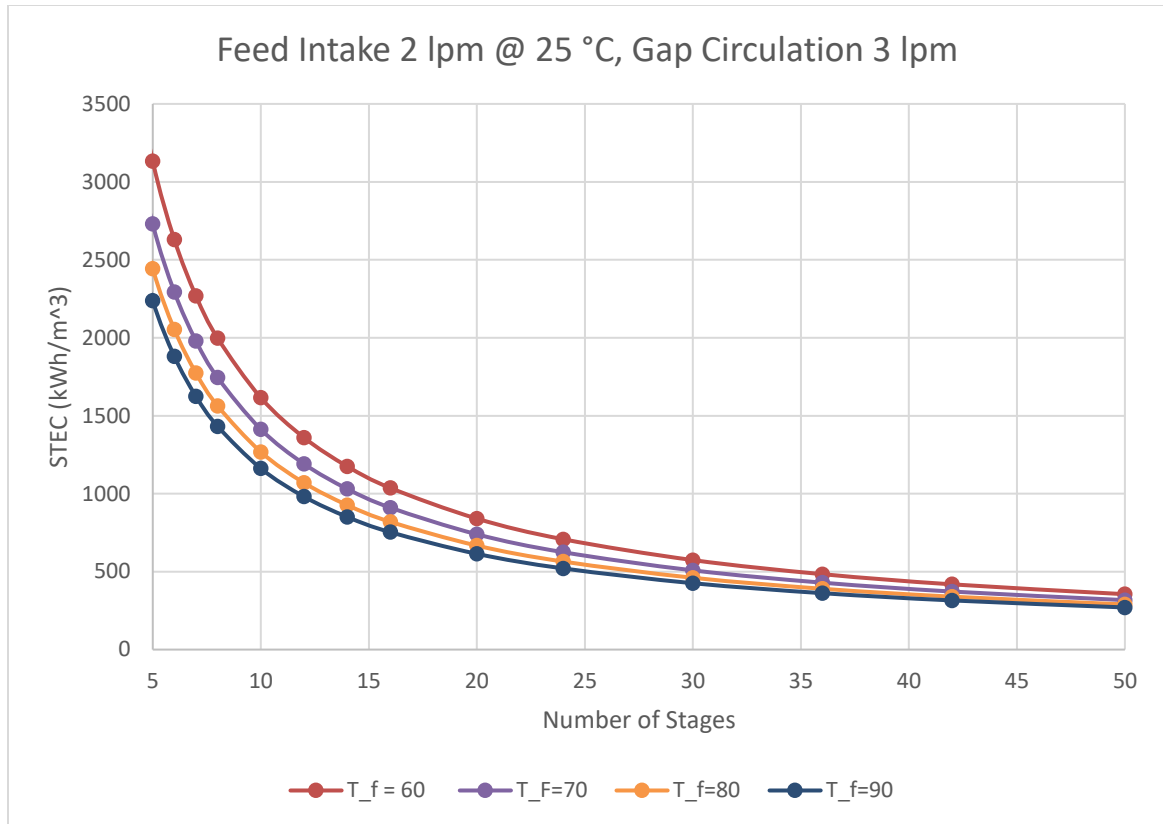


Figure 7.8 the effect of different numbers of stages on the STEC at different feed injection temperature

The analyses of the previous figure is supported through the values of GOR value, presented in Figure 7.9, That shows higher GOR values at higher feed injection temperature, which also increases as the number of stages increase to break the Value of 1.0 meaning that the distillate energy of vaporization is greater than the energy input because of the heat recovery.

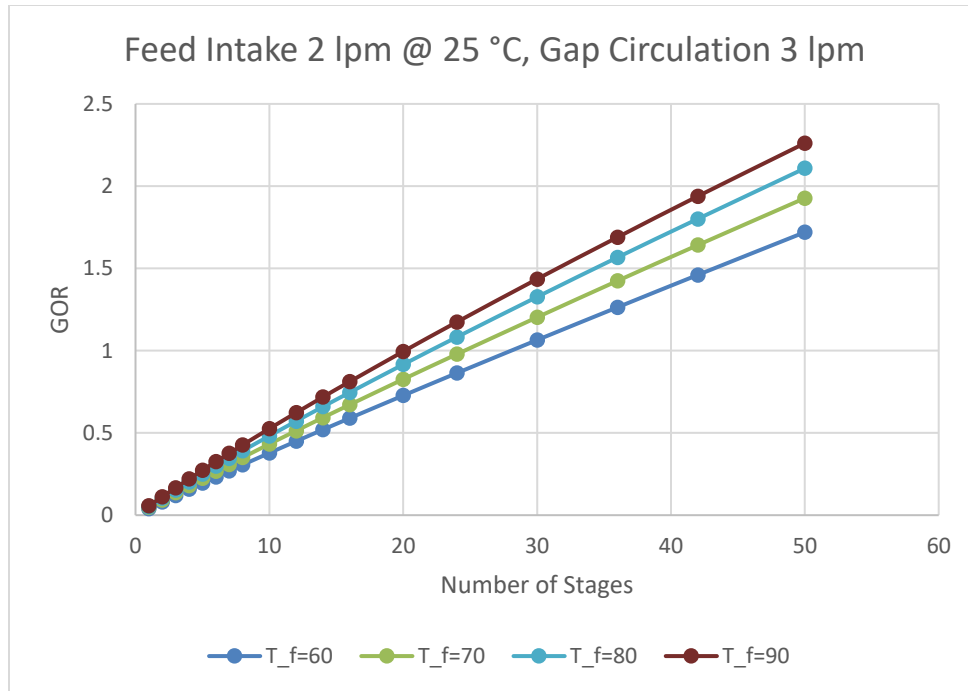


Figure 7.9 the effect of different numbers of stages on the GOR at different feed injection temperature

The reason of the better energy performance at higher feed injection temperature is that the system provides better flux because of the higher driving force of mass transfer for higher temperatures. This is shown in Figure 7.10 that claims a 150 % and 65% more flux when using 5 and 50 stages respectively when increasing the feed injection temperature of from 60 °C to 90 °C. Even though increasing the number of stages decreases the average flux, as previously explained, the system presented higher energy efficiency performance because the heat recovery characteristics.

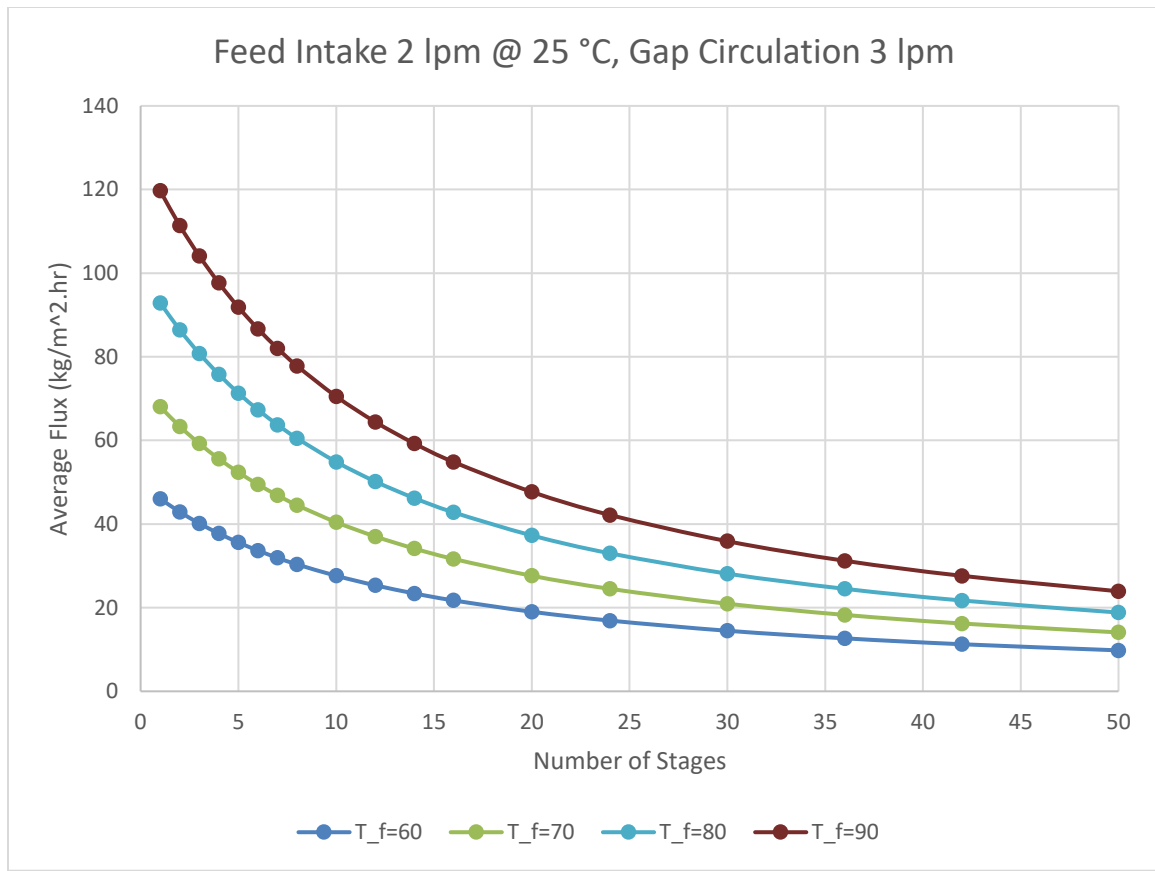


Figure 7.10 the effect of different numbers of stages on the Average Flux at different feed injection temperature

Regardless of the decrease in the average flux, the system presented an increase in the feed recovery ratio when increasing the number of stages. In a multi-stage membrane distillation system, the feed water passes through several stages where it undergoes partial desalination in each stage. The added stages provide more opportunities for water vapor to be separated from the feed water, resulting in a higher overall recovery of freshwater. This is because the additional stages provide more opportunities for the separation of water vapor from the feed water, resulting in a greater total amount of freshwater produced. However, it is essential to note that there is a diminishing return on increasing the number of stages, as adding more stages may also increase the complexity and costs associated with the system. Therefore, an optimal number of stages should be determined based on the desired recovery

ratio, energy efficiency, and economic factors. Numerically, increasing the feed temperature from 60 °C to 90 °C presented 135 % increase in the feed recovery ratio achieving a value of 6.6% for the studied conditions at 90 °C for 50 stages.

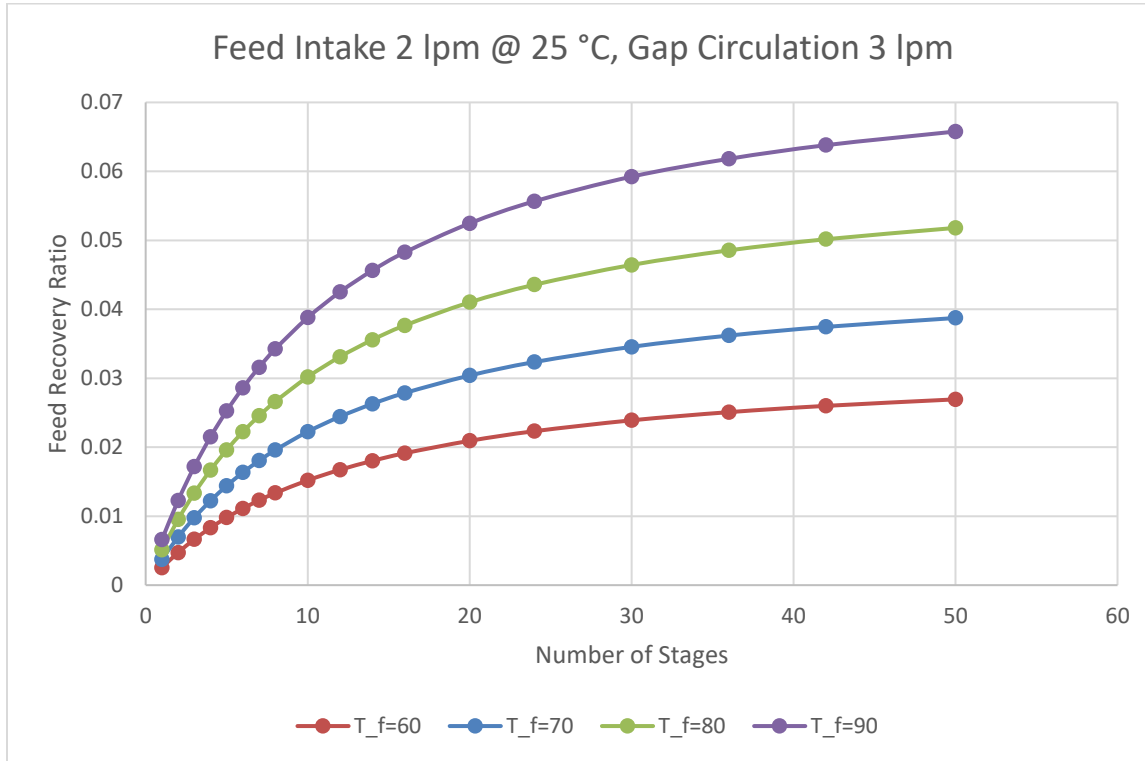


Figure 7.11 the effect of different numbers of stages on the feed recovery at different feed injection temperature

The analysis of varying the coolant intake temperature on the performance of a multi-stage membrane distillation system, has been performed and presented in Figure 7.12 to Figure 7.15. The coolant intake temperature is the temperature at which the coolant, typically the intake water, is introduced into the system to condense the water vapor on the cooler side of the membrane. In this analysis, the coolant intake temperature is varied between 10, 15, 20, and 25 °C, with a fixed feed injection temperature of 90 degrees Celsius with same feed injection and gap circulation rates of 2 l/min and 3 l/min, respectively.

At a lower coolant intake temperature (e.g., 10°C), the temperature difference between the feed water and the permeate gap is higher. This greater temperature difference provides a higher driving force for mass transfer, resulting higher flux and feed recovery ratio as shown in Figures 7.12-7.15. However, it is a margin increase in the flux, and it is only 12 % for 5 stages and 8 % for 5 stages. Indeed, the system presented a higher energy efficiency in terms of STEC and GOR at higher feed intake temperature (e.g., 25°C) because of the reduction in the energy needed to heat the feed water at the last stage, as the incoming coolant is already closer to the desired feed injection temperature. This presented in lower STEC and higher GOR values for higher intake temperature. In other words, higher intake temperatures (e.g., 25°C) promote better heat recovery across the stages, as the temperature difference between the feed water and the coolant decreases. The enhanced heat recovery allows the system to use less energy to achieve the desired feed injection temperature, improving the overall energy efficiency.

Optimal coolant intake temperature: The optimal coolant intake temperature should be determined based on the desired performance, energy efficiency, and economic factors. A balance must be achieved between the benefits of a higher coolant intake temperature, such as increased energy efficiency and GOR, and the potential drawbacks, such as slightly lower flux and feed recovery.

In conclusion, varying the coolant intake temperature in a multi-stage membrane distillation system can have a significant impact on its performance. The optimal temperature should be determined based on a comprehensive analysis of the system's mass transfer rate, energy efficiency, feed recovery ratio, and potential for fouling, while considering the economic and practical aspects of the system's operation. However, using

lower intake temperature could leads to the requirement of an external cooling device that increases the energy consumption and the overall cost of the water product. In facts, the feed intake temperature is determined by the environmental and external weather conditions.

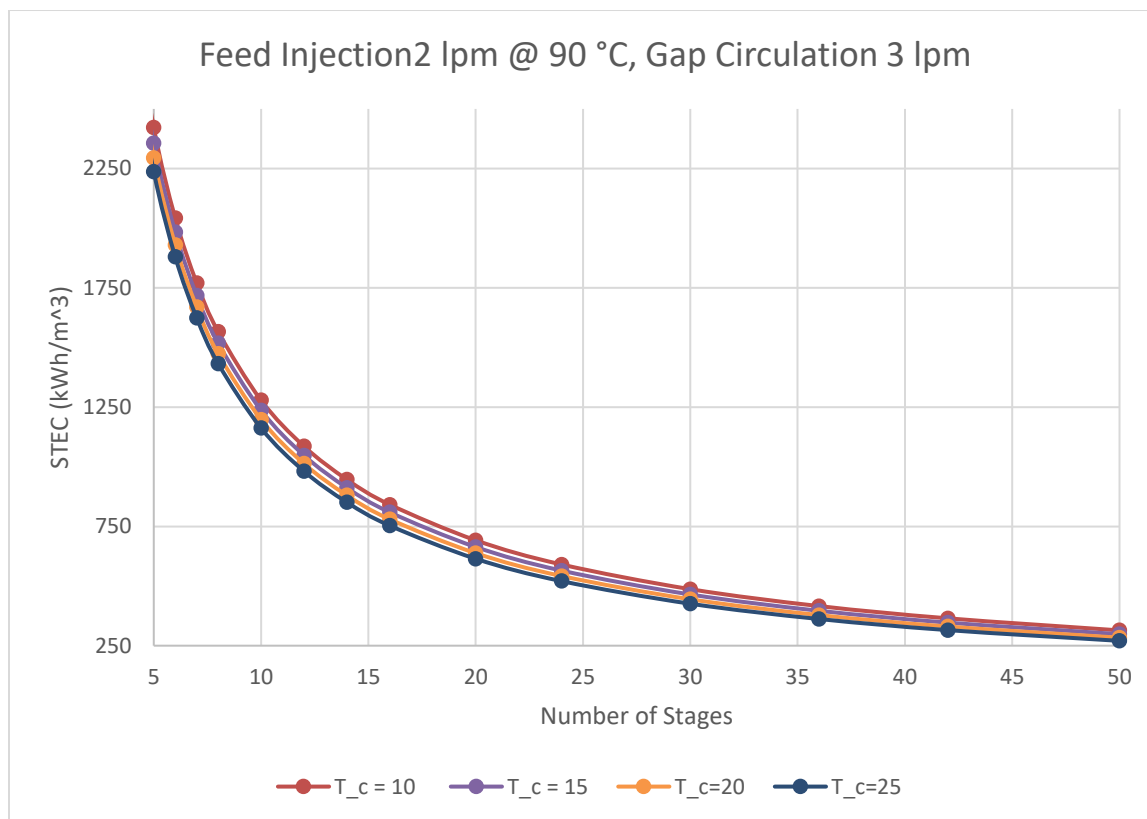


Figure 7.12 the effect of different numbers of stages on the STEC at different feed intake temperature

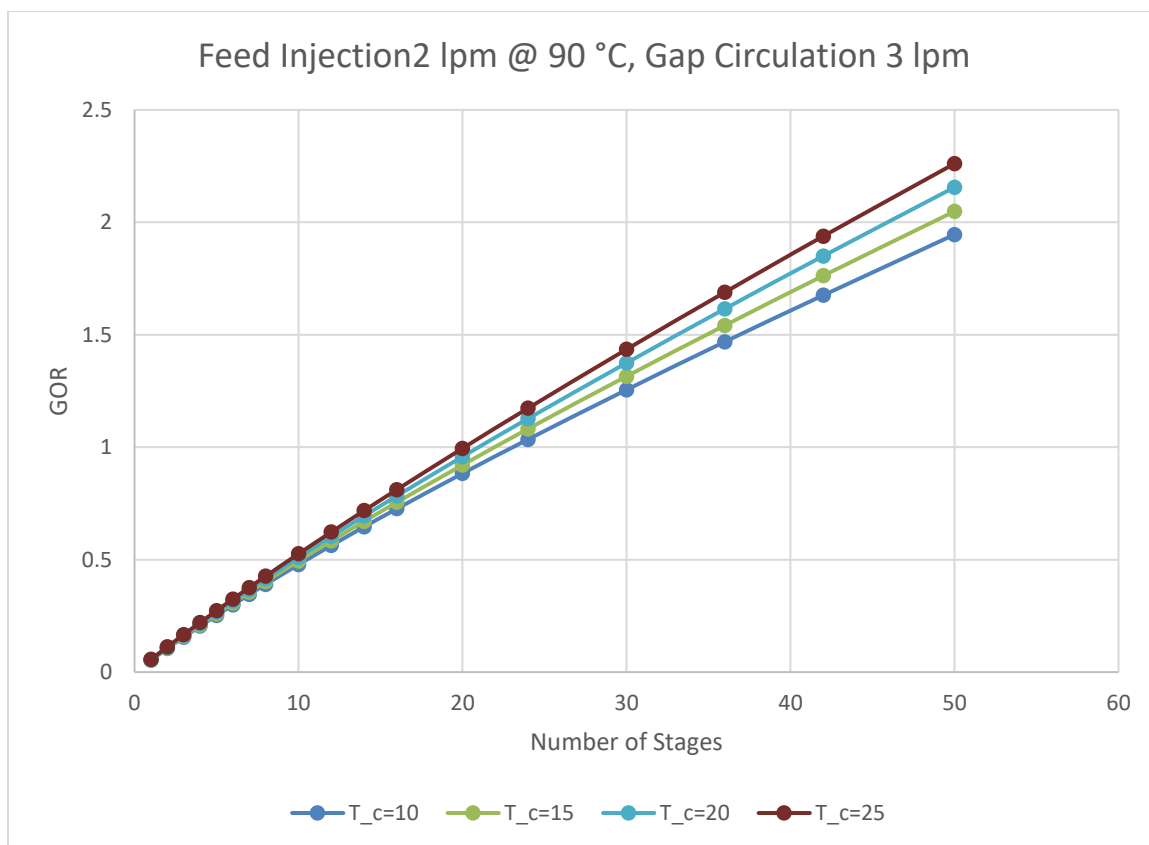


Figure 7.13 the effect of different numbers of stages on the GOR at different feed intake temperature

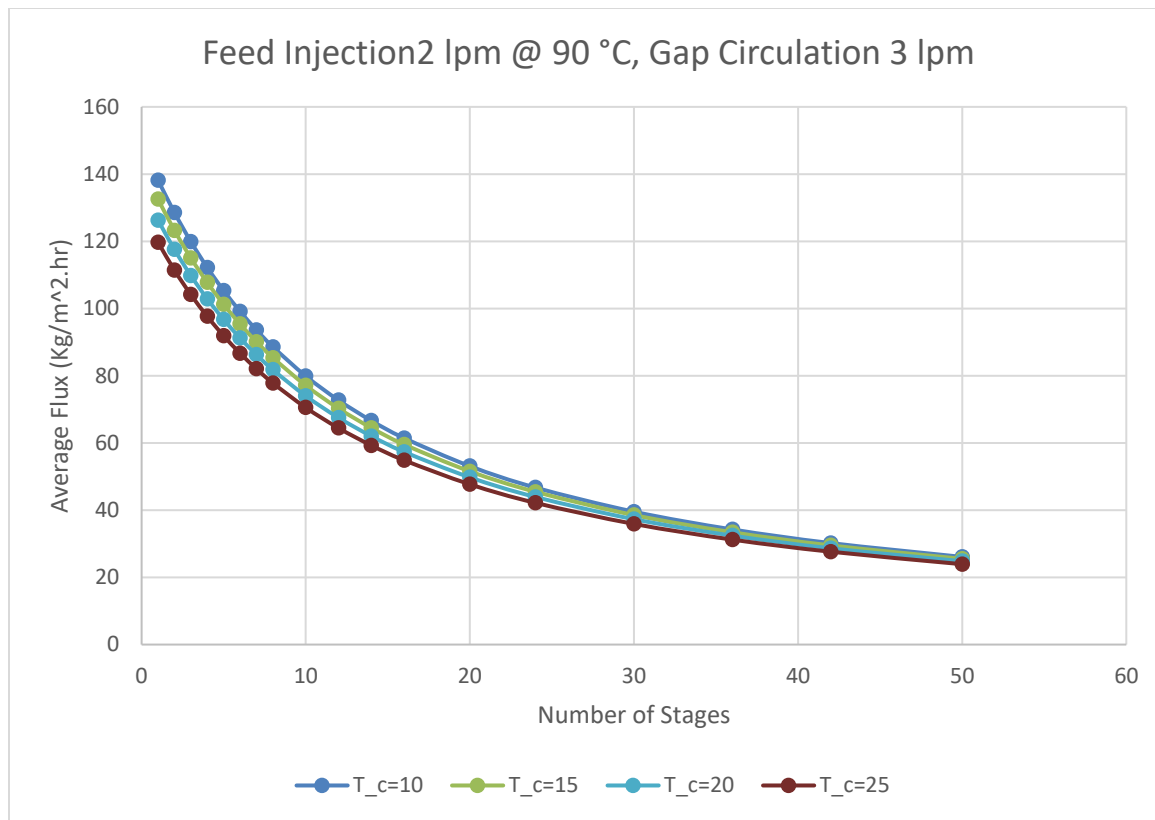


Figure 7.14 the effect of different numbers of stages on the Average Flux at different feed intake temperature

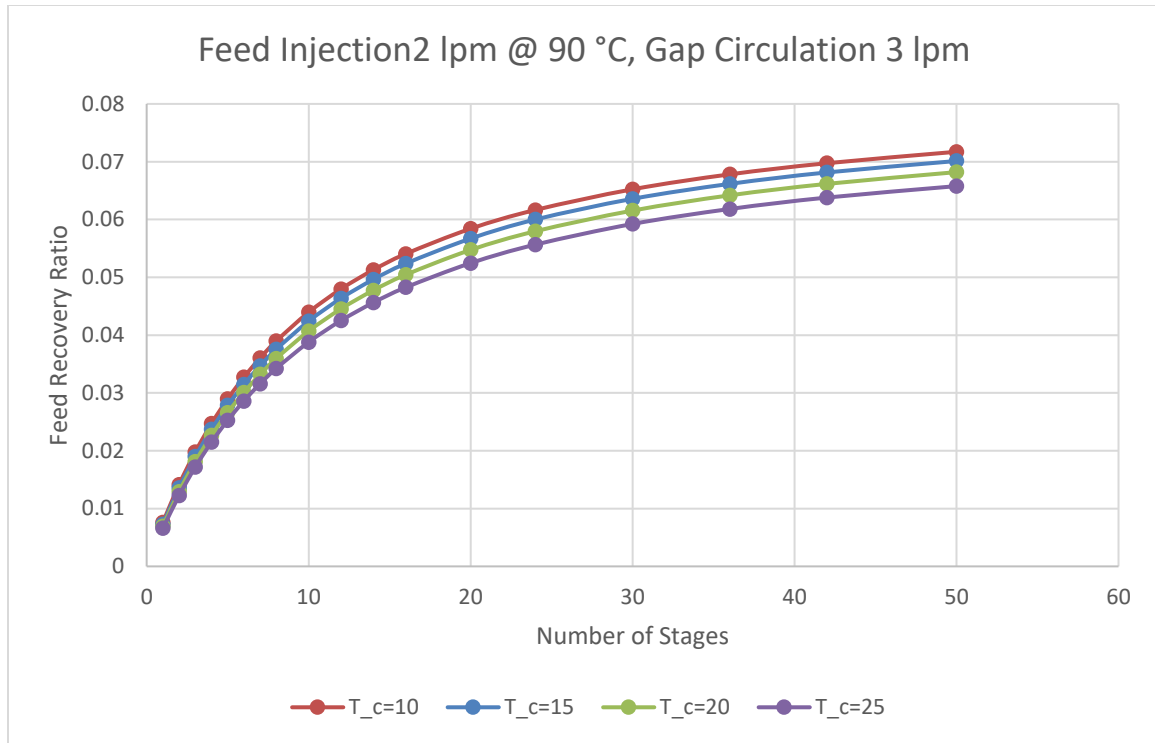


Figure 7.15 the effect of different numbers of stages on the feed recovery at different feed intake temperature

7.2. Alternative Designs and Configurations

While the counterflow multi-stage permeate gap membrane distillation (CFPGMD) system demonstrates promising results in terms of energy efficiency and feed recovery, there are certain limitations associated with increasing the number of stages and incorporating gap circulation pumps. To address these challenges and further optimize the performance of membrane distillation systems, alternative designs and configurations can be considered such as:

1. Enhanced turbulence and mixing: Modifying the chamber configuration to create increased turbulence and mixing can enhance the driving force for water vapor transfer across the membrane. By promoting more uniform temperature and

concentration profiles, this approach can help to maintain higher flux and recovery rates, even in systems with a reduced number of stages.

2. Larger membrane area: Utilizing larger membrane areas within the PGMD stages can lead to improved mass transfer and heat recovery, resulting in better overall performance. This can be achieved by increasing the length, and width of membrane layers within each stage.

7.2.1. Analyses of Alternative Feed, Coolant, and Gap Chambers Deign

In this modified design, the chamber configuration is adjusted to have channels with a width of 2 cm each, instead of the original design with two channels of 6 cm width each as shown in Figure 7.16.

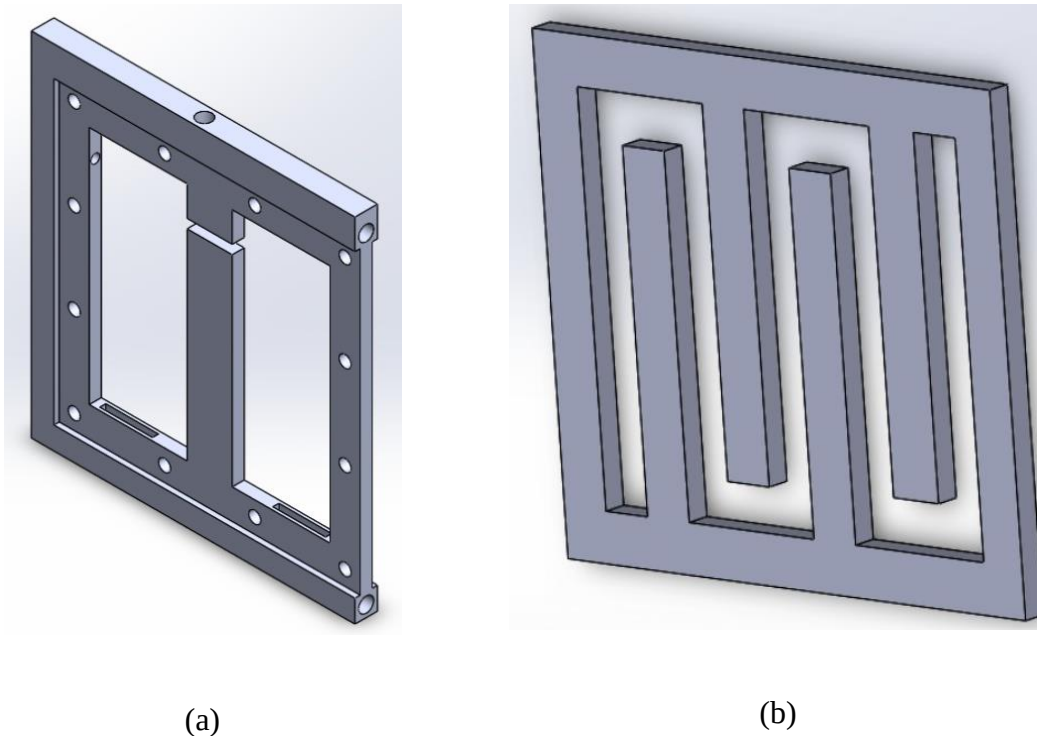


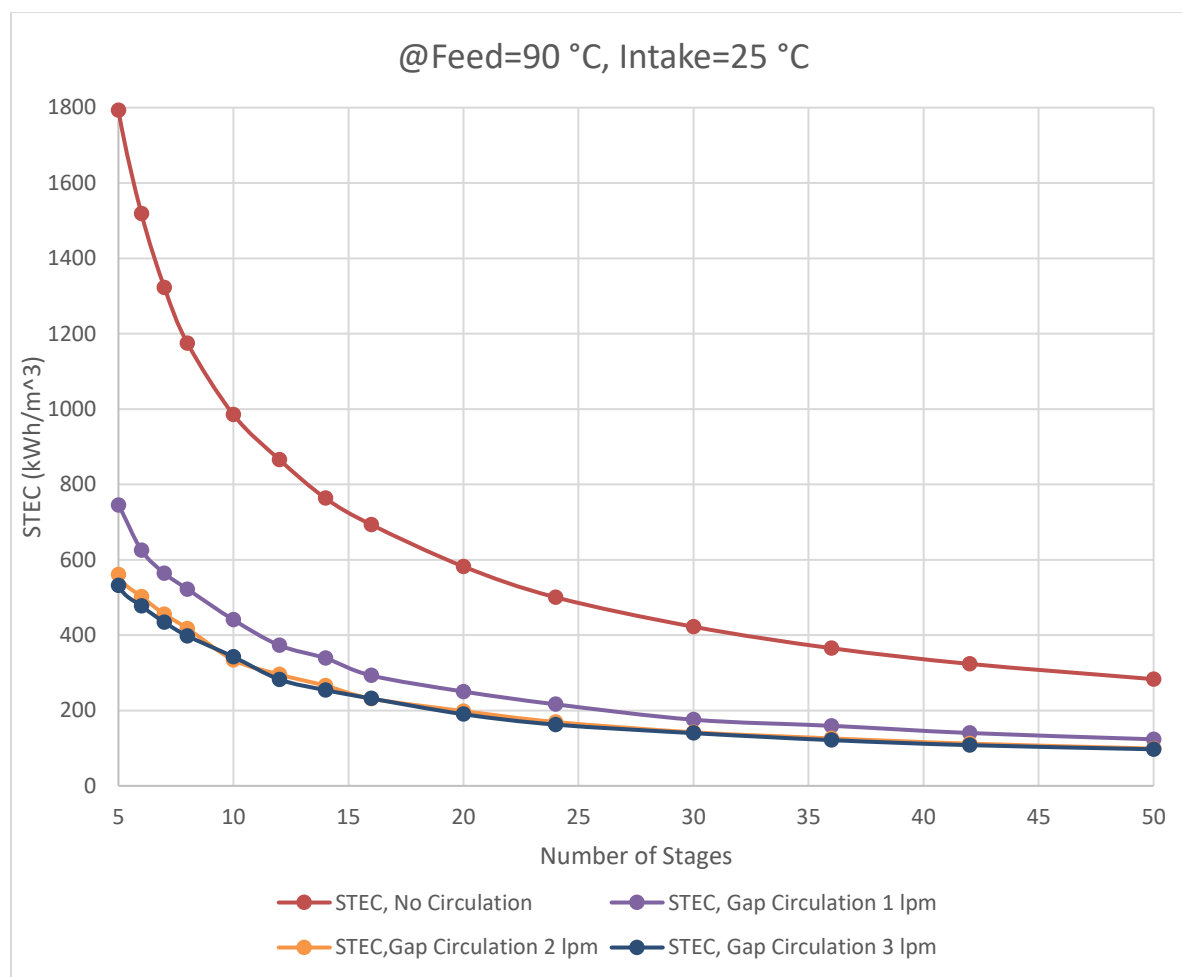
Figure 7.16 The old (a), and roposed (b) feed, coolant, and gap channels.

With narrower channels, turbulence and mixing within the feed and coolant streams are boosted leading to more uniform temperature profiles across the membrane surface, which, in turn, improved the driving force for water vapor transfer. Moreover, the reduced channel width can result in higher velocity flows, which can improve convective heat and mass transfer coefficients. This enhanced heat and mass transfer can potentially lead to increased flux rates and energy efficiency in the membrane distillation process.

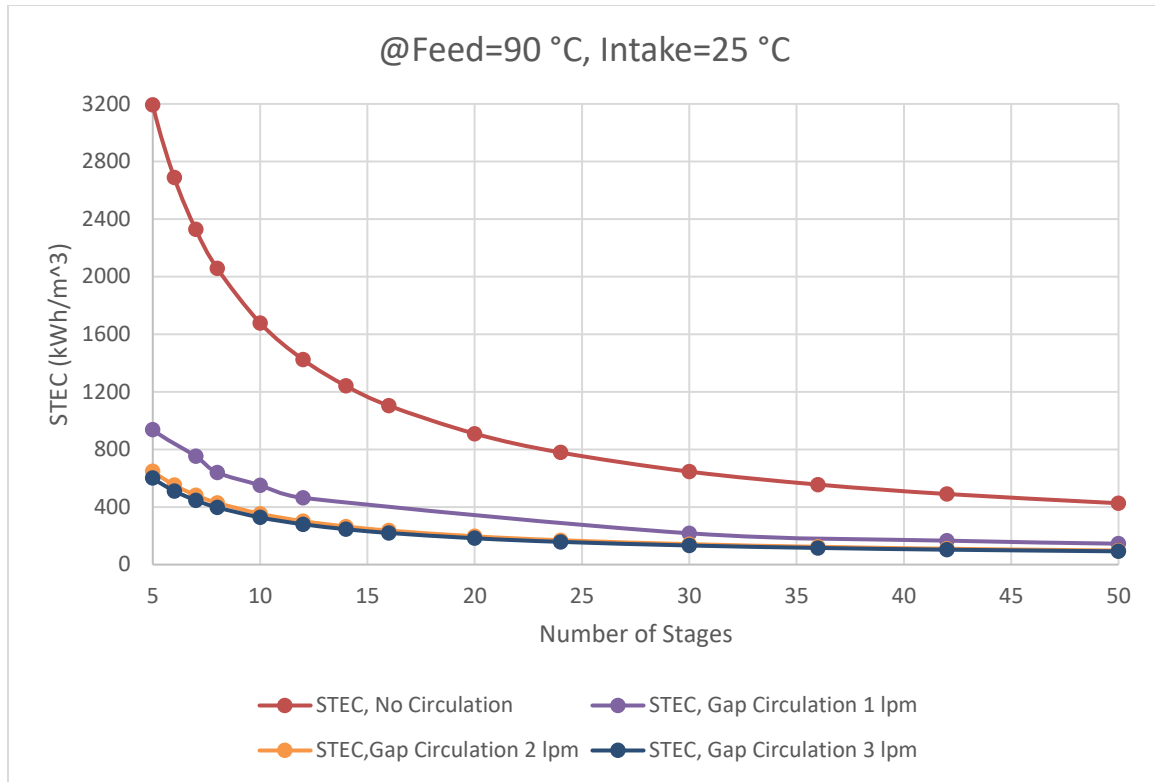
This increase in the vapor transfer driving force is attributed in the enhanced flux as shown in Figure 7.20. Comparing the 2 l/min feed intake and 3 lpm circulation of the new design to the old design, it shown a 112%, and 18% increase in the flux for 5, and 50 stages, respectively. The more percentage increase in the average flux is shown for lower number of stages. This increase in the flux allowed more freshwater condensation in water gap, which in turns transmitted to coolant flow allowing it to raise to values closer to the feed temperature making the system more energy efficient. To illustrate, the new system configuration broke the minimum obtained value of 600 kWh/m³ with just 5 distillation stages whereas the old design broke this value with 20 distillation stages at a feed of 2 l/min and 3 l/min gap circulation rate. The use of narrower channels can lead to a more compact system design, that can use less MD stages, which may be advantageous in terms of space requirements, materials usage, and overall system cost. In the old design, the poor mixing resulted in a margin increase in the flux as the feed rate increases making 1 l/min feed intake rate more suitable for energy recovery. However, in the enhanced mixing design, increasing the feed rate boosted the average flux proportionally to the increase amount of energy making a feed intake rate of 2 l/min more suitable.

It is also shown that the turbulent mixing, and specially in permeate water gap, resulted in an enhanced performance comparing to the stagnant permeate water gap as shown in the Figures 7.17 to 7.21.

The enhanced water mixing, in different channels, allowed the STEC to decrease 77% for a gap circulation rate of 3 l/min compared to the stagnant water achieving a minimum value of 96 kWh/m³ at 50 distillation stages.



(a)

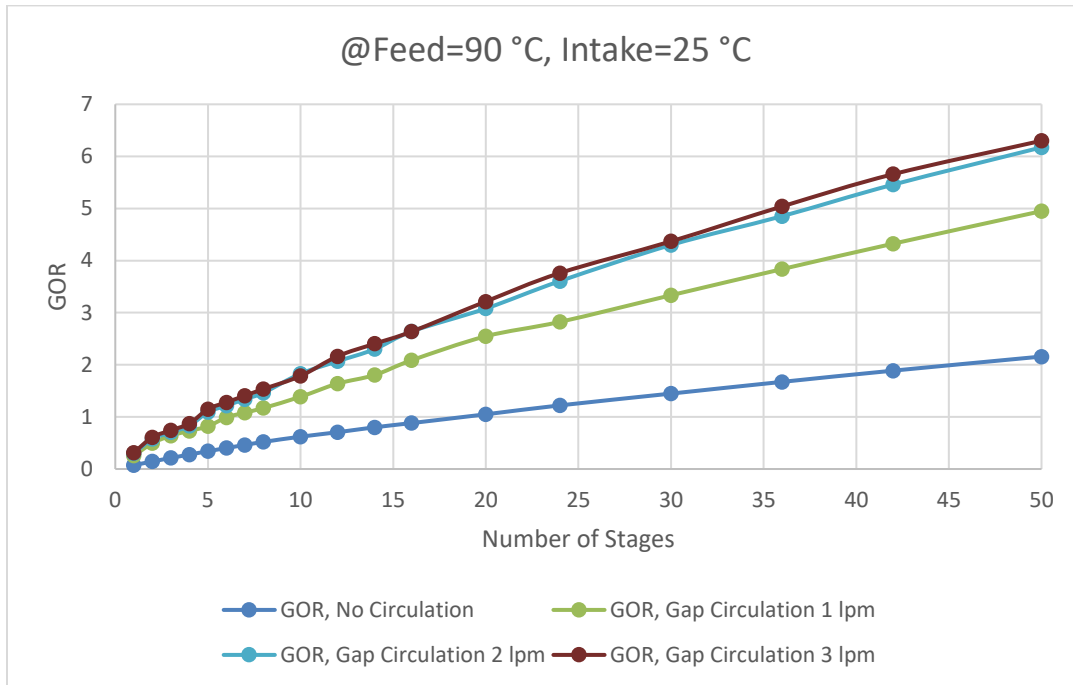


(b)

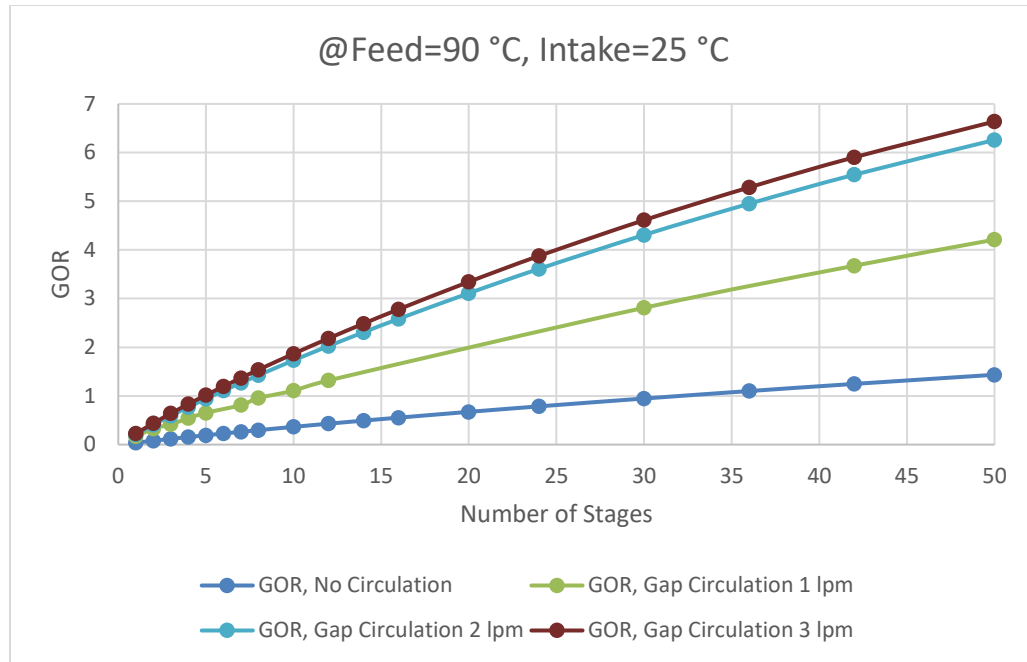
Figure 7.17 the effect of different numbers of stages on the STEC for 1 lpm (a) and 2 lpm (b) feed intake rate at different gap circulation rate for the new chambers design

As shown in in Figure 7.18, the GOR value presented a linear increase as the number of stages increases. This can be explained by the consistent heat recovery and transfer efficiency between each stage. When the heat recovery efficiency between stages remains consistent, the increase in the number of stages results in a proportional increase in the heat energy utilized for distilled water production. Since GOR is the ratio of the heat energy utilized for distilled water production to the heat energy input required for the process, a proportional increase in the heat energy utilized for distilled water production will result in a linear increase in the GOR values. The linear increase in GOR values signifies that the system consistently and effectively recovers and utilizes waste heat as more stages are added, contributing to improved overall energy efficiency in the desalination process.

A maximum GOR value of 6.3 is observed at a feed injection rate of 2 l/min and 3 l/min gap circulation rate compared to a maximum value of 2.26 in the old chamber design configuration for the same conditions. The effect of gap circulation is obviously observed on the GOR values for different conditions. Indeed, for this design, a 200 % and 371% increase in the GOR values is observed for 1 l/min, and 2 l/min feed intake rate, respectively, at 50 distillation stages when a gap circulation rate of 3 l/min. However, increasing the gap circulation from 2 to 3 l/min is insignificant because the more velocity in the turbulent characteristic is not much effective.



(a)



(b)

Figure 7.18 the effect of different numbers of stages on the GOR for 1 lpm (a) and 2 lpm (b) feed intake rate at different gap circulation rate for the new chambers design

The system also showed a rapid increase in the heat recovery ratio, 79, when increasing the number of stages to 20, achieving a value of more than 80 % when using a gap circulator of 2 or 3 l/min for both intake conditions. However, increasing the number of stages beyond that limit to 50 stages would present a margin increase of 10 % to achieve a value of approximate 90% meaning that the other 30 stages are not effective. This is because of the decrease in the average flux as shown in 80.

Analytically, for only 20 stages using a circulating pump of 3 l/min showed 110% increase in the heat recovery ratio because of the 50% increase in the flux compared to the stagnant water gap at a feed intake of 2 l/min.

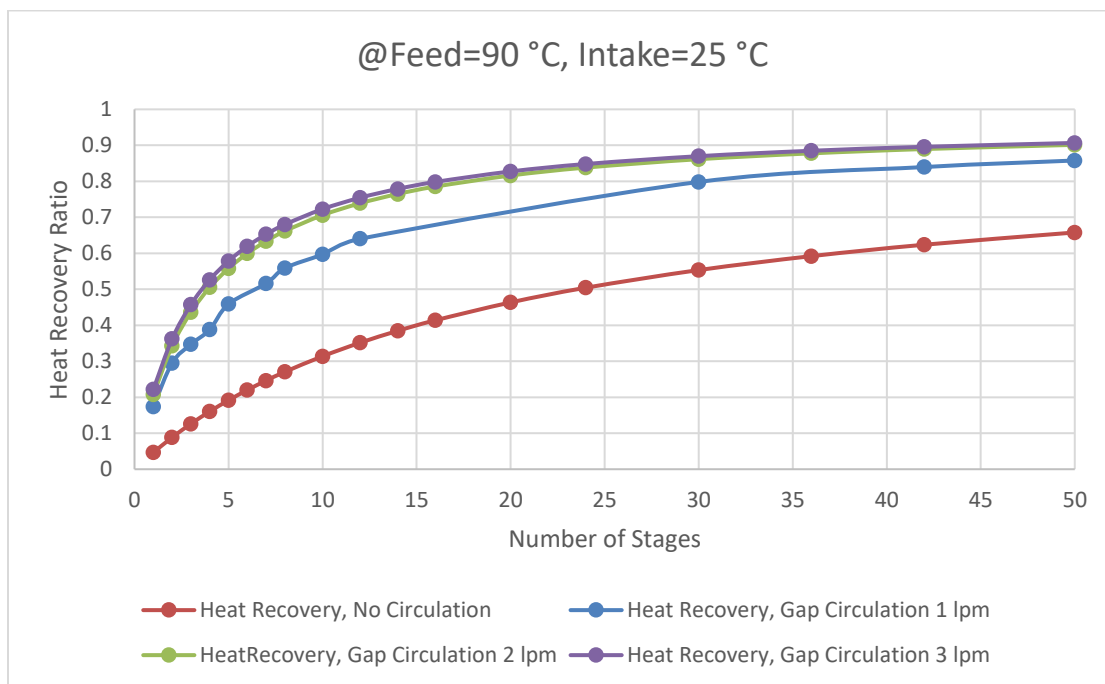
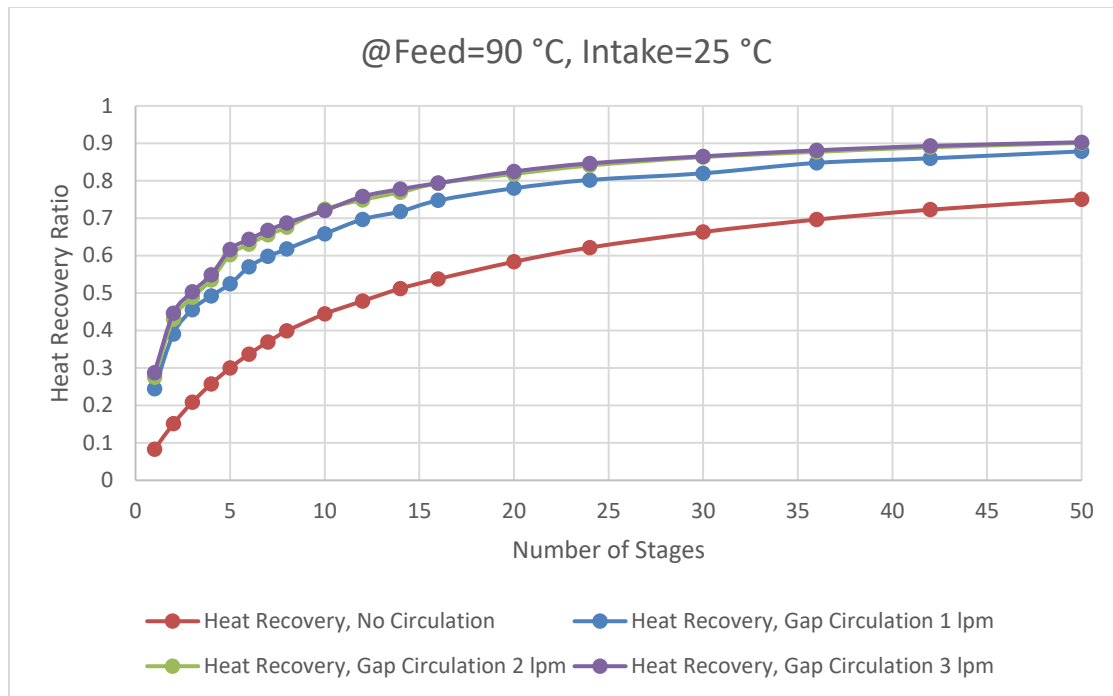
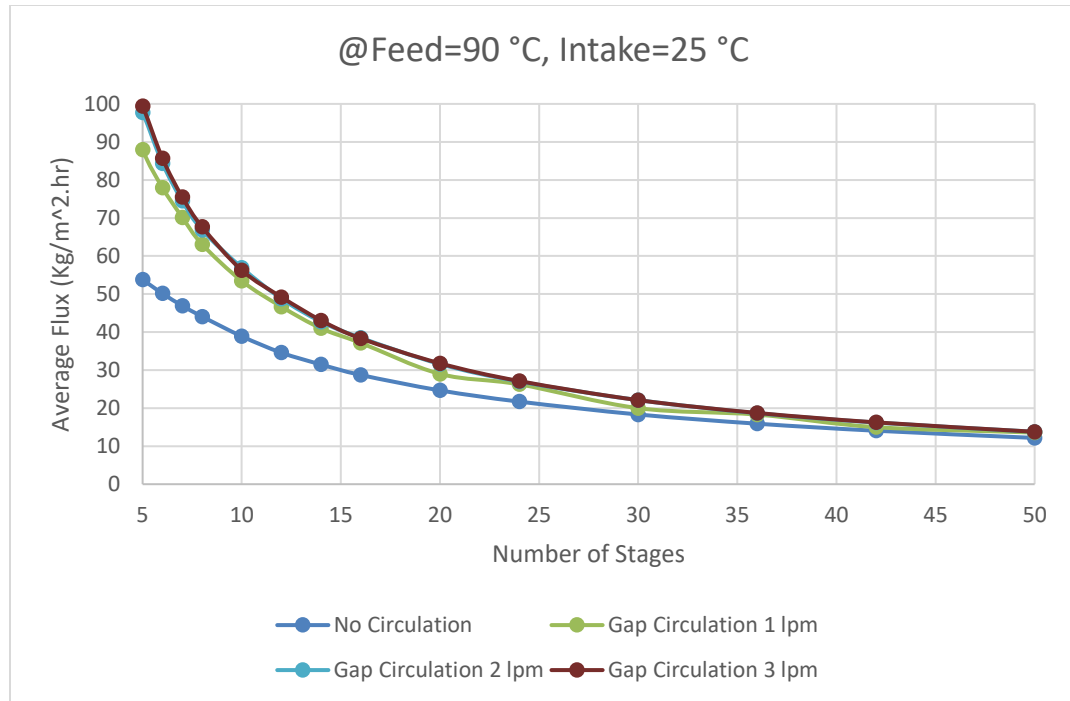
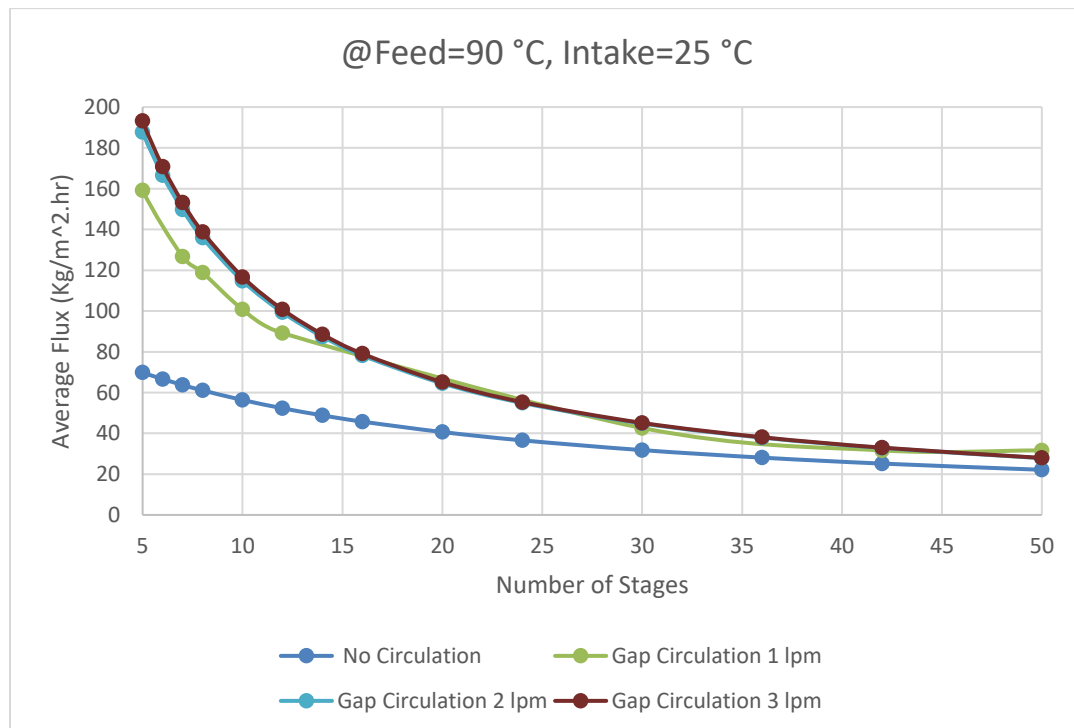


Figure 7.19 the effect of different numbers of stages on the Heat Recovery Ratio for 1 lpm (a) and 2 lpm (b) feed intake rate at different gap circulation rate for the new chambers design



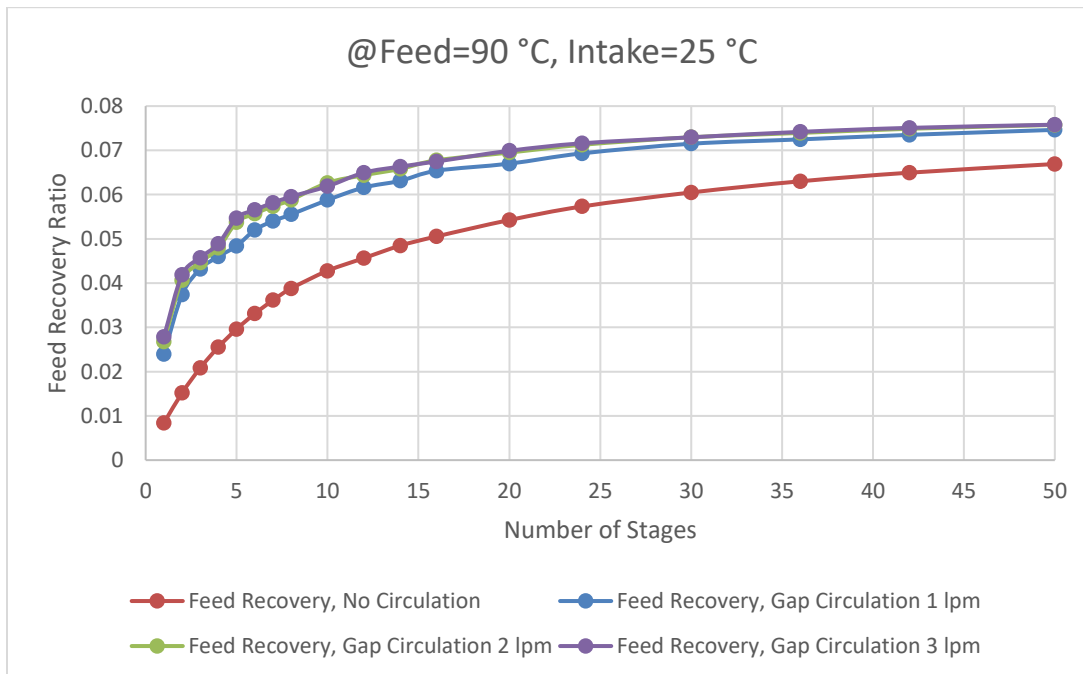
(a)



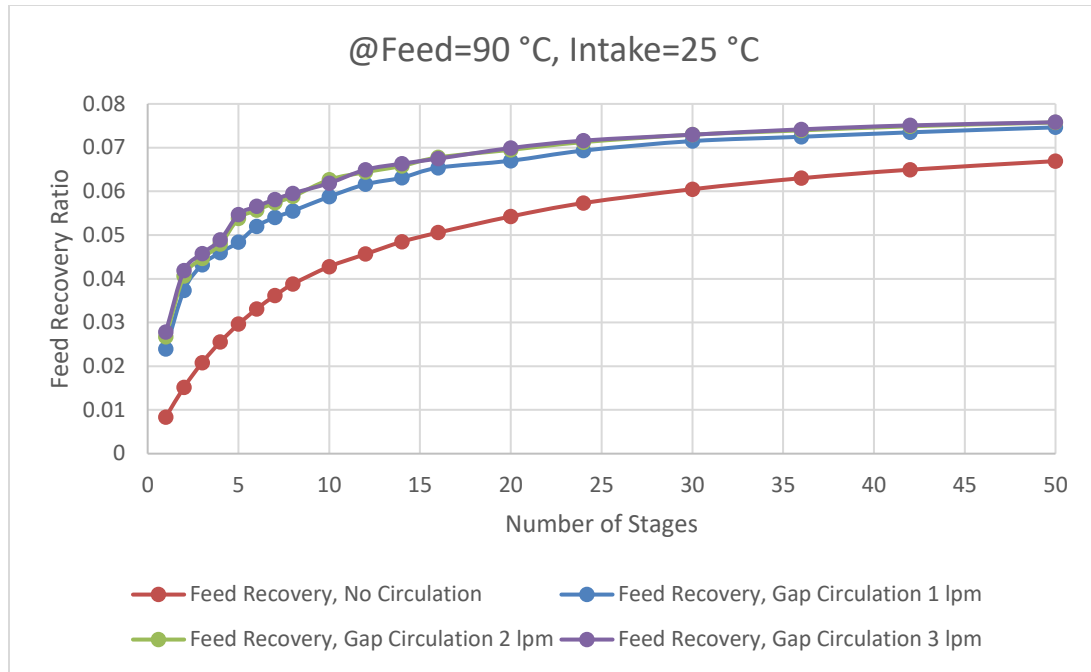
(b)

Figure 7.20 the effect of different numbers of stages on the average flux for 1 lpm (a) and 2 lpm (b) feed intake rate at different gap circulation rate for the new chambers design

The increased mixing performance resulting in the enhanced freshwater transfer resulted in a higher feed recovery for the new proposed design. For a feed intake flow rate of 2 l/min, 20 stages of the same design increased the feed recovery ratio from 5.5% to 7 % when using a 3 l/min permeate circulating pumps compared to stagnant water pump. On the other hand, for the same condition of feed intake flow rate of 2 l/min, 3 l/min permeate circulating pumps the feed recovery ratio was only 5.8 % in the old design.



(a)



(b)

Figure 7.21 the effect of different numbers of stages on the feed recovery ratio for 1 lpm (a) and 2 lpm (b) feed intake rate at different gap circulation rate for the new chambers design

7.2.2. Increasing the Area of PGMD

Another proposed alternative design is to increase the area of the PGMD without permeate gap circulation. The intake water temperature, which is used for cooling and pre-heated is considered constant in the whole analysis at 25 °C. The analyses are being investigated at different intake flow rates of 1, 2, 3, and 4 liters per minute (lpm) for the membrane type and geometric composition for the analyses presented in chapter 4. However, the only difference is that each membrane stage is scaled up 4 times in terms of area for all module components.

Figure 7.22 (a), (b), and (c) presents the energy performance, STEC & GOR, of the counter-flow heat recovery multi-stage water gap membrane distillation system for hot feed injection temperatures of 90, 80, and 70 °C, respectively.

From the figures, it can be observed that as the number of stages increases from 1 to 24, the GOR increases, indicating a higher energy efficiency in the system, for the conditions presented in the Figure. Simultaneously, the STEC decreases, meaning that less energy is consumed to produce one cubic meter of water product as the number of stages increases. The analyses of Figure 7.22, (a, b, & c) proves that, for different conditions, the higher feed injection temperature provides the highest MD efficiency with a highest GOR value of 3.1 at a feed injection temperature of 90 °C and 24 stages. This GOR value corresponds to STEC of 197 kWh/m³. The feed injection temperature seems to be an effective parameter for the heat recovery of an MD system. For a constant feed injection rate of 2 lpm increasing the feed injection temperature from 70 °C to 80 & 90 °C the GOR value increases by 13.5% & 27.5 % respectively for using 24 recovery stages. The increase in GOR with increasing feed injection temperature can be attributed to the higher temperature difference between the hot and cold streams, leading to a higher driving force for water vapor transport across the membrane, resulting in higher distillate production and energy efficiency. However, it is worth noting that increasing the intake flow rate beyond a certain limit may lead to decreased energy efficiency due to the increased energy required to raise the feed temperature at higher flow rates without a significant increase in flux. As an evidence, increasing the intake flow rate from 1 to 2 lpm results in a higher GOR for different feed injection temperature, indicating enhanced energy efficiency. Whereas, when the intake flow rate is increased to 3 and 4 lpm, lower energy performance is observed, as evidenced by a lower GOR and a higher STEC. This observation is the same for different feed injection rate. This can be attributed to the increased energy required to raise the feed temperature at higher flow rates (mass rates) without a significant increase in flux when

exceeding the turbulent feed limit, as would be explained in the detailed stage analyses in Figure 7.22.

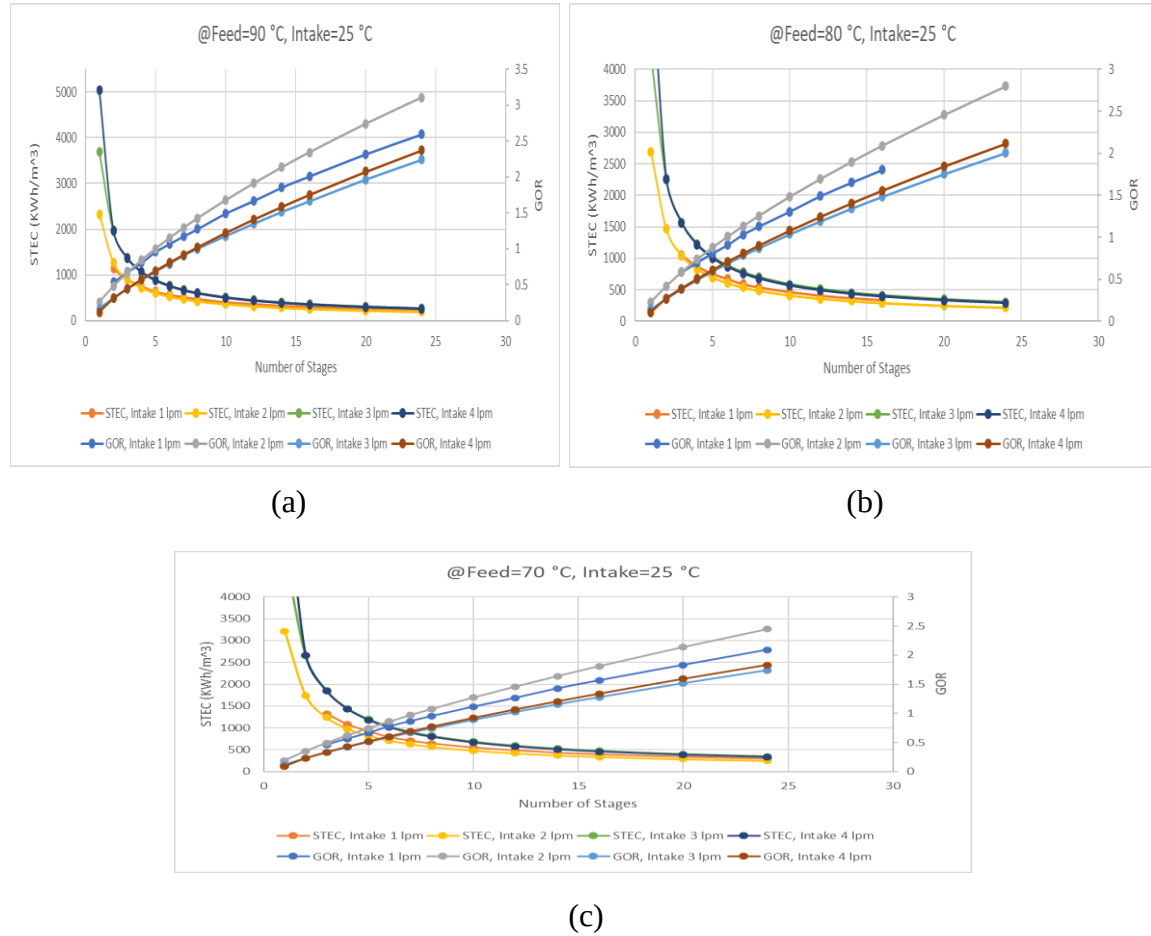


Figure 7.22 the energy performance, STEC & GOR, of the counter-flow heat recovery multi-stage stagnant water gap membrane distillation system for hot feed injection temperatures of 90 °C (a), 80 °C (b), and 70 °C (c), respectively.

In the counter flow CFPGMD some of the heat is also recovered in the coolant stream, so that a GOR value of greater than one and STEC of less than 600 kWh/m³ are observed. The main reason is that a heat is recovered efficiently by coolant water in this configuration. The equation that defines heat recovery ratio for the CFPGMD system (Equation 101) indicates that the amount of heat recovered in the system is inversely proportional to the amount of heat that needs to be added to raise the intake water temperature to the feed

injection temperature. As the number of stages in the CFPGMD system increases, the temperature difference between the hot and cold streams across the heating element becomes smaller due to the counterflow configuration, meaning that less input energy is required to raise the intake water temperature to the feed injection temperature.

To justify, the analyses of the heat recovery ratio have been performed and presented in Figure 7.23. The results claim that more energy is recovered at lower flow rates for different stage number or feed injection temperature. The main reason is that intake water serves as a coolant for product condensation, and its temperature increases gradually after each stage as it passes through each coolant channel. At lower flow rates, the intake water spends more time in each coolant channel, allowing it to absorb more heat from the product condensation process so that it achieves higher temperature before it is introduced to the heating element. On the other hand, at higher flow rates, the intake water spends less time in each coolant channel, which limits its ability to absorb heat and increase in temperature. As a result, it enters the heating element at a lower temperature, which increases the temperature difference between the hot and cold streams and requires more input energy to raise the temperature of the cold stream to the desired feed injection temperature. This leads to a lower heat recovery ratio and a lower energy efficiency in the system, as less of the heat generated during the membrane distillation process is recovered.

Therefore, to maximize energy recovery and efficiency in the CFPGMD system, it is important to optimize the intake flow rate to achieve the right balance between sufficient cooling and heat recovery, and minimizing the energy required to raise the temperature of the cold stream to the desired feed injection temperature.

To summarize, as more stages are added to the CFPGMD system, the hot stream is used more efficiently to transfer its heat to the cold stream, resulting in less energy needed to raise the temperature of the cold stream to the desired feed injection feed temperature. This translates to a higher heat recovery ratio and a higher energy efficiency in the system, as indicated by the higher GOR values observed in Figure 7.22 (a), (b), and (c) for 24 stages compared to 1 stage.

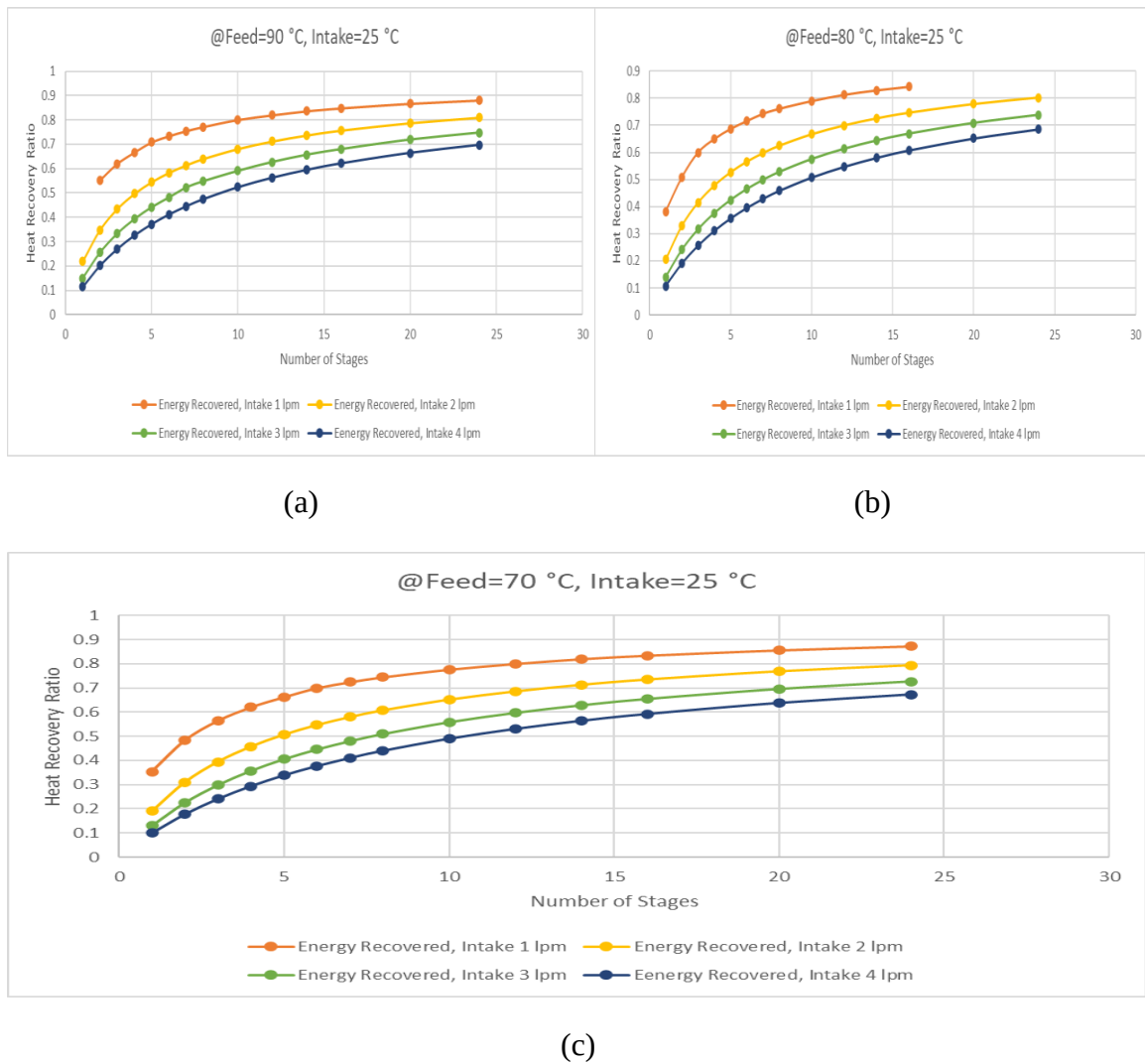
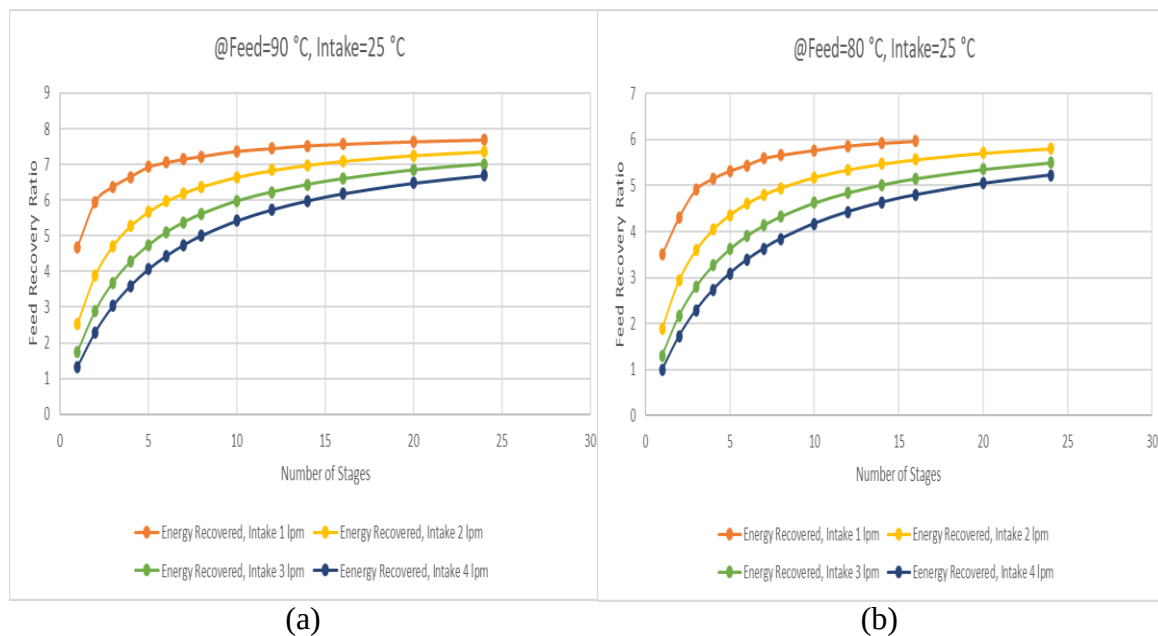
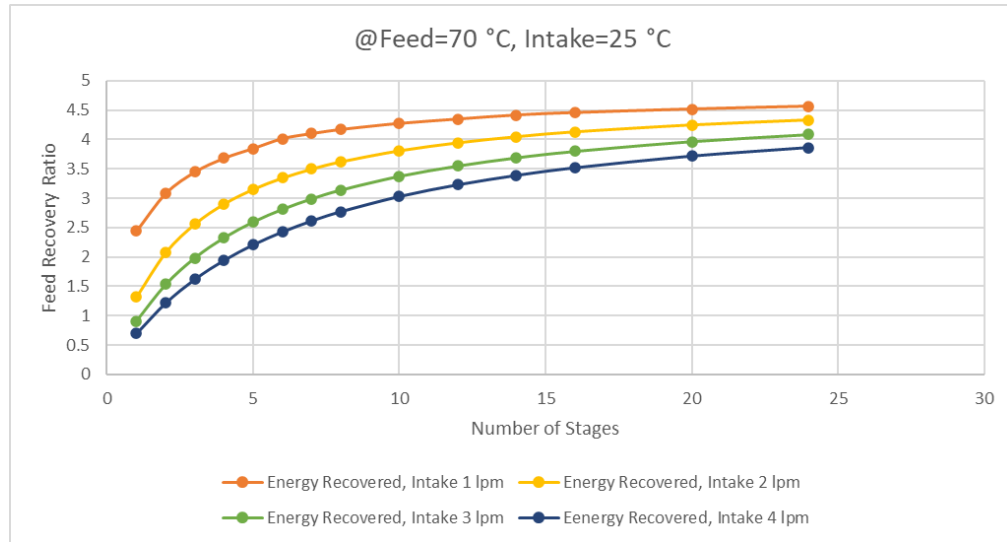


Figure 7.23 The heat recovery ratio for different feed intake rate for injection temperatures of 90 °C (a), 80 °C (b), 70 °C (c)

The analyses of Figure 17.24 represent the feed recovery ratio is a measure of the rate of freshwater production to the rate of the intake water. The condition of Figure 7.24 (a), (b), and (c) are like those in Figure 7.21. The results show that as the number of stages in the CFPGMMD system increases, the feed recovery ratio also increases. The maximum feed recovery ratio was observed at an intake flow rate of 2 lpm for different intake temperatures, for the same reason stated before, that any increase in the flow beyond this value would not represent an increase in the flow proportionally. The results claim that increasing the number of stages presents a rapid increase in the feed recovery ratio for a number of stages of 15 for different conditions then only a marginal increase beyond this value. To validate these observations a detailed analysis for each stage separately is performed in. Numerically, a maximum recovery ratio of 7.75%, 6%, and 4.6 % were observed for feed injection temperatures of 90 °C, 80 °C, and 70 °C respectively.





(c)

Figure 7.24 The feed recovery ratio for different feed intake rate for injection temperatures of 90 °C (a), 80 °C (b), 70 °C (c)

To validate the previous observations, separate analyses has been performed for each stage separately in terms of flux and the temperature difference between the feed inlet and outlet temperature per stage, the results are being presented in Figures 7.25 and 7.26 respectively for a feed injection temperature of 90 °C. The results of Figure 7.25 a, b, c, and d respectively represents flux per stage for 1, 8, 16, and 24 stages, respectively. It can be noticed that the flux per stage decreases as the number of stages increases for all flow rates. This is because the intake water temperature gradually increases as it passes through each stage, reducing the temperature difference between the feed and the coolant channels, and thus reducing the driving force for water vapor transport across the membrane. However, the flux reduction is more significant at lower flow rates. For example, at a flow rate of 1 lpm, the reduction in average flux per stage is about 93% (from 96.98 to 0.93) as the number of stages increases from 1 to 24.

On the other hand, at a flow rate of 4 lpm, which is highly turbulent, the average flux per stage drops from 110.5 to 23.1, a reduction of about 79 % as the number of stages increases from 1 to 24.

To understand more the reason to obtain an optimum flow rate of 2 lpm, the increase in average flux per stage is investigated, from Figure 7.25 compared to the increase amount of energy required to provide the heat temperature. According the results in Figure 7.25 increasing the feed injection flow rate from 1 to 2 lpm resulted in an average increase in the average flux by 77%, 87.5%, and 91 % for 8, 16, and 24 stages, respectively. Whereas the increase in energy is computed to be only 60%, 59%, and 60.5% which is less than the increase in the flux for 8, 16, and 24 stages, respectively. On the other hand, increasing the feed injection rate from 2 to 4 lpm the increase in the flux is computed to be 57 %, 74%, 81% with a higher percentage increase in the energy input to the system with 117%, 131%, 135% for 8, 16, and 24 stages, respectively. The increase of the energy at higher intake flow rates is because of the increase in the mass flow rate and the decrease of temperature input to heating element. While 2 lpm is considered the lower limit of the turbulent flow, which provided better mixing than the laminar 1 lpm. Increasing the flow while achieving the turbulent flow would nit be that beneficial on flux improvements.

The results also show that the highest flux per stage is achieved for the first stage, which is the stage closest to the feed inlet. This is because the temperature difference between the feed and the coolant channels is highest at the first stage, providing the highest driving force for water vapor transport across the membrane. As the intake water temperature increases in subsequent stages, the temperature difference decreases, leading to a reduction in the driving force and the flux per stage.

In summary, the results of the separate analyses for each stage confirm the observations from the previous figures and support the conclusion that the system can achieve higher energy efficiency and flux per stage at lower flow rates. The findings also suggest that the first stage is the most critical stage for achieving high flux and that the system's design should optimize the heat recovery and temperature difference between the feed and the coolant channels at this stage.

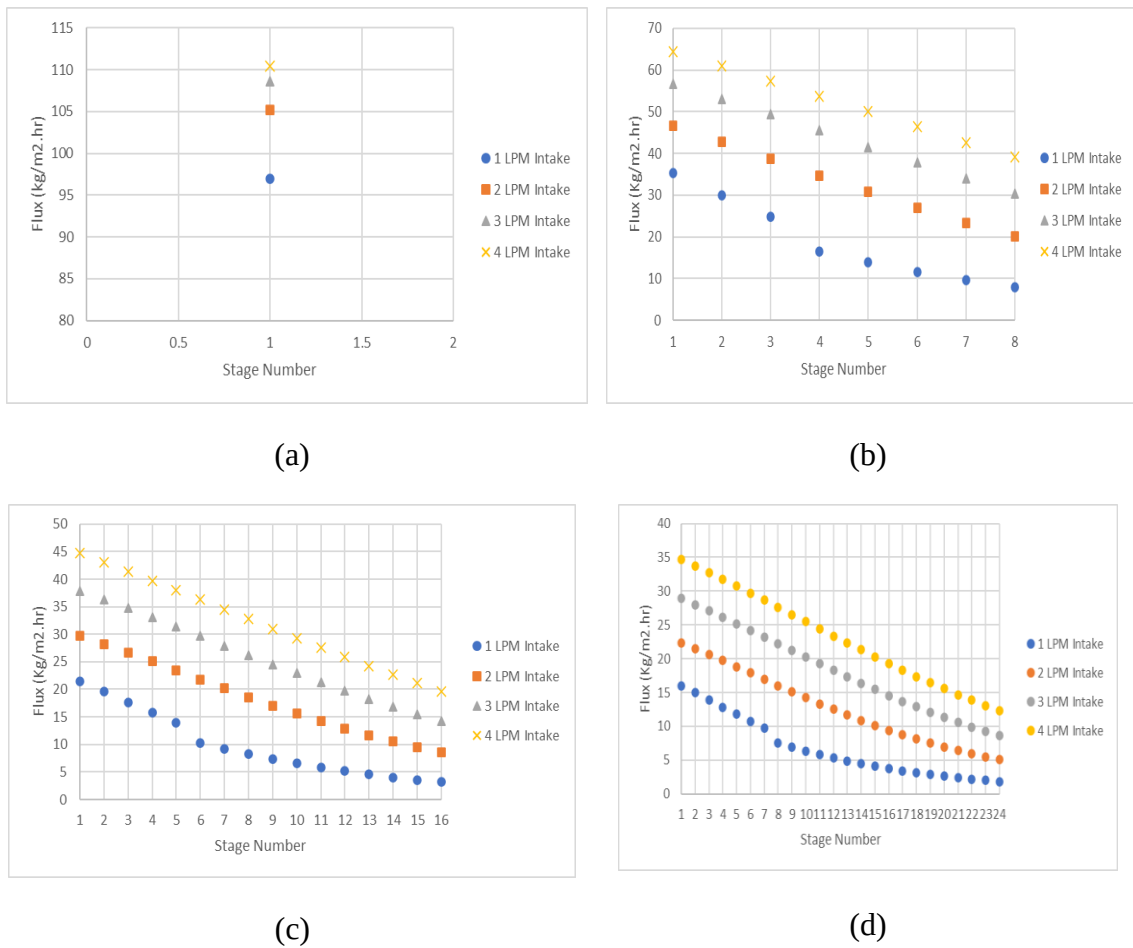
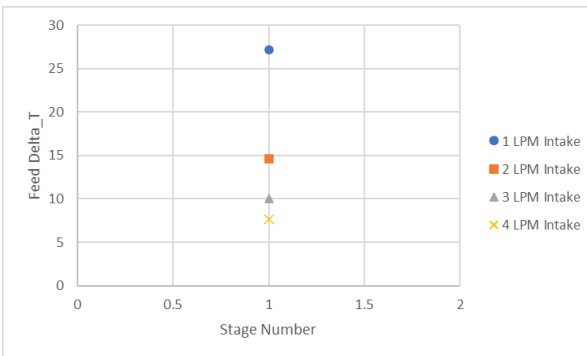


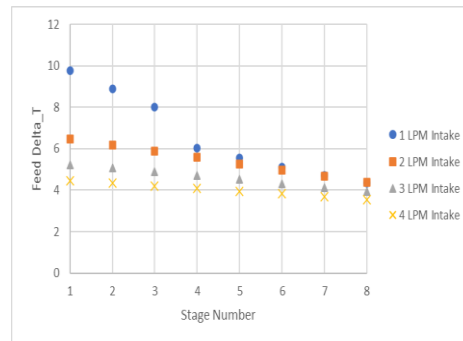
Figure 7.25 Detailed flux analyses for each stage at 1 (a), 8 (b), 16 (c), 24 (d) total number of stages

The analyses in Figure 7.26, are presented at the same conditions of Figure 7.25. They present the difference in the inlet and exit temperature of the feed temperature for each

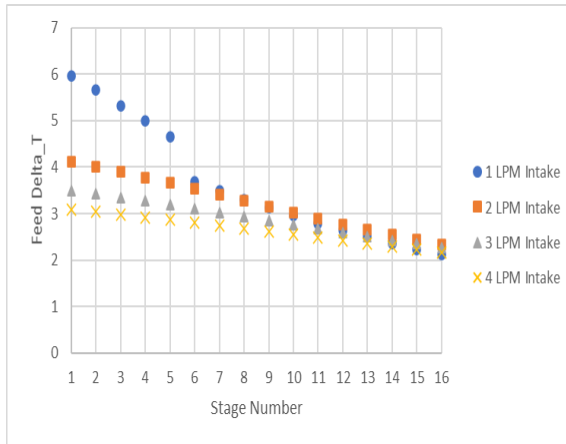
stage at 1, 2, 3, and 4 lpm for different stage numbers (parts a, b, c, and d respectively). It is shown that for different number of stages, the higher feed flow rate presents the lower temperature difference across each feed channel. This is due to the higher mass flow rate, which is not affected significantly when it losses heat by vapor transfer. It is also shown a decreasing temperature difference as the number of stages increases. This is because as the number of stages increases, the intake water temperature gradually increases after each stage, approaching the feed injection temperature. This reduces the temperature difference between the feed and the coolant channels, leading to a reduction in the driving force for water vapor transport across the membrane and a decrease in the flux per stage.



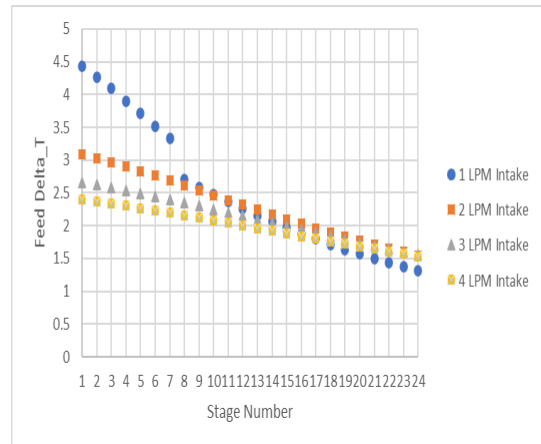
(a)



(b)



(c)



(d)

Figure 7.26 Detailed feed inlet and outlet temperature difference analyses for each stage at 1 (a), 8 (b), 16 (c), 24 (d) total number of stages

CHAPTER 8

COMPARATIVE STUDIES

In this chapter, the performance and efficiency of the proposed permeate gap membrane distillation (PGMD) system with gap water circulation and its integration with solar thermal Flat Plate Collector (FPC) will be thoroughly evaluated through a series of comparative studies. These studies have been designed in alignment with the research objectives, which encompass the development of detailed theoretical models, comprehensive performance assessments, optimization of design parameters and operating conditions, and the investigation of energy recovery strategies for the integrated MD desalination system.

Through these comparative studies, we aim to provide insights into the benefits and potential drawbacks of the proposed system and its configurations, the most cost-effective and efficient power sources for the PGMD system, and its performance relative to existing solar-MD technologies. This comprehensive evaluation will contribute to a better understanding of the system's capabilities, allowing for informed decision-making and optimization in future research and practical applications.

8.1. Comparison of the PGMD with/without Gap Water Circulation

The comparison of the PGMD with and without gap water circulation has been analyzed in detail in chapter 4 for an MD system without heat recovery, and in chapter 7 for a system with heat recovery. However, this section is to recap the main results to move through other comparison of the PGMD.

8.1.1. Without Heat Recovery

The WGMD system with circulated gap shows significant improvements in terms of permeate flux and energy performance compared to the conventional WGMD system without circulation. The analysis revealed better improvements for various operating parameters on the system's performance.

In terms of energy performance, the circulated gap system shows better energy efficiency (higher GOR and lower SEC) compared to the non-circulated gap system for circulation rates up to 4 L/min. However, at a circulation flow rate of 8 L/min, the energy performance is slightly reduced due to the increased energy loss between the cold and hot streams. Despite this reduction in energy efficiency, the significant enhancement in the production rate makes the circulated gap system a promising approach for improving the overall performance of water gap MD systems.

8.1.2. With Heat Recovery

The comparative study of the CFPMD system with and without permeate water gap circulation reveals significant differences in performance metrics. The analysis demonstrates that implementing gap circulation leads to improved energy efficiency,

higher GOR, and increased heat recovery, which ultimately results in a lower STEC across various intake flow rates and stage numbers.

The use of gap circulation reduces the STEC by up to 43% and increases the GOR by up to 80% compared to a stagnant water gap system. Additionally, the heat recovery ratio is higher with gap circulation, achieving up to 85%. The improvement in performance metrics is more pronounced at lower intake flow rates, as higher intake flow rates lead to reduced energy efficiency and lower GOR values.

The results also indicate that increasing the number of stages can enhance the feed recovery ratio, with the highest value of 7% achieved at an intake flow rate of 1 l/min. However, beyond 15 stages, the increase in feed recovery ratio is marginal. Implementing gap circulation further improves the feed recovery ratio, with an average increase of 7% observed across different intake flow rates.

This study highlights the importance of permeate gap water circulation in enhancing the performance of the CFPGMD system. The benefits of gap circulation are evident in the improved energy efficiency, higher GOR, increased heat recovery, and higher feed recovery ratio. These results suggest that incorporating gap circulation in the CFPGMD system can lead to a more energy-efficient and cost-effective desalination process, especially when optimizing the design parameters and operating conditions.

8.2. Comparison of the Solar Powered multi-stage CPGMD with other solar-MD Configurations

The novelty of this work lies in the development PGMD-GC combined with a flat plate collector (FPC), which demonstrates superior performance and cost-effectiveness

compared to other solar membrane distillation (MD) systems in the literature, as shown in Table 8.1. By harnessing solar energy to produce clean water, this system can provide a sustainable and cost-effective alternative for water desalination and purification, reducing reliance on fossil fuels and lowering the environmental impact of water treatment processes.

Table 8.1 Comparison of the proposed Solar System with other work in the literature

Reference	MD/Solar Systems	Best System Production	Minimum Water Production Cost Per Liter
Current Work	CPGMD-GC with FPC	6 Liters/Day/m ² of Collection Area	US\$ 0.016
[131]	SGMD with ESTC	4.15 Liters/Day/m ² of Collection Area	US\$ 0.0183
[132]	SGMD with FPC	4.15 Liters/Day/m ² of Collection Area	US\$ 0.85
[178]	PGMD with FPC	15.92 Liters/Day/m ² of Collection Area (Summer)	Euros 0.03196
[179]	DCMD with Parabolic Trough Collector	1.14 Liters/Day/m ² of Collection Area	US\$ 0.021
[180]	AGMD Spiral Wound with and Cooled concentrator photovoltaic	NA	US\$ 0.024

With a production rate of 6 Liters/Day/m² of collection area and a water production cost of \$0.016 per liter, the proposed system outperforms alternatives such as SGMD, DCMD, and AGMD configurations combined with different solar collectors. The improved performance and cost-effectiveness of the CPGMD-GC with FPC system could be attributed to the efficient heat transfer properties of the FPC, optimized design of the CPGMD-GC system, and better utilization of solar energy. This makes the proposed system a promising candidate for sustainable water production using solar energy, with potential for further optimization through additional research.

8.3. Comparison of the Optimum Water Cost of the Circulating PGMD under different Power Conditions (Solar, Electrical, and Waste Heat)

In this section, we present a comparison of the optimum water production cost for the circulating PGMD system under three different power conditions: electrical power, solar power, and waste heat. The results, as summarized in Figure 8.1 below, and they are extracted from different analyses from this study under similar membrane design and for the optimized parameters.

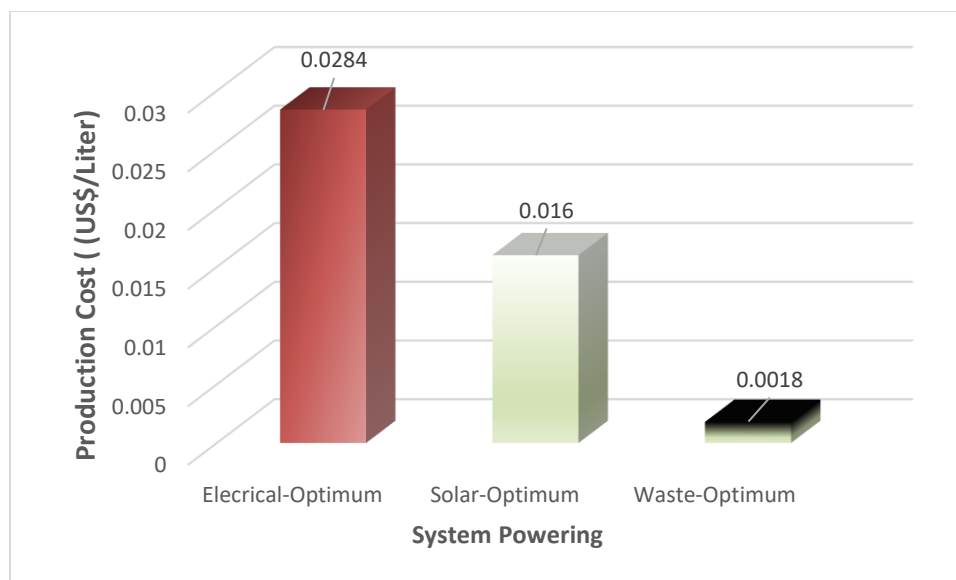


Figure 8.1 Optimum Cost of the Circulating multi-stage under different powering conditions (Electrical, Solar, and Waste Heat)

It is evident that utilizing waste heat as a power source yields the lowest water production cost, at US\$0.0018 per liter. This can be attributed to the fact that waste heat is an abundant and readily available energy source, which significantly reduces the operational costs associated with the PGMD system. On the other hand, the solar-powered configuration offers an optimum water production cost of US\$0.016 per liter, which is considerably lower than the cost associated with the electrical power source (US\$0.0284 per liter). The reduced production cost in the solar-powered configuration can be attributed to the utilization of renewable solar energy, which decreases the dependency on conventional electricity sources and thereby lowers the operational expenses.

Although both solar and waste heat power sources offer significant reductions in water production costs when compared to conventional electrical power, it is important to emphasize why solar power is more expensive than waste heat. The primary reason behind the higher cost associated with solar power is the initial capital investment required for the installation of solar panels or solar collectors, which are needed to harness and convert

solar energy into usable thermal or electrical energy for the desalination process. The cost of solar panels, mounting systems, and other auxiliary components can be substantial, increasing the overall cost of the solar-powered PGMD system. Additionally, the efficiency of solar energy conversion is inherently dependent on various factors such as solar irradiance, geographic location, and weather conditions. These factors can lead to fluctuations in the amount of solar energy available to power the system, which might require additional energy storage solutions or backup power sources to ensure uninterrupted operation. These supplementary systems can further increase the overall cost of the solar-powered desalination system.

In contrast, waste heat is a byproduct of various industrial processes and is readily available at little or no cost. Since waste heat is often considered a "free" energy source, the operational costs associated with harnessing and utilizing it are considerably lower than those for solar power. Moreover, waste heat is typically more consistent and reliable in terms of availability, which reduces the need for additional energy storage or backup power solutions.

In conclusion, while both solar and waste heat offer significant advantages over conventional electrical power sources, the higher initial capital investment and the variable nature of solar energy contribute to the higher costs associated with solar-powered PGMD systems compared to those powered by waste heat.

8.4. Comparison of the Heat Recovery Circulating PGMD with Data from the Literature

This section presents a comparative analysis of the proposed CFPGMD system with gap water circulation against various membrane distillation configurations reported in the

literature for the purpose of heat recovery. Table 8.2 displays the best system flux, energy metrics, and recovery ratio (RR) obtained from different studies.

Table 8.2 Comparison of the proposed CFPGMMD with gap water circulation System with other work in the literature for the purpose of heat recovery

Reference	MD Configuration	Best System Flux (Kg/m².hr)	Best Energy Metrics	Best RR (%)
Current Work	CFPGMD with Gap Water Circulation	28	GOR = 6.6 STEC = 92 kWh/m ³	7.7
[133]	DCMD with Hollow Fiber Membranes	NA	1.15 kg/MJ (241 kWh/m ³)	6.5
[134]	DCMD with external Heat Recovery	6	GOR = 4.5	NA
[135]	Multi-stage VMD	25	STEC = 240 kWh/m ³	NA
[136]	AGMD	1	GOR = 0.9	NA
[137]	AGMD with Hollow Fiber Membranes	2	13.8	NA
[138]	AGMD	4	STEC = 294 kWh/m ³	2.25
[139]	PGMD	6.5	GOR = 2.5	NA

The CFPGMD system with gap water circulation offers several advantages compared to other membrane distillation configurations, as demonstrated by its superior performance in terms of energy efficiency and heat recovery. Some of these advantages include:

1. Higher flux rate: The CFPGMD system achieves a high flux rate of $28 \text{ kg/m}^2\cdot\text{hr}$, which is significantly higher than most other configurations. A higher flux rate implies that less membrane area is required to achieve the same level of water production. This reduces the overall size and cost of the membrane modules and contributes to a more compact and economical system design.
2. Lower STEC: The system's Specific Thermal Energy Consumption (STEC) value of 92 kWh/m is considerably lower than those reported for other configurations. A lower STEC means that the system consumes less energy per unit of water produced, resulting in reduced energy costs and lower environmental impact.
3. Higher GOR: With a Gained Output Ratio (GOR) of 6.6, the CFPGMD system demonstrates superior energy efficiency. A higher GOR value indicates that the system effectively recovers and utilizes waste heat, reducing the need for external heat input and minimizing energy consumption.
4. Competitive recovery ratio (RR): The recovery ratio of 7.7% is competitive with other published works, indicating efficient water recovery and reduced waste in the form of brine. A higher RR means that more freshwater is produced from the same volume of feed water, contributing to the overall efficiency and effectiveness of the desalination process.
5. Effective heat recovery mechanism: The combination of counterflow and multi-stage design enables efficient heat transfer between stages, reducing the amount of

external heat input required. This results in a more energy-efficient system and contributes to the superior GOR and lower STEC values.

In contrast, other configurations in the literature, such as Direct Contact Membrane Distillation (DCMD), Multi-stage VMD, and Air Gap Membrane Distillation (AGMD), exhibit lower GOR values and higher STEC values, suggesting less efficient heat recovery and utilization. The combination of the counterflow and multi-stage design in the CFPGMD system offers a notable advantage over these configurations, resulting in superior energy efficiency and performance. On the other hand, the CFPGMD system with gap water circulation outperforms other membrane distillation configurations reported in the literature in terms of energy efficiency and heat recovery. Its unique design and effective heat recovery mechanism contribute to its enhanced performance, making it a promising technology for water desalination and purification applications.

8.5. Case Study of Multi-Stage PGMD with Gap Water Circulation with/without Heat Recovery

This section presents a case study that compares the performance of a multi-stage PGMD system with gap water circulation, both with and without heat recovery. The aim is to assess the impact of heat recovery on various performance metrics and operating conditions. The comparison is made between two designs: Multi-Stage PGMD-CG without heat recovery and CFPGMD with gap water circulation for the purpose of heat recovery. Both designs utilize 50 stages with 6 cm channel width, and the operating conditions for each are outlined in Table 8.3. For consistency, as shown in the table, both systems share similar operating conditions, including feed flow rate, temperature, coolant flow rate, and temperature. The

gap circulation rate is also consistent at 3 l/min per stage. However, there are notable differences in the performance metrics.

Table 8.3 Comparison table for different metrics performance and operating conditions comparing the multi-stage design with/without heat recovery

	CFPGMD with Gap Water Circulation for the Purpose of Heat Recovery	Multi-Stage PGMD-CG without Heat Recovery
Number of Stages	50 stages with 6 cm channel width	50 stages with 6 cm channel width
Feed Flow Rate & Temperature	1 l/min (Same Flow across all stages) at 90 °C	1 l/min/stage (20 l/min overall) at 90 °C
Coolant Flow Rate & Temperature	1 l/min (Same Flow across all stages) at 25 °C	1 l/min/stage (20 l/min overall)
Gap Circulation Rate	3 l/min/stage	3 l/min/stage
Average Flux (Kg/m².hr)	12.862	121
Daily Production (l/day)	102	960 l/day
Feed Recovery Ratio (%)	7.07	1.31

STEC (kWh/m³) & Corresponding GOR	161 kWh/m ³ GOR = 3.7	600 kWh/m ³ GOR = 0.95
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The CFPGMD system with heat recovery has the same feed flow rate across all stages, while the Multi-Stage PGMD-CG without heat recovery has a feed flow rate of 1 l/min per stage, resulting in a higher overall flow rate of 20 l/min. This difference in feed flow rate may be a contributing to obtain a feed recovery ratio is lower in the Multi-Stage PGMD-CG without heat recovery system, at 1.31% compared to 7.07% in the system with heat recovery. This suggests that a smaller proportion of the feed water is converted to distilled water in the system without heat recovery.

The average flux in the Multi-Stage PGMD-CG without heat recovery system is significantly higher at 121 Kg/m².hr compared to 12.862 Kg/m².hr in the system with heat recovery. This substantial increase in flux translates to a higher daily production of 960 liters per day in the system without heat recovery, as opposed to only 102 liters per day in the system with heat recovery. The reason behind this observation is that the system without heat recovery works under a higher temperature difference between the feed and coolant channel (90 -25 °C), while the system of with a heat recovery works under an average of difference of just 15 °C

In terms of energy efficiency, the STEC in the Multi-Stage PGMD-CG without heat recovery system is higher, at 600 kWh/m³, while the corresponding GOR is lower at 0.95. Conversely, the system with heat recovery has a lower STEC of 161 kWh/m³ and a higher GOR of 3.7. This indicates that the system with heat recovery requires less thermal energy

input per unit of water produced and achieves better utilization of heat energy for distilled water production.

In conclusion, the case study demonstrates that the absence of heat recovery in the multi-stage PGMD system with gap water circulation can lead to a significant increase in flux and daily production, meaning that smaller area is needed to obtain the same amount of product water. However, the energy efficiency of the system is compromised, as evident by the higher STEC and lower GOR values. Therefore, it is essential to carefully consider the trade-offs when designing and optimizing a multi-stage PGMD system with gap water circulation and heat recovery.

CHAPTER 9

CONCLUSIONS & RECOMMENDATIONS

In conclusion, the present study provides a comprehensive analysis of water desalination methods, with a focus on enhancing productivity and energy efficiency through the use of circulating water gap membrane distillation (MD) systems. By conducting a thorough theoretical investigation and validating the model against experimental data, in the study establishes the considerable potential of circulating water gap MD systems in improving output flux while maintaining energy efficiency.

One of the key findings of this study is the significant enhancement of output flux achieved by circulating the water in the gap. This improvement results from increased mixing in the distillate channel, which ensures a uniform temperature distribution within the gap. The circulation of gap water also enhances heat and mass transfer coefficients, leading to increased condensation rates and, consequently, higher freshwater production. While this improvement comes at the expense of increased energy consumption, the overall energy efficiency still experiences a slight increase, making circulating water gap MD systems a promising option for desalination.

In addition to investigating the performance of circulating water gap MD systems, the study explores the potential of solar-powered circulating permeate gap membrane distillation (PGMD) systems in providing a sustainable, cost-effective source of clean water. This is particularly relevant for rural areas and eco-tourism resorts, where access to conventional energy sources may be limited. The analysis underscores the importance of carefully

selecting the permeate gap circulation rate and the number of flat plate collectors (FPCs) to achieve optimal system performance, energy efficiency, and water production cost reduction. The study also provides guidelines for designing and operating pilot-scale solar-MD units, which can help reduce reliance on fossil fuels and promote sustainable development.

Moreover, an artificial neural network (ANN) model is employed to predict and optimize freshwater production using a multi-stage circulating permeate gap membrane distillation system. The model demonstrates the feasibility of the proposed system and its ability to accurately predict freshwater production, with a mean absolute percentage error (MAPE) of 1% and a strong correlation with actual output. The optimization results reveal the optimal feed temperature (90 °C), the minimum number of stages (12), and the gap circulation rate per stage (5 l/min) required for efficient freshwater production of around 1000 L/day.

The cost analysis and optimization study of freshwater production using the proposed multi-stage MD system show that the selection of the energy source plays a crucial role in determining the cost of water production. In cases where waste heat is used as an energy source, increasing the number of MD stages significantly reduces the annualized capital cost, leading to a substantial decrease in water production cost. However, when an electrical energy source is used, the cost of energy dominates the total production cost, and the impact of the initial capital cost is minimal.

The potential of heat recovery in circulating water gap MD and solar-powered circulating PGMD systems should not be overlooked, as it plays a crucial role in enhancing the overall energy efficiency and cost-effectiveness of these desalination technologies. By

implementing effective heat recovery strategies, it is possible to minimize the energy input required for the process, thereby reducing the environmental impact, and lowering the water production cost. Furthermore, heat recovery can enable these systems to operate using waste heat or low-grade heat sources, promoting the integration of desalination technologies with other industrial processes or renewable energy systems. Therefore, future research and development efforts should focus on identifying and optimizing innovative heat recovery solutions, ultimately contributing to the advancement of sustainable and energy-efficient desalination technologies.

In summary, these findings contribute valuable insights and guidelines for designing and implementing advanced water desalination systems, ultimately promoting a sustainable and cost-effective approach to clean water production. By exploring the potential of circulating water gap MD and solar-powered circulating PGMD systems, this study demonstrates that innovative desalination technologies can play a crucial role in addressing global water scarcity challenges. Furthermore, the use of artificial intelligence techniques, such as ANN, for predicting and optimizing system performance highlights the potential for integrating advanced computational tools in the development of more efficient and sustainable water treatment solutions.

Based on the findings and conclusions of this study, the following recommendations are proposed to further advance the development and implementation of circulating water gap membrane distillation (MD) and solar-powered circulating permeate gap membrane distillation (PGMD) systems:

1. Pilot-scale implementation: To validate the performance of the proposed circulating water gap MD and solar-powered circulating PGMD systems under

real-world conditions, it is recommended to conduct pilot-scale implementations in rural areas. These pilot projects will provide valuable insights into the scalability, durability, and reliability of the proposed systems while addressing the water scarcity challenges in such locations.

2. Investigate alternative energy sources: Given the significant impact of the energy source on water production cost, exploring the feasibility of integrating alternative renewable energy sources, such as wind or geothermal energy, with circulating MD systems is recommended. This would help further reduce the environmental impact and enhance the sustainability of these desalination technologies.
3. Optimize system components: Further research should be conducted to optimize various system components, such as membrane materials, module design, and gap size, to enhance the performance and energy efficiency of circulating water gap MD and solar-powered circulating PGMD systems. This optimization process could involve investigating novel materials, conducting computational fluid dynamics (CFD) simulations, and exploring innovative module designs.
4. Hybrid systems: Investigate the potential of combining circulating water gap MD or solar-powered circulating PGMD systems with other desalination technologies, such as reverse osmosis or electrodialysis, to create hybrid systems that can achieve higher water recovery rates and overall system efficiency.
5. Advanced control strategies: Develop advanced control strategies that can dynamically adjust the operational parameters of circulating water gap MD and solar-powered circulating PGMD systems in response to varying environmental

and operational conditions. These control strategies could include model predictive control (MPC), adaptive control, or artificial intelligence-based techniques.

6. Long-term performance and fouling studies: Conduct long-term performance studies to evaluate the stability and durability of the circulating water gap MD and solar-powered circulating PGMD systems, particularly in the context of membrane fouling and scaling. Such studies will help in developing appropriate maintenance and cleaning strategies to ensure sustained performance and a long operational life.
7. Socioeconomic impact assessment: Assess the socioeconomic impact of implementing circulating water gap MD and solar-powered circulating PGMD systems in rural areas and eco-tourism resorts. This includes evaluating job creation potential, impact on local communities, and contribution to regional sustainability goals.

By following these recommendations, researchers, policymakers, and industry stakeholders can contribute to the ongoing development and implementation of circulating water gap MD and solar-powered circulating PGMD systems, ultimately promoting sustainable, cost-effective, and innovative solutions to global water scarcity challenges.

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