

University of Bahrain

College of Engineering

Department of
Chemical Engineering



Performance, Modeling and Optimization of Solar Desalination using NVD

Thesis submitted in partial fulfillment of the requirements for the degree of PhD
In Chemical Engineering

Prepared by:
Ajaz Rashid Ghulam
ID number: 19953880

Supervised by:
Dr. Abderrahim Abbas
(Professor)
Dr. Teoman Ayhan
(Professor)

University of Bahrain

Kingdom of Bahrain
February 2018

Defense committee's approval page

University of Bahrain

College of Engineering

Department of Chemical Engineering

PhD in Chemical Engineering

Student Ajaz Rashid Ghulam, ID number 19953880 defended his thesis by which he fulfilled the requirement for the degree of PhD in Chemical Engineering entitled “Performance, Modeling and Optimization of Solar Desalination using NVD” on this day of (/ / 2018).

The committee recommends awarding of PhD Degree with the grade of:

(Defense Committee)

Name

Signature

Prof. Abderrahim Abbas

Supervisor

Prof. Teoman Ayhan

Co-supervisor

Prof. Ibrahim Al-Mutaz

External examiner

Prof. Ahmed Youssef

Internal examiner

Academic department's stamp

Abstract

This study presents an experimental and theoretical investigation of a novel cone-shaped desalination system. The system used gravity and atmospheric pressure to create a vacuum condition wherein water can be evaporated at relatively lower temperatures. This lowered temperature allows the use of low-grade heat sources to be used for the desalination process. The uniqueness of this study is the special inverted cone-shape geometry that allows the evaporator and condenser to be under the same roof.

The experimental setup including the tanks, piping and instrumentation has been designed and built as a part of this study. The main NVD (Natural Vacuum Distillation) tank which embodies the evaporation chamber as well as the fresh water chamber is placed at a height of 10.3 m above the ground level to balance the barometric height of water. The evaporation chamber is connected to the saline water feed tank, the brine disposal tank and the fresh water storage tank. All the storage tanks are at the ground level.

In this research, the process was studied theoretically and experimentally to investigate the effects of various operating conditions. Experimental studies were carried out on batch and continuous modes. The main factor that affects the production rate is the saline water temperature. Freshwater production rates of 24.4 L/day for the batch process and 15.65 L/day for the continuous process were measured using a heating temperature of 70 °C.

A theoretical model was also developed using the COMSOL simulation software, to simulate the performance of the system, which agreed with the experimental results. Freshwater production amount was found to be in direct proportion to the evaporation temperature and inversely proportional to the freshwater side temperature. The freshwater production amount was also found to increase non-linearly with the evaporation surface area. The recovery ratio and the thermal efficiency both increased with the increase in the heating temperature for the continuous operation while the increase in freshwater side temperature caused the production rate, recovery ratio and the thermal efficiency to fall. The brine withdrawal rate had a big impact on the system performance. The thermal efficiency showed a peak at a withdrawal rate of 5 kg/h. The depth of water in the evaporation chamber also affected the production rate in a directly proportional manner in the investigated range of 25 – 40 cm.

Dedication

This thesis is lovingly dedicated to my father, who toiled hard to make this dream come true...

and to my mother, for whose services and sacrifices, no words will ever do justice.

Acknowledgements

I would like to express my deep sense of appreciation and thanks to my supervisor Prof. Dr. Abderrahim Abbas and my co-supervisor Prof. Dr. Teoman Ayhan for being such wonderful mentors for me. I would like humbly wish to thank them for their constant encouragement, support and guidance throughout the period of research. Their advice has always been priceless. I would also like to thank Mr. Juma Mohammad Abbas for his assistance in setting up and rendering valuable help in conducting some of the experimental procedures. Thanks are also due to all the other colleagues who have directly or indirectly helped me during the various stages of my PhD work.

I must mention special thanks to my family. Words cannot express how grateful I am to my mother and father for all of the sacrifices that they have made towards the fulfillment of my dream. Their prayers for me were what sustained me thus far. Last but not the least, I would like to express utmost appreciation to my beloved wife who spent sleepless nights with me and without her utmost support, day in and day out, it would not have been possible to achieve the Himalayan task of finishing the PhD work.

Ajaz Rashid

University of Bahrain

February 2018

Publications

Journals:

1. Rashid, A., Ayhan, T., Abbas, A., 2016, Natural Vacuum Distillation for seawater desalination – A review, *Desalination and Water Treatment*, 57 (56), pp 26943-26953.
2. Rashid, A., Abbas, A., Ayhan, T., Experimental investigation of a batch operation of a cone-type NVD desalination system (submitted).
3. Rashid, A., Abbas, A., Ayhan, T., Study of a novel NVD seawater desalination system using solar thermal energy (submitted).

Conferences:

1. Rashid, A., Ayhan, T., Abbas, A., Natural Vacuum Desalination – A Review, Middle East Process Engineering Conference, 29 September – 2 October 2013, Manama, Kingdom of Bahrain.
2. Rashid, A., Abbas, A., Ayhan, T., Design and Performance of a Parabolic Trough Collector, World Renewable Energy Congress, 4-7 December 2016, Manama, Kingdom of Bahrain.
3. Rashid, A., Abbas, A., Ayhan, T., Study of a novel NVD seawater desalination system using solar energy, World Renewable Energy Congress, 4-7 December 2016, Manama, Kingdom of Bahrain.
4. Rashid, A., Abbas, A., Ayhan, T., Steady-state analysis of a desalination based on the principle of Natural Vacuum Distillation, WSTA 12th Gulf Water Conference, 28-30 March 2017, Manama, Kingdom of Bahrain.

Table of Contents

Subject	Page No.
Defense Committee's Approval	i
Abstract	ii
Dedication	iii
Acknowledgements	iv
Publications	v
Table of Contents	vi
List of Tables	ix
List of Figures	x
Appendices	xv
Nomenclature	xvi
Chapter One Introduction	1
1.1 Water Desalination	2
1.2 Overview of Desalination Technologies	3
1.2.1 Reverse Osmosis (RO)	4
1.2.2 Multi-Stage Flash Distillation for desalination	6
1.2.3 Multiple Effect Distillation	7
1.2.4 Comparison of RO, MSF and MED	9
1.3 Renewable Energy in Water Desalination	9
1.4 Research Objectives	10
1.5 Research Significance	11
Chapter Two Literature Survey	12
2.1 Natural Vacuum Distillation	13
2.2 Single Stage NVD Systems	15
2.3 Multi-stage NVD Systems	29
2.4 Thermal Energy Storage	32
2.5 Benefits of NVD Systems	34
Chapter Three Theoretical Analysis	36
3.1 The Proposed NVD System	37
3.2 Mathematical Model	38
3.2.1 Lumped Parameter Model	38
3.2.2 Distributed Model	41
3.3 Modeling with COMSOL Multiphysics ®	46

3.3.1 Overview of COMSOL Multiphysics ®	46
3.3.2 NVD Modeling in COMSOL Multiphysics ®	49
3.4 Modeling of the Parabolic Solar Collector	54
3.4.1 Geometrical Aspects of a PTC	57
3.4.2 Thermal Analysis of a PTC	59
Chapter Four Experimental Study	64
4.1 Experimental Setup	65
4.1.1 Evaporation Chamber	68
4.1.2 Freshwater Chamber	69
4.1.3 Condenser Space	69
4.1.4 Piping and Storage Tanks	71
4.1.5 Instrumentation	73
4.2 Experimental Procedure	78
4.2.1 Batch Operation	79
4.2.2 Continuous Operation with Heating in Evaporation Chamber	81
4.2.3 Continuous Operation with Heating in Feed Water Storage Tank	82
4.3 Uncertainty Analysis	83
4.3.1 Uncertainty and Confidence Intervals	83
4.3.2 Propagation of Errors	86
4.3.3 Uncertainty in the Measured Values	88
Chapter Five Results and Discussion	92
5.1 Experimental Results	93
5.1.1 Batch Operation	93
5.1.1.1 Effect of Heating Temperature	99
5.1.1.2 Effect of Water Salinity	102
5.1.2 Continuous Operation with Heating in Evaporation Chamber	103
5.1.3 Continuous Operation with Heating in Feed Water Storage Tank	108
5.2 COMSOL Model	112
5.3 Model Validation	116
5.3.1 Batch Operation	116
5.3.2 Continuous Operation	118
5.3.3 Simulation Results	120
5.4 Parametric Studies using COMSOL	134
5.4.1 Batch Operation	134
5.4.1.1 Effect of Evaporation Temperature	135
5.4.1.2 Effect of Freshwater Side Temperature	136

5.4.1.1 Effect of Evaporator Size	137
5.4.2 Continuous Operation	137
5.4.2.1 Effect of Evaporation Temperature	138
5.4.2.2 Effect of Freshwater Side Temperature	139
5.4.2.3 Effect of Brine Withdrawal Rate	142
5.4.2.4 Effect of Water Height in Evaporation Chamber	144
5.5 NVD using Solar Parabolic Trough Collector	145
5.6 Cost Analysis	151
Chapter Six Conclusion	155
Conclusion	156
References	160
Appendices	176

List of Tables

No.	Title	Page No.
1.1	Comparison of RO, MSF and MED	9
2.1	Summary of the various NVD systems reported in literature	15
4.1	Accuracy, measurement and error for the measuring instruments used	83
5.1	Model-fitting parameters for different set temperatures	97
5.2	Values of the parameters used in the COMSOL model	112
5.3	Definitions of the variables used in the model	113
5.4	Parameters used in the PTC model	146

List of Figures

No.	Figure	Page No.
1.1	Allocation of the world's water resources	2
1.2	Global installed desalination capacity by technology for 2014	4
1.3	Illustration of the principle of reverse osmosis	5
1.4	Simple illustration of the RO process	5
1.5	Multi-stage flash process	7
1.6	Multi-effect distillation process	8
1.7	Possible combinations of renewable energy systems with desalination units	10
2.1	Illustration of the Torricelli vacuum created over a column of liquid mercury or water	14
2.2	Illustration of the NVD desalination system proposed by Parker (1982)	16
2.3	Simple representation of a desalination system using NVD principle	18
2.4	Schematic representation of desalination system using the NVD process by Al-Kharabsheh and Goswami (2003)	20
2.5	Schematic representation of desalination system using the NVD process by Gude and Nirmalakhandan (2008)	23
2.6	NVD based humidification de-humidification desalination process by Eames et al. (2007)	26
2.7	Schematic representation of NVD system adapted from Moore et al. (2008)	27
2.8	Schematic representation of a multistage NVD system	29
2.9	Renewable energy and energy storage options for desalination technologies	33
3.1	Schematic representation of the proposed NVD system	37
3.2	The NVD process with the inlet/outlet streams	39
3.3	Different physics modules in COMSOL®	47
3.4	Different 3D meshing element types	48
3.5	Concentration change with time in the evaporation chamber	53
3.6	Concentration distribution in the evaporation chamber	54
3.7	Types of solar collectors	56
3.8	Geometrical aspects of a parabolic trough collector	57
3.9	Concentration ratio as a function of rim angle for different Incident angles	58

3.10	Flow of energy in the PTC absorber tube	59
3.11	Thermal resistance network showing the heat losses in the PTC	60
4.1	Photos of the experimental setup	66
4.2	Schematic representation of the proposed NVD system	67
4.3	Photos of the evaporation chamber	68
4.4	Photos of the condenser space	70
4.5	Water-ring vacuum pump allows removal of non-condensable gases and recovers the water vapor	71
4.6	Photos of the storage tanks and thermal fittings	72
4.7	Measuring instrumentation locations	73
4.8	Photos of the temperature measurement instruments	75
4.9	Photos of the pressure measurement instruments	76
4.10	Photos of the level measurement tubes	77
4.11	Photos of the ambience measurement instruments	78
4.12	Water heater in the feed water tank	82
4.13	Standard deviation and confidence intervals	86
5.1	Temperatures recorded at different depths in the evaporation chamber	95
5.2	Near surface water temperature variations with different heating temperatures	96
5.3	Pressure variations in the tank with time for two different operating temperatures	98
5.4	Evaporated amount with time at different operating temperatures	100
5.5	Evaporation and condensation amounts at two different operating temperatures	101
5.6	Total evaporated amount as a function of the operating temperature	102
5.7	Evaporation amount as a function of salt concentration	103
5.8	Temperatures recorded at different depths in the evaporation chamber at 45 °C set temperature	104
5.9	Temperatures recorded at different depths in the evaporation chamber at 70 °C set temperature	105
5.10	Near surface water temperature in the evaporation chamber at different set temperatures	105
5.11	Pressure change in the evaporation chamber with time	106
5.12	Freshwater production amounts at four different set temperatures	108
5.13	Temperatures recorded at different depths in the evaporation chamber	109

5.14	Average temperatures recorded in the evaporator chamber and the freshwater chamber	110
5.15	Pressure change in the evaporation chamber with time	111
5.16	Freshwater production amounts at different set temperatures	111
5.17	Geometry of the evaporator section defined in COMSOL	114
5.18	Meshing of the evaporator section showing refinements in the inlet/outlet regions	115
5.19	Comparison of water temperature from experimental and simulated results during batch operation	117
5.20	Comparison of freshwater production amounts from experimental and simulated results during batch operation	118
5.21	Comparison of water temperature from experimental and simulated results during continuous operation	119
5.22	Comparison of freshwater production amounts from experimental and simulated results during continuous operation	120
5.23	The plane selected for the temperature and velocity measurements in the evaporation chamber	121
5.24	Temperature and velocity profiles at different times in a horizontal plane in the evaporator	122
5.25	Temperature distribution along the centerline at different times on the top surface in a horizontal plane in the evaporator	123
5.26	The shaded area indicates the vertical plane selected for the temperature measurement in the evaporation chamber	124
5.27	Temperature distribution at different times in a vertical plane in the evaporator	125
5.28	Temperature distribution at different times along a vertical line through the center of the evaporator	126
5.29	Temperature distribution at different times along a vertical line above the heater in the evaporator	127
5.30	Temperature recorded by the probes at various heights in the evaporator	128
5.31	Changes in the evaporation rate with time in a continuous operation	128
5.32	Temperature and velocity distribution at the bottom of the evaporation chamber at different times	129

5.33	Temperature distribution along the centerline at different times on the top surface in a horizontal plane in the evaporator	130
5.34	The vertical plane selected for the temperature and velocity measurement in the evaporation chamber	131
5.35	Temperature and velocity distribution at the selected vertical plane along the center of the evaporation chamber at different times	132
5.36	Temperature distribution along the vertical line passing through the center of the evaporation chamber at different times	133
5.37	Temperature distribution along the vertical line passing through the center of the evaporation chamber at different times	134
5.38	Accumulated production amount as a function of the evaporation temperature	135
5.39	Accumulated production amount as a function of condenser side temperature	136
5.40	Accumulated production amount as a function of evaporator size	137
5.41	Accumulated production amount, recovery ratio and thermal efficiency as a function of the evaporation temperature	138
5.42	Accumulated production amount, recovery ratio and thermal efficiency as a function of condenser side temperature	140
5.43	Variation of daily production rate with temperature difference	141
5.44	Accumulated production amount, recovery ratio and thermal efficiency as a function of brine withdrawal rate	143
5.45	Accumulated production amount, recovery ratio and thermal efficiency as a function of the water column height in the evaporation chamber	145
5.46	Solar irradiation and input and output temperatures of the PTC (during the summer period)	147
5.47	Solar irradiation and input and output temperatures of the PTC (during the winter period)	147
5.48	Accumulated production amount for the summer and winter operation	149
5.49	Comparison of the recovery ratios during the summer and winter operation	150

5.50	Breakdown of the total energy requirements for the batch desalination process at different operating temperatures	152
5.51	Specific energy requirements for the batch desalination process at different operating temperatures	153

Appendices

No.	Appendix	Page No.
A	Appendix A – Evaporation Rate Calculation	A-1
	Evaporation Rate Calculation	A-1
	The Mass Accommodation Factor	A-6
	Comparison of the Evaporation Equations	A-8
B	Appendix B – Thermophysical Properties of Water	B-1
	Water Density as a Function of Temperature and Salinity	B-1
	Water Specific Heat as a Function of Temperature and Salinity	B-2
	Water Thermal Conductivity as a Function of Temperature and Salinity	B-3
	Water Specific Enthalpy as a Function of Temperature And Salinity	B-4
	Water Latent Heat of Vaporization as a Function of Temperature and Salinity	B-5
	Water Vapor Pressure as a Function of Temperature and Salinity	B-6
	Water Thermal Diffusivity as a Function of Temperature and Salinity	B-7

Nomenclature

List of symbols	
a	correlation parameter for thermophysical properties of saline water
A	area, m ²
A	absorptivity, dimensionless
b	correlation parameter for thermophysical properties of saline water
c	concentration, mass fraction
c	molar concentration, mol/m ³
C	specific heat capacity, J/kg.K
d	diameter, m
D	diffusion coefficient, m ² /s
E	absolute error, units depend on the quantity
f	correction factor for salt concentration, dimensionless
f	focal length, m
g	acceleration due to gravity, m/s ²
h	enthalpy, J
h	height, m
h	Convective heat transfer coefficient, W/m ² .K
I	solar radiation, W/m ²
k	thermal conductivity, W/m.K
K	Gain
L	length, m
m	mass, kg
$\dot{m}$	mass flow rate, kg/s
N	number of measurements
p	vapor pressure, Pa
P	pressure, Pa
Q	heat flow, W
r	reflectance, dimensionless
R	reaction rate expression, mol/m ³ .s
R	radius, m
S	salinity, g/kg
S	standard deviation
t	time, s
T	temperature, K
U	uncertainty, units depend on the quantity

V	volume, m ³
$\dot{V}$	volumetric flow rate, m ³ /s
W	width, m
X	any measured quantity, units depend on the quantity
$\bar{X}$	arithmetic mean, units depend on the quantity
Vector notations (in bold letters)	
$\mathbf{F}$	volume force vector, N/m ³
$\mathbf{I}$	identity matrix
$\mathbf{q}$	heat flux vector, W/m ²
$\mathbf{Q}$	heat sources, W/m ³
$\mathbf{S}$	strain rate tensor, 1/s
$\mathbf{u}$	velocity vector, m/s
$\boldsymbol{\tau}$	viscous stress tensor, Pa
Greek letters	
α	mass accommodation factor, dimensionless
δ	uncertainty pertaining to a certain confidence interval
ε	relative error, dimensionless
ε	emissivity, dimensionless
η	efficiency, dimensionless
θ	Angle
κ	factor for incident angle modifier, dimensionless
μ	viscosity, Pa.s
π	constant pi (ratio of a circle's circumference to its diameter)
ρ	density, kg/m ³
σ	Stefan-Boltzmann constant, W/m ² .K ⁴
τ	time constant, s
Subscripts	
0	initial or reference state
a	absorber tube
ap	aperture
b	brine
c	cross-sectional
cond	conductive
conv	convective
dn	direct normal
ev	evaporation
ext	external
f	fresh (or distilled) water

f	fluid
fg	latent heat of vaporization (from fluid to gas)
focus	directly focused on the glass tube surface
g	glass tube
H	Supplied
i	incident
in	inlet
L	lost
m	modified (when used with mass accommodation factor)
max	maximum
min	minimum
out	outlet
P	at constant pressure
r	rim
rad	radiative
s	saline (or feed) water
set	set or desired value of the heater temperature
sky	sky (as a blackbody)
sw	saline water
th	thermal
u	useful
w	pure water
∞	referring to the ambient condition far from the body's surface
Abbreviations	
ARS	Absorption Refrigeration System
ASTM	American Society for Testing and Materials
BDF	Backward Differentiation Formula
BHD	Bahraini Dinar
CAD	Computer Aided Design
CFD	Computational Fluid Dynamics
CPC	Compound Parabolic Collector
CR	Concentration Ratio
CTC	Cylindrical Trough Collector
DNI	Direct Normal Irradiance
FEM	Finite Element Method
HFC	Heliostat Field Collector
IAM	Incident Angle Modifier
LCOE	Levelized Cost of Electricity
LFC	Linear Fresnel Collector
MED	Multi Effect Distillation

MSF	Multi Stage Flash
NVD	Natural Vacuum Distillation
PDC	Parabolic Dish Collector
PDE	Partial Differential Equation
PPR	Polypropylene Random
PTC	Parabolic Trough Collector
PV	Photovoltaics
RHPDS	Refrigeration Heat Pump Desalination System
RO	Reverse Osmosis
RR	Recovery Ratio
STE	Solar Thermal Energy
TES	Thermal Energy Storage
USD	United States Dollar

Chapter 1

Introduction

Table of Contents

Subject	Page No.
Chapter One Introduction	1
1.1 Water Desalination	2
1.2 Overview of Desalination Technologies	3
1.2.1 Reverse Osmosis (RO)	4
1.2.2 Multi-Stage Flash Distillation for Desalination	6
1.2.3 Multiple Effect Distillation	7
1.2.4 Comparison of RO, MSF and MED	9
1.3 Renewable Energy in Water Desalination	9
1.4 Research Objectives	10
1.5 Research Significance	11

Chapter 1: Introduction

1.1. Water Desalination

The worldwide demand for “fresh” water is growing exponentially, while the supply of readily available fresh water is dwindling nearly as quickly. Several diverse techniques have been implemented to try to meet the growing demand for fresh water, with variable degrees of success. One technique that is growing in application is desalination. Desalination encompasses a host of technologies so that clean water may be generated regardless of location, make-up source, and/or energy source.

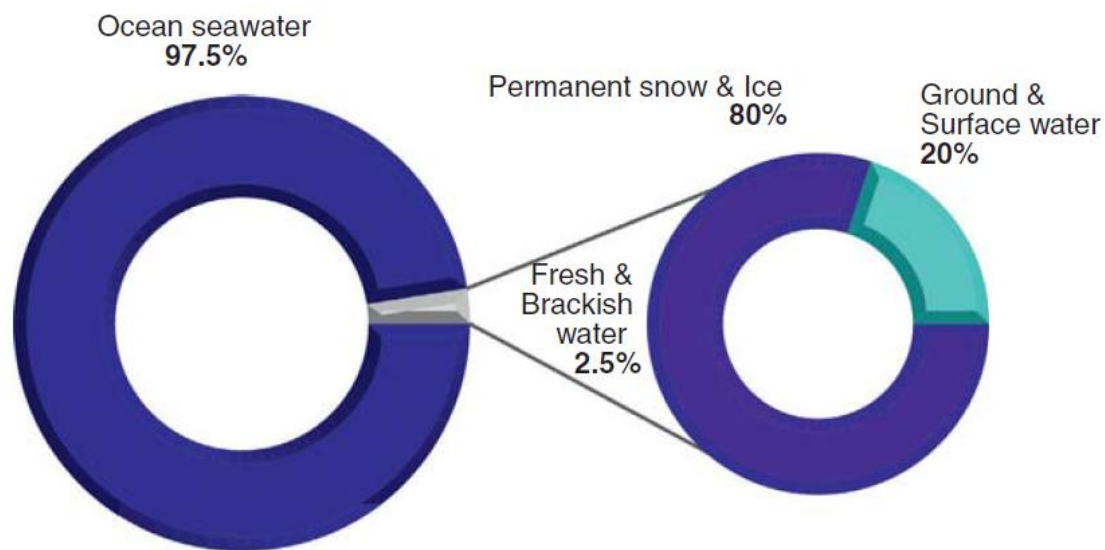


Figure 1.1: Allocation of the world’s water resources (Kucera, 2014)

The allocation of the world’s water is shown in Fig. 1.1. More than 97%, or about 1338 million km³, of the world’s water is seawater (Kucera, 2014), and of the remaining water, around eighty percent is in the form of snow in glaciers or as permafrost. Hence, only 0.5% of the world’s water is present as “freshwater” in the lakes and rivers and other low-salinity reservoirs, which can be used directly by humans.

For the population residing in the cities of the developed world, freshwater was and is not even a problem that they consider. It has always been there at the tap whenever they needed it. However, in many other parts of the world, the demand for freshwater is increasing sharply and there are very few options available to fulfil this demand. Based on the specific circumstances in any given region, different methods may need to be explored to find and develop water sources to overcome the shortages in freshwater supply. Some of these methods are listed below:

1. Transport of water from water-rich to water-scarce areas
2. Conservation and reuse
3. Development of new sources of fresh water
4. Desalination

Transport of water is an expensive option, especially for regions where there are no sources of freshwater nearby. Conservation and reuse may be applicable to areas where freshwater sources exist but for arid and semi-arid regions, this option might not be applicable. Similarly, such regions which have no rivers or very little or no rainfall, development of new sources or reservoirs would also not be possible. For such reasons, desalination seems to be the only promising and dependable option for many parts of the world.

1.2. Overview of Desalination Technologies

The world-wide installed desalination capacity in 2014 was about 86.5 million m³/d. Fig. 1.2 shows the relative breakdown of installed capacity of various desalination technologies for 2014 (Burn et al., 2015).

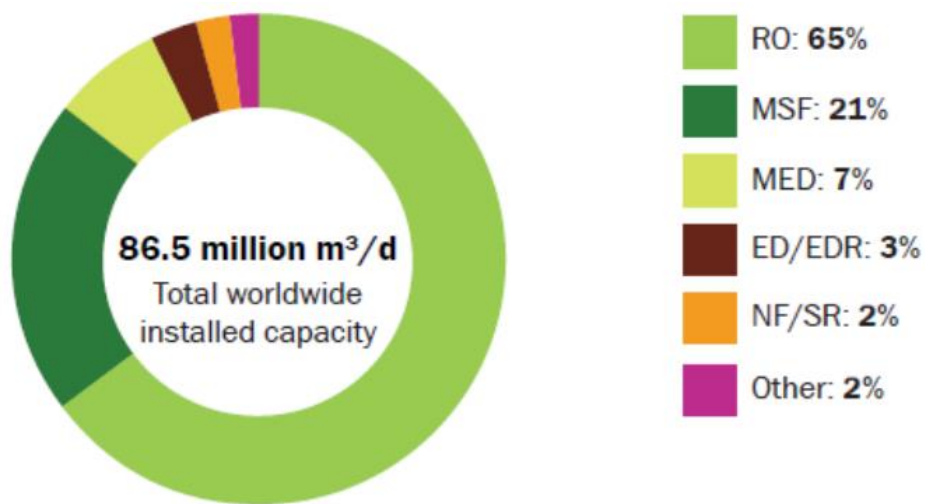


Figure 1.2: Global installed desalination capacity by technology for 2014
(Burn et al., 2015)

As can be seen from Fig. 1.2, the most popular methods for water desalination are reverse osmosis (RO), multi-stage Flash (MSF) and multi-effect distillation (MED). A brief description of each of these processes is given below:

1.2.1. Reverse Osmosis (RO)

A reverse osmosis membrane is a semi-permeable membrane that allows the passage of water molecules but not the majority of dissolved salts, organics, bacteria and pyrogens. If a saline water solution is separated from pure water by a semi-permeable membrane, water molecules move from the desalinated side to the saline solution. Osmotic pressure is defined as the pressure required to prevent such transport of the water molecules. Applying a pressure greater than the osmotic pressure helps push the water molecules across the membrane, from the saline side to the desalinated water. Such a process is called reverse osmosis (RO). Fig. 1.3 outlines the process of reverse osmosis. When pressure is applied to the concentrated

solution the water molecules are forced through the semi-permeable membrane, while the salt molecules and other contaminants are held back.

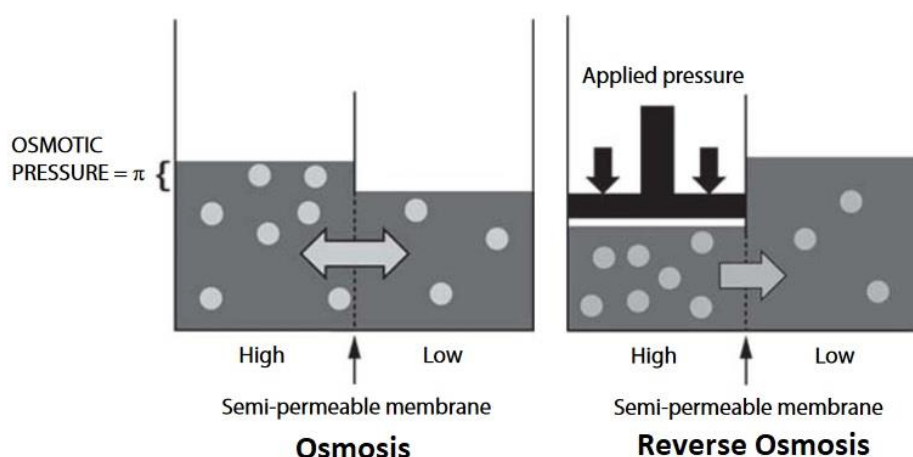


Figure 1.3: The principle of Reverse Osmosis (Kucera, 2015)

In a reverse osmosis desalination process a high pressure pump is used to increase the pressure on the salt side of the RO membrane and force the water across the semi-permeable RO membrane as shown in Fig. 1.4. Almost all (around 95% to 99%) of the dissolved salts are left behind in the reject stream. The amount of pressure required depends on the salt concentration of the feed water. The more concentrated the feed water the more pressure is required to overcome the osmotic pressure.

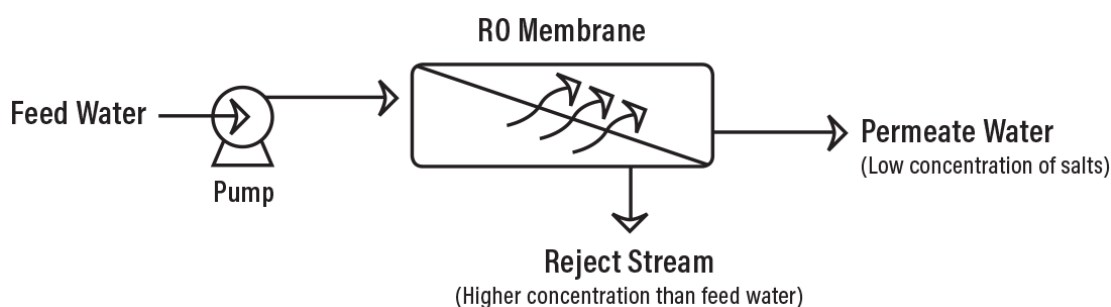


Figure 1.4: Simple illustration of the RO process (Puretec, 2015)

A typical seawater RO plant consists of the following four major stages (Khawaji et al., 2008):

1. Feed water pre-treatment (to remove suspended solids and unwanted contaminants that cause membrane fouling),
2. High pressure pumping (to provide the required pressure for the water molecules to overcome the osmotic pressure),
3. Membrane separation (the membrane has to be strong enough to withstand the high pressures),
4. Permeate post-treatment (includes pH adjustment, removal of dissolved gases and disinfection).

1.2.2. Multi-Stage Flash Distillation for Desalination

The Multi-Stage Flash (MSF) distillation process is based on the principle of flash evaporation. In the MSF process, seawater evaporation is achieved by lowering the pressure instead of increasing the temperature. For better energy economics, MSF desalination is carried out in a pressure vessel that is divided into numerous sections that have a set of pipes with seawater pumped through them. These pipes serve two purposes: condensing the steam produced by the flash evaporation and at the same time, preheating the seawater to almost the required temperature for evaporation.

Seawater heating is accomplished in the seawater heater by low-pressure steam externally supplied by a cogeneration power plant. The seawater is heated to a temperature between 70 °C and 100 °C; the seawater now being considered to be brine. Lower temperatures maybe used to decrease the formation of scale. The brine now passes into the different

stages which are maintained at low pressure and, as it is at a higher temperature than the sections, some of it flashes to steam. The vapor produced is condensed by the cold seawater entering the pipes at the top of the sections. The condensate falls onto the collection trays under the pipes. This process continues to the final stage where the brine is discarded and the distilled water is collected and passed on to storage tanks. A typical multi-stage flash distillation plant is shown in Fig. 1.5.

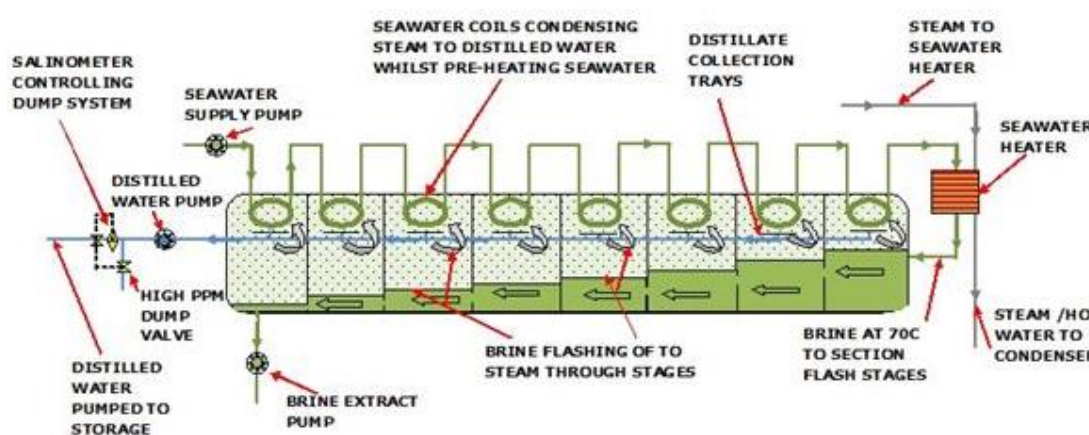


Figure 1.5: Multi-stage Flash process (Scott, 2011)

1.2.3. Multiple Effect Distillation

The multiple-effect distillation (MED) process is the oldest desalination method (Al-Shammiri & Safar, 1999), yet very efficient. MED is a low temperature thermal process of obtaining fresh water by recovering the vapor of boiling seawater in a sequence of vessels, (called effects) each maintained at a lower temperature than the last. The process makes use of the fact that the boiling point of water decreases as pressure decreases. Only the first effect (which is maintained at the highest pressures) requires external heating, while the vapor boiled off in one effect is used to heat the water in the subsequent one. A schematic diagram of the MED process is shown in Fig. 1.6.

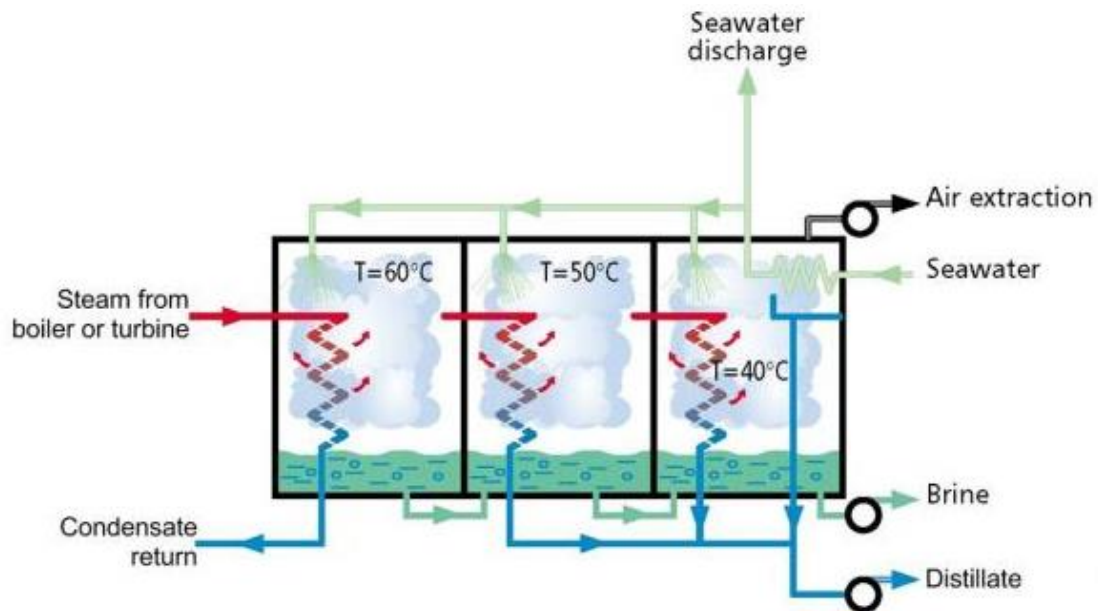


Figure 1.6: Multi-effect distillation process (Veolia, 2016)

Each effect contains a horizontal tube bundle. Incoming seawater is sprayed from the top of the effect onto the top of the bundle and it then flows down from tube to tube by gravity. The steam used for heating is sent through these tubes. The steam inside the tube cools and condenses as the cold seawater comes into contact with the outer surface of the tubes. At the same time, the heat released by the condensation (latent heat) causes the seawater to be heated and partly evaporates it. Due to evaporation, the seawater concentration increases and brine is collected at the bottom of the cell. The vapor produced by the evaporation of seawater is at a lower temperature than heating steam but it can still be used as a heating medium for the next effect. The process repeats in all effects. The brine and distillate are collected from each effect until the last one, from where they are extracted by pumps.

1.2.4. Comparison of RO, MSF and MED

Table 1.1 presents a comparison of some important parameters for these three popular desalination technologies, i.e. RO, MSF and MED. The data are obtained from a recent article (Youssef et al., 2014).

Table 1.1: Comparison of RO, MSF and MED

	RO	MSF	MED
Feed water salinity (ppm)	45,000	70,000	45,000
Product water salinity (ppm)	10	10	10
Electrical energy required (kWh/m ³)	8.2	5.2	3.8
Thermal energy required (kWh/m ³)	0.0	19.4	16.4
Water production cost (\$/m ³)	0.75	0.96	0.86
CO ₂ emissions (kg/m ³)	3.8	6.9	5.5

As can be seen from the table, all three processes are able to handle high levels of feed water salinity. The product quality obtained by the processes is almost 10 ppm. Generally, RO has a product water salinity of around 200 ppm (Shenvi et al., 2015). The important thing to note here is that the energy required (electrical, thermal or both) is substantial. Moreover, the CO₂ emissions associated with these processes is also a worrying number. Hence, due to economic and environmental factors, it is almost necessary to look for alternative desalination technologies.

1.3. Renewable Energy in Water Desalination

The cost of water desalination varies depending on water source access, source water salinity and quality, specific desalination process, power costs, concentrate disposal method, project delivery method, and the distance to the point of use. Power costs in water desalination may account for 30% to 60% of the operational costs; thus, slight variations in power rates directly impact the cost of treated water.

Using renewable energy sources in water desalination has many advantages and benefits. The most common advantage is that they are renewable and don't deplete. They provide a clean energy, not polluting the air, and they do not contribute to global warming or greenhouse gas emissions. Because their sources are natural, operational costs are reduced and they require less maintenance of their plants. Using these resources in water desalination in remote areas also present the best option because of the very high cost involved in providing energy from the grid.

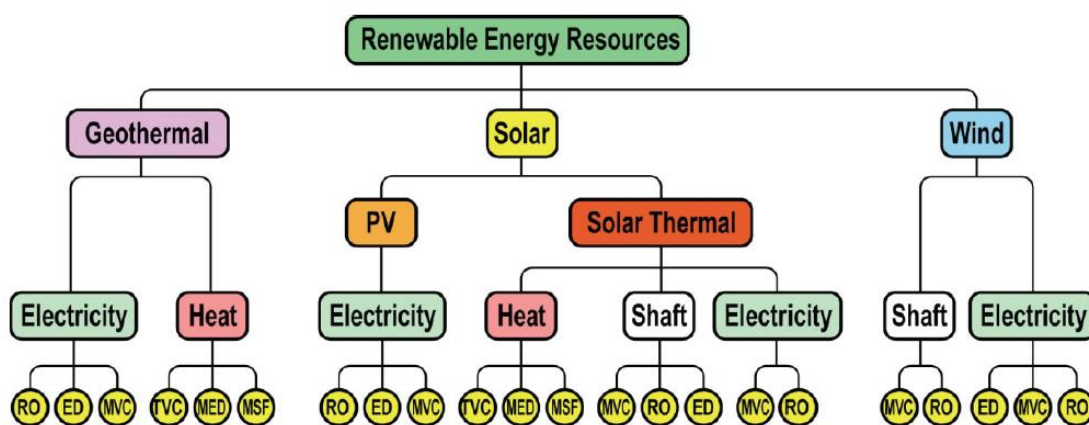


Figure 1.7: Possible combinations of renewable energy systems with desalination units (Eltawil et al., 2009)

Renewable energy can be used for seawater desalination either by producing the thermal energy required to drive the phase-change processes or by producing electricity required to drive the membrane processes. The major sources of alternative energy are solar, wind, and geothermal. Fig. 1.7 shows the possible combination of renewable energy systems with desalination units.

1.4. Research Objectives

The primary objective of this research is to study the performance of a water desalination system that is operated using the natural vacuum distillation (NVD) technique. The research will focus on a theoretical and

experimental investigation. A pilot scale experimental system will be built using a novel inverted cone design. A parametric optimization will also be carried out using the simulation model. A rudimentary economic analysis will also be done.

1.5. Research Significance

This research work has the potential to provide great benefit to the scientific community and researchers. The proposed desalination system operates under natural vacuum and leads to an efficient integration of renewable energy sources and low grade or waste heat sources for water desalination. The proposed NVD system has a novel inverted cone design, which could enhance the condensation rate. This new geometry also is not prone to the problem of the movement of water from the evaporation chamber to the condenser side, as was experienced by previous researchers. Unlike the previous works that have been on smaller scales, the proposed experimental setup is on pilot scale. Previous studies have been carried out using very simplistic models. In this work, a 3D model is used for theoretical analysis, with simulation using COMSOL Multiphysics (a finite element software).

Chapter 2

Literature Survey

Table of Contents

Subject	Page No.
Chapter Two Literature Survey	12
2.1 Natural Vacuum Distillation	13
2.2 Single Stage NVD Systems	15
2.3 Multi-Stage NVD Systems	29
2.4 Thermal Energy Storage	32
2.5 Benefits of NVD Systems	34

Chapter 2: Literature Survey

In this chapter, an overview of the natural vacuum distillation process is presented followed by a critical review of the studies published on desalination systems using this principle.

2.1. Natural Vacuum Distillation

Natural vacuum distillation (NVD) is based on the simple principle of using the barometric pressure of a column of water to create a vacuum chamber, which allows for evaporation of water at relatively lower temperatures. If a column of water, with a sealed top, is allowed to drain into an open container of water, then it stops draining after the head of the water column equals the atmospheric pressure acting on the water in the container. The space above the water column will thus have a natural vacuum created. Vacuum attained by such means is called “Torricelli vacuum” named after the Italian physicist Evangelista Torricelli (Daintith, 2009). During his investigation into the problem of the failure of pumps to deliver water to heights of above 32 feet, Torricelli was able to demonstrate this principle using a column of mercury (Tokaty, 1994). An illustration of this principle is presented in Fig. 2.1. The same principle is used in the construction of a barometer. Due to the difference of the densities of mercury and water, the barometric head is significantly different. A column of mercury with a height of only 0.76 m is required to balance atmospheric pressure, whereas the height required to balance the same pressure with a column of water is about 10.3 m.

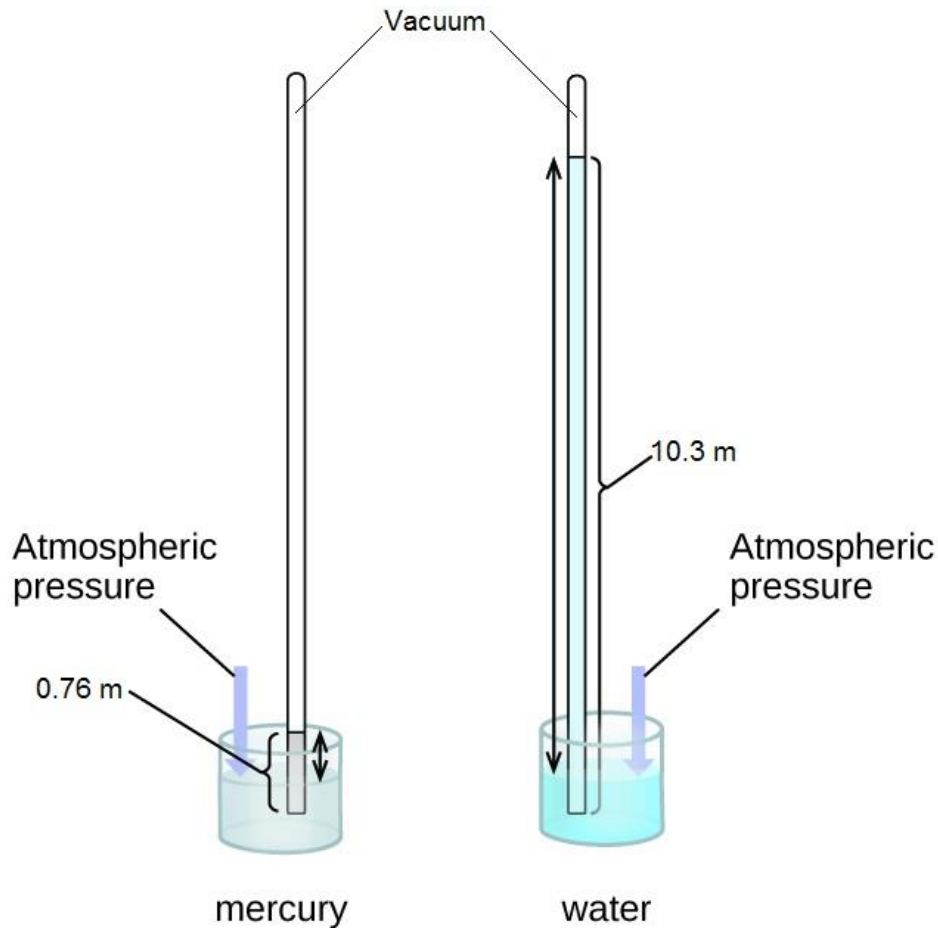


Figure 2.1: Illustration of the Torricelli vacuum created over a column of liquid mercury or water

The vacuum space created above the water column using such a technique does not require any evacuation pump. At low-pressure conditions, saline water can be easily evaporated at low temperatures, with minimal energy input, and subsequently condensed to obtain distilled water.

The different studies that have been conducted on desalination systems using NVD have been classified into two categories: 1) single stage systems and 2) multi-stage systems. Most of the earlier investigations have been done on single-stage systems only. But recently, some efforts have been made to study the multi-stage systems. Table 2.1 summarizes

the findings of the different studies. Detailed discussion of these systems is presented in the sections to follow.

Table 2.1. Summary of the various NVD systems reported in literature

Year	Author(s)	E/T*	Primary heating source	Production Rate (kg/day)	Energy/Power Requirements	Production cost	Evaporator Size (m ²)
Single Stage Processes							
1982	Parker	T	-	-	-	-	-
1984	Realı	T	heat pump	-	-	-	-
1995	Bemporad	T	Solar	1.7	50 W	-	0.047
2003	Al-Kharabsheh and Goswami	T	Solar	0.58	150.3 W	-	0.1
		E	Solar	0.65	158 W	-	0.1
2003	Midilli and Ayhan	T	Solar	235	-	-	0.16
		E	Solar	95	2.6 kWh/kg	-	0.16
2004	Realı et al.	T	Solar	1×10 ⁵	2.162 kWh/m ³	\$ 5/m ³	3.68×10 ³
2007	Eames et al.	E	Solar	30	4.7 m ² solar panels	-	
2008	Gude and Nirmalakhandan	T	Solar	108	15 m ² solar collector	-	0.2
		T	solar PV	108	23 m ² of PV panels	-	0.2
		T	solar/TES	108	15 m ² panel/1 m ³ TES	-	0.2
		T	waste heat	108	260 W	-	0.2
		E	Solar	192	reflector	-	0.2
2008	Moore et al.	T	electrical	350	1.97 kWh/m ³	-	-
2010	Ayhan and Al-Madani	E	Solar	130	1.6 kW	\$ 0.7/m ³	-
2012	Jitsuno and Hamabe	E	electrical	10	550 W	-	-
2016	Ambarita	T	electrical	17.7	-	-	0.2
		E	electrical	3.6	1500 W	-	0.2
2016	Attia	T	-	-	40.6 kWh/m ³	\$ 0.62/m ³	-
Multistage Processes							
2007	Realı	T	Solar	36×10 ³	1.0 kWh/m ³	\$ 5/m ³	-
2008	Abutayeh and Goswami	E	electrical	40	4.87 kW	-	0.1
2009	Maroo and Goswami	T	Solar	26	1 m ² solar collector	-	0.2
2012	Gude et al.	E	-	500	8.7 kW	\$ 3/m ³	1.5

* E = Experimental, T = Theoretical

2.2 Single Stage NVD Systems

Possibly, the earliest desalination system designed to work using the principle of NVD was suggested by Parker in 1982, under a US Patent (Parker, 1982). A simple representation of this system is shown in Fig. 2.2.

It consisted of two columns, one for saline water and the other for distilled water. The columns were to have a height greater than the barometric head of water. Parker (1982) suggested using a height of approximately 12.2 m (40 ft) to 18.3 m (60 ft) for each of the saline water and the distilled water column. Both columns were connected at the top through a vapor crossover conduit as shown in Fig. 2.2. This conduit inclined downward to the distilled water column to allow vapor to travel from the saline water column to the distilled water column.

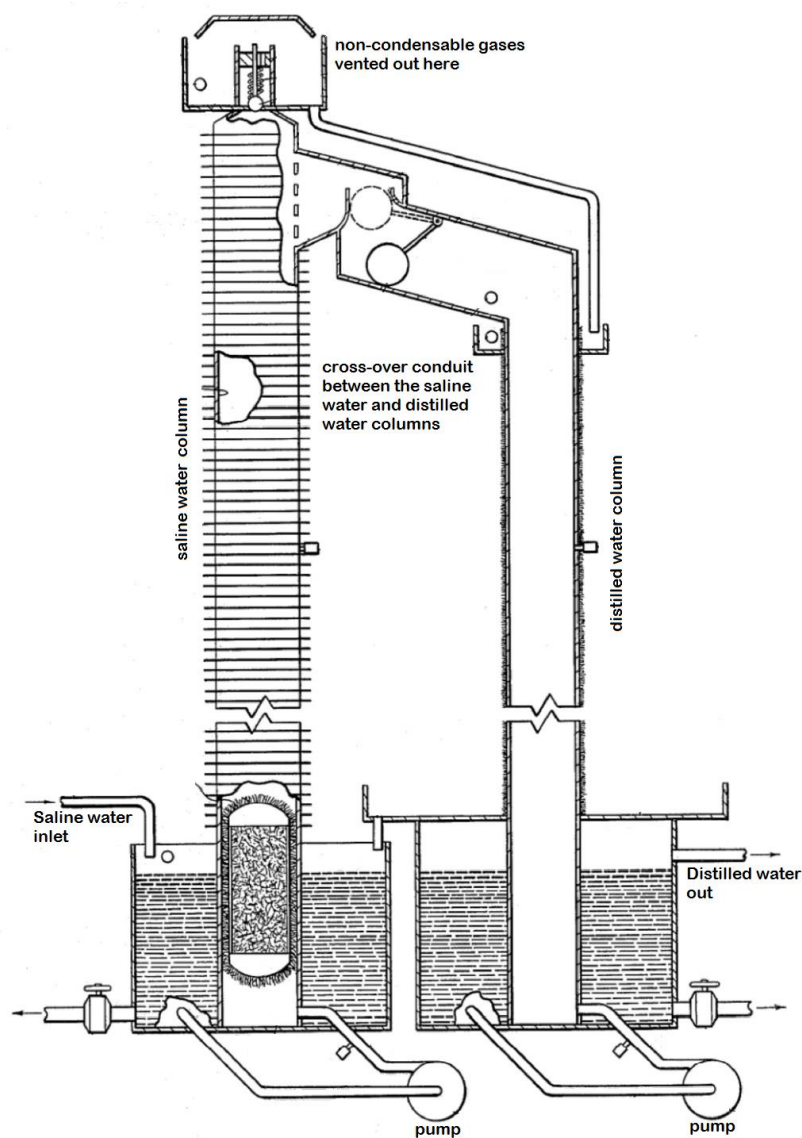


Figure 2.2: Illustration of the NVD desalination system proposed by Parker (1982)

Parker (1982) stated that any form of low-grade heat source, including heat from ambient air could be used to heat the water in the saline water column to cause vaporization. This column was painted black to increase the absorption of heat radiation from the sun. Parker suggested that external radial fins could be attached to enhance the heating process of the water during the daylight hours. The distilled water column was fitted with an external wetting wick to keep it sufficiently cool for the condensation of the water vapor. Both the saline water and the distilled water should be continuously pumped into and out of the system, respectively to ensure that the system remains under a steady process. The production rates or energy requirements of the proposed system were not mentioned.

The benefits of such a distillation process, as mentioned by the author, are as follows:

- Low temperatures can be used to carry out single stage distillation of saline water.
- Due to the low pressures, hazardous gases dissolved in the saline water are more efficiently removed. Appropriate provisions have to be made for the removal of such gases.
- As the water is subjected to a sudden pressure drop from atmospheric pressure to almost total vacuum, the DNA of most of the living organisms in the water is destroyed resulting in safer distilled water.

The importance to be mentioned of this work is that it was the first study to introduce a desalination system based on NVD. Although no theoretical or experimental investigation was reported on the system, it did serve as a starting point for other researchers.

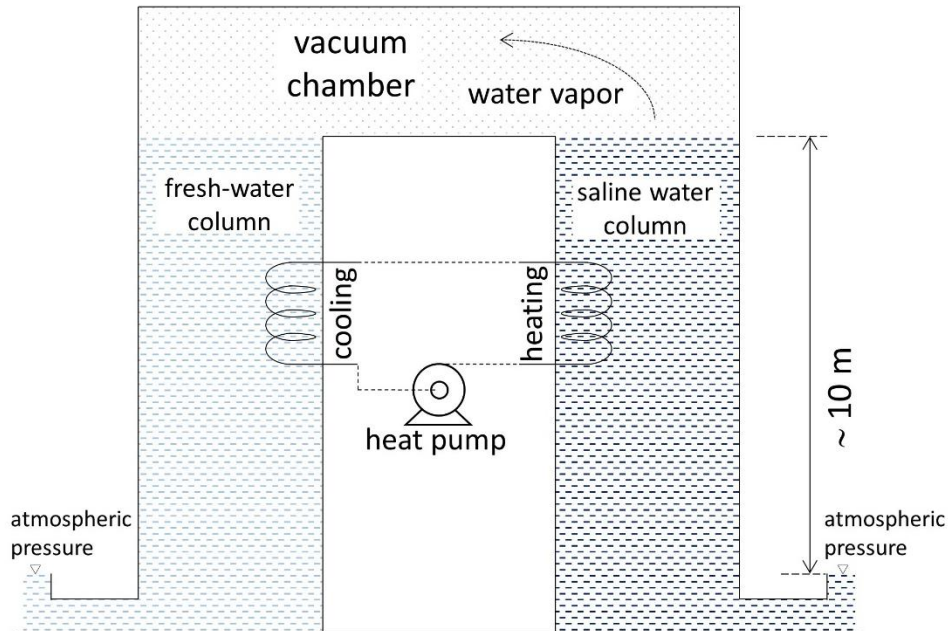


Figure 2.3: Simple representation of a desalination system using NVD principle

In 1984, M. Reali presented an NVD scheme consisting of two columns of water that were connected at the top by a horizontal vacuum chamber (Reali, 1984). The saline water column had a water level almost 10 m above the sea level, into which the seawater was pumped automatically by the atmospheric pressure. The scheme was named as the refrigeration-heat-pump desalination system (RHPDS). A refrigeration heat pump was used to absorb the heat from the distilled water column and transfer it to the saline water to raise its temperature is presented in Fig. 2.3. The concentrated brine moved downward under the influence of gravity from where it could be withdrawn using a pump. The water evaporated at the surface, along with the dissolved gases like N_2 , O_2 and CO_2 . These gases could be removed from the system using an appropriate venting scheme in the vacuum chamber. The water vapor condensed into the fresh-water chamber, which was maintained at a temperature below the saline water chamber. Calculations for energy requirements were shown

based on a fresh-water temperature of 20 °C and a saline water temperature of 25 °C. The author estimated that 2461 kJ of energy was required to produce 1 kg of water under these conditions. No detailed calculation of energy requirements or further investigation of the process was shown in this study. In addition to the distilled water produced, Reali (1984) suggested that salt could be produced from the brine. He also mentioned the idea of a multi-stage process to operate in series for a better production rate. Industrial waste heat and renewable energy sources, including solar and wind energy, were mentioned as options to provide the required energy for operation of the process.

Bemporad carried out a theoretical investigation of the hydrodynamic behavior of the system proposed by Reali (Bemporad, 1995). In this study, a proposed equation to predict the evaporation rate and a mathematical model to simulate the process were presented. A numerical integration scheme was employed to solve the main equations and study the system. The obtained results predicted water production rates of 1.78 L/day under steady-state condition and 1.08 L/day for the unsteady-state operation (initial start-up period). A column diameter of 0.25 m and fresh-water temperature of 10 °C was assumed in the simulation.

A comprehensive theoretical and experimental study on a desalination system based on the NVD was first presented by Al-Kharabsheh and Goswami in 2003 (Al-Kharabsheh & Goswami, 2003a, 2003b). A schematic representation of the investigated system is shown in Fig. 2.4. A new feature in this system was the introduction of a third column for brine withdrawal in addition to the saline and the distilled water columns. The saline water in the evaporator was heated using a heat exchanger. The heat exchange medium was heated using a solar collector. The produced vapor

was then condensed and the distilled water was collected in the distilled water tank.

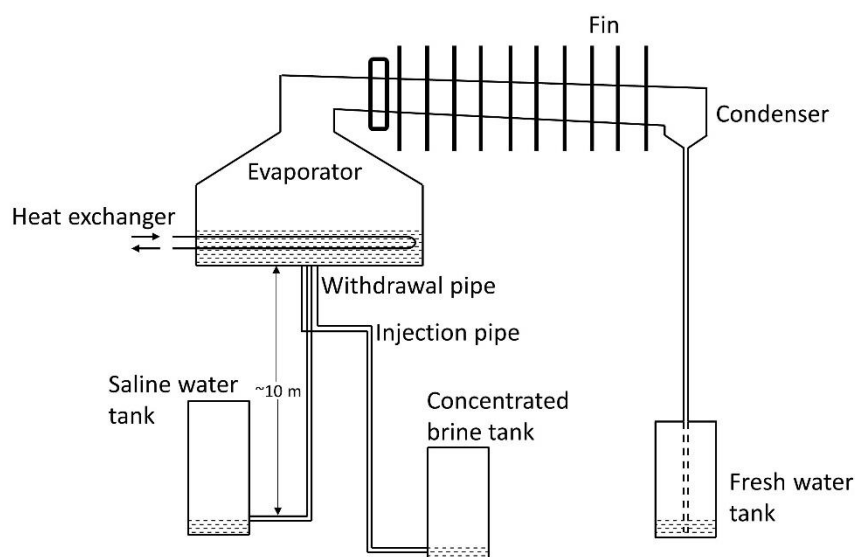


Figure 2.4: Schematic representation of desalination system using the NVD process by Al-Kharabsheh and Goswami (2003) [*reproduced with permission*]

The system was theoretically modelled using energy, overall and solute material balances (Al-Kharabsheh & Goswami, 2004). Evaporation rates in the evaporator were modelled according to the model developed by Bemporad (1995). The model was used to find optimum working conditions and parameters for the system. Four values (4, 6, 8 and 10 cm) for the depth of water in the evaporator were investigated and it was seen that a lower depth was preferable. Lower water depth meant less water used and a lower energy input. Different brine withdrawal rates were also investigated. The production rate was found to decrease with an increase in the brine withdrawal rate above 0.1 kg/h. The output rate was shown to be mainly dependent on the saline water temperature. An increased temperature on the saline water side allowed for a higher temperature difference between the two sides (which was the driving force). A production rate of 0.174 kg/h was predicted by the model under an

operating pressure of 4.8 kPa, a heat source temperature of 60 °C and a brine withdrawal rate of 0.1 kg/h.

This theoretical investigation was followed by a batch experimental study. The tests were carried out with an operating period of just six hours. The experimental system had an evaporator area of 0.2 m² and was heated by an electric heater to simulate the effect of solar heating. A water depth of 0.08 m was selected for the experimental study. The experimental production rate was lower than that predicted by the simulated process by about 11% for the 6 hour test. The authors claimed an 18% decrease in the experimental value for the steady-state condition. This discrepancy was attributed to the two assumptions used in the simulation model. The assumptions were that all the vapor produced in the evaporation chamber condensed on the distilled water side and that the heat losses in the pipes and connectors were negligible (Al-Kharabsheh & Goswami, 2003b).

Another theoretical study, almost concurrent to the work of Al-Kharabsheh and Goswami, was presented by Midilli and Ayhan (Midilli & Ayhan, 2004a). The investigated system was composed of two water columns connected by an overhead vacuum space. It was designed in the form of an inverted U-shape. Thermodynamic analysis was presented for the process under free and forced convection. For the latter case, a fan was needed to drive the vapor from the evaporator side to the condenser. The process was designed to use renewable energy.

An experimental system was setup by Midilli and Ayhan based on the theoretical investigation (Midilli & Ayhan, 2004b). The experiments were carried out with free and forced mass convection. They yielded a product rate much lower than that predicted from the model. From the figures presented in the paper, the maximum distilled water production rates corresponding to the theoretical and experimental investigations are 9.79

kg/h and 3.96 kg/h, respectively. The energy consumed by the experimental setup was almost three times higher than that predicted by the theoretical analysis. The authors attributed the disparities in the production rates and the energy consumption to heat and pressure losses in the connections.

Reali and co-workers proposed another system based on NVD, though they chose to call it differently as “solar barometric desalination” (Reali et al., 2004). The system used intake from the sea and the storage tanks were all kept at almost 10 m below the sea level to achieve the required barometric height of water. A special venting system was used to remove non-condensable gases. Another important feature of this system was the use of a “demister” to separate the water vapor from the salt-water droplets. A solar collector field using flat plate solar collectors was used to heat the water. Theoretical calculations were carried out for the proposed system based on a production rate of 100 m³/day and an operation time of 10 hours per day. Assuming a 20-year plant lifetime, the cost of production was estimated to be around 5 \$/m³, with 4 \$/m³ for capital costs and 1 \$/m³ for operation and maintenance costs. The cost was high due to the high cost of solar photovoltaic panels at the time of the study. The price should be much lower now, given the rate at which the cost of solar photovoltaic panels has dropped. The total power consumption was found to be 2.162 kWh/m³. The solar collector area required to produce the afore-mentioned amount of water was calculated as 3.68×10³ m², which is almost the size of a football field. The numbers reported by this study appear to be very high.

Building on these early ideas, Gude and Nirmalakhandan presented a modified system, which could use low-grade heat sources to produce distilled water (Gude & Nirmalakhandan, 2008) as shown in Fig. 2.5. Theoretical models were presented to make use of various renewable

energy sources to be used in the system including solar energy, solar photovoltaic, thermal energy storage and geothermal energy as well as process waste heat. The process was built and operated on a continuous mode. The heating energy required was supplied by grid power. A production rate of 8 L/d was obtained using 3122 kJ per kg of distilled water. They extended the study to include a thermal storage tank coupled with solar photovoltaic and heat rejected from an absorption refrigeration system (ARS). Results of this study showed that the heat rejected by an ARS of cooling capacity of 3.25 kW of refrigeration, along with an additional energy input of 208 kJ/kg of distilled water is adequate to produce desalinated water at an average rate of 4.5 kg/h. The absorption refrigeration system was driven by solar energy during sunlight hours and by an auxiliary electric heater during non-sunlight hours (Gude et al., 2011).

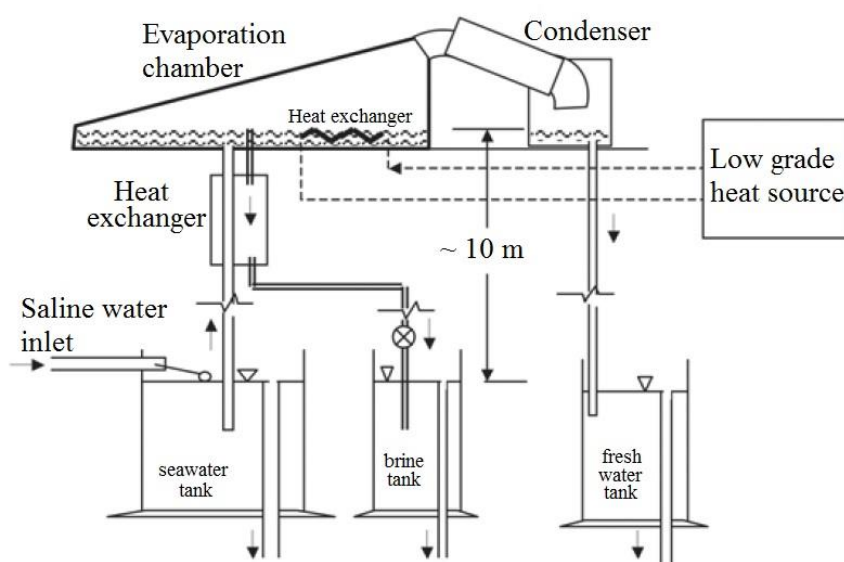


Figure 2.5: Schematic representation of desalination system using the NVD process by Gude and Nirmalakhandan (2008) [*reproduced with permission*]

Gude et al. (2012) also performed a theoretical investigation on this system using flat plate solar collectors with a thermal energy storage (TES) tank. The TES was used to drive the process during the nighttime when solar energy was not available. They concluded that the system could produce 100 L/d of distilled water with a solar collector area of 15 m² with 1 m³ of TES volume or 18 m² with 3 m³ of TES volume. Such a low-energy input system was found to be very economical and a payback period of only 2 years was estimated. The authors also investigated the possibility of using this system for wastewater reclamation (Nirmalakhandan & Gude, 2010). Results showed removal of more than 95% of the dissolved salts and 100% of the bacteria from the feed water.

From the different setups considered by Gude et al. (Gude & Nirmalakhandan, 2009, Gude et al., 2010, Gude & Nirmalakhandan, 2010), it was found that the best arrangement was to use the solar collector coupled with photovoltaic battery (Gude et al., 2012), where the solar energy was used directly for heating during the sunlight hours and the energy stored in the photovoltaic battery was used when sunlight was not available. A comparison of this arrangement was done with the previous studies (Gude & Nirmalakhandan, 2008, 2009), and it was found that this system was completely based on renewable energy and did not rely on the grid power or fossil fuels for heating or mechanical pumping power requirements. The condenser used was just a cooled space kept under the shade. This is feasible in areas where the difference between the temperatures in the open area and the shaded area is above 10 °C. However, for climatically hot regions a cooling mechanism needs to be developed. An exergy analysis was also carried out for the different setups. It was found that low-grade heat source had the least exergy destruction while

solar energy was more suited to thermal processes due to its higher exergy values (Gude et al., 2012).

Ayhan and Al-Madani presented a feasibility study of using NVD in hot climate regions (Ayhan & Al-Madani, 2010). They showed some promising results with a production rate of around 130 L/day for an evaporator area of 1 m². They also calculated the production cost of distilled water in the Kingdom of Bahrain environment as 0.7 \$/m³. The authors claimed that this is significantly lower than the water production cost in Bahrain (which is close to 2 \$/m³). The production rate was shown to significantly increase by creating a combined effect of evaporation and cavitation (Ayhan & Al-Madani, 2015). The temperature difference between the evaporation chamber and the saline water supply was adjusted in a manner to cause cavitation within the chamber and the bubbling helped in increased evaporation rates.

Recently, Jitsuno and Hamabe have conducted experiments on a single-stage setup evacuated using a vacuum pump (Jitsuno & Hamabe, 2012). They asserted that NVD could be used for actual operation in the arid lands of Africa, which was the prime objective of their study. They reported a production rate of 10 kg/m² per day and claimed a life span of more than 30 years for such a system.

Eames and co-workers developed a set-up that used natural vacuum to create an evacuated flash chamber (Maidment et al., 2006, Eames et al., 2007). Cold saline water was entered into the condensation side of the chamber, before being sent to a solar collector for heating and then sprayed through a nozzle into the flash chamber. Brine settled on one side while the vapor was condensed by the cold feed on the condenser side. In a naturally evacuated chamber like this, a flash evaporation process can be used to desalinate water at relatively low temperatures, typical of those available

from low-cost solar-thermal collectors. The setup of this theoretical process is shown in Fig. 2.6.

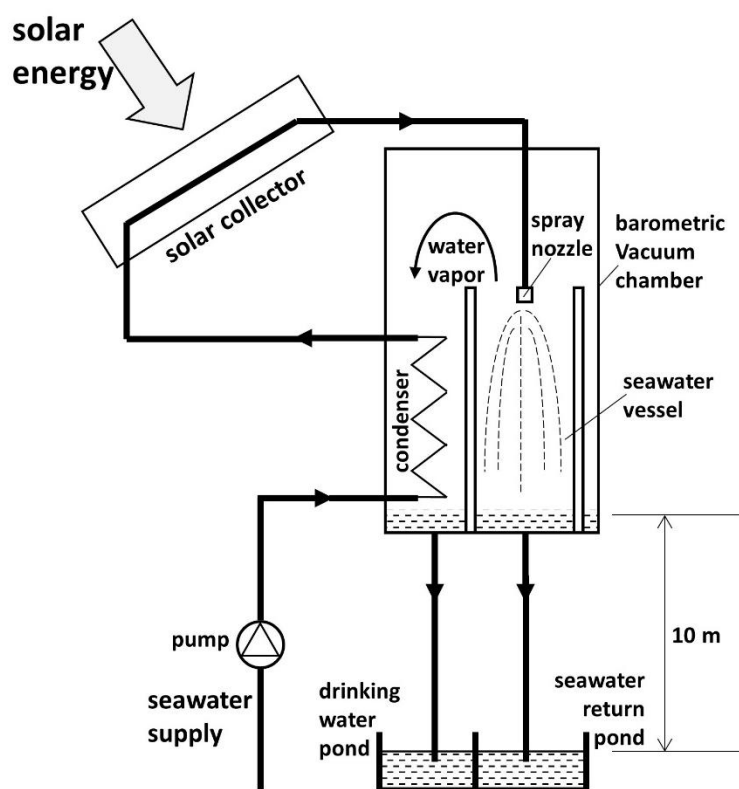


Figure 2.6: NVD based humidification de-humidification desalination process by Eames et al. (2007) [*reproduced with permission*]

Theoretical investigation of the system showed that a distilled water rate of 30 L/day is possible using a 4.727 m² of solar collector area. Experimental studies were carried out on the system using an electrical heater to simulate the solar collector. The duration of operation for the experimental tests was not reported. However, from the 19 runs carried out with the experimental setup, production rates ranging from 3.8 to 9.7 kg/h were reported based on varying feed water rates (from 1.5 to 5 kg/min) and heating power (3.7 to 7.2 kW). A theoretical investigation of a desalination plant based on this idea using waste heat from a power plant in Cyprus has also been presented by the authors (Maidment et al., 2006, 2007). In this study, they calculated a distilled water production rate of 11.25 million m³

per annum on a basis of 84 MW of heat rejection, inlet saline water at 30 °C and cooling water at 15 °C.

Moore et al. presented another desalination system using NVD, that required continuous pumping of both the saline and distilled water (Moore et al., 2008). The two water columns, with a height of 10 m, which acted as condenser and heater with distilled water and saline water, respectively on either side, were both connected to a vacuum chamber. This setup is illustrated in Fig. 2.7. As the hot and cold vapor meet in the vacuum chamber, they will eventually reach a thermal equilibrium. To overcome this issue, large amounts of water should be pumped continuously on both

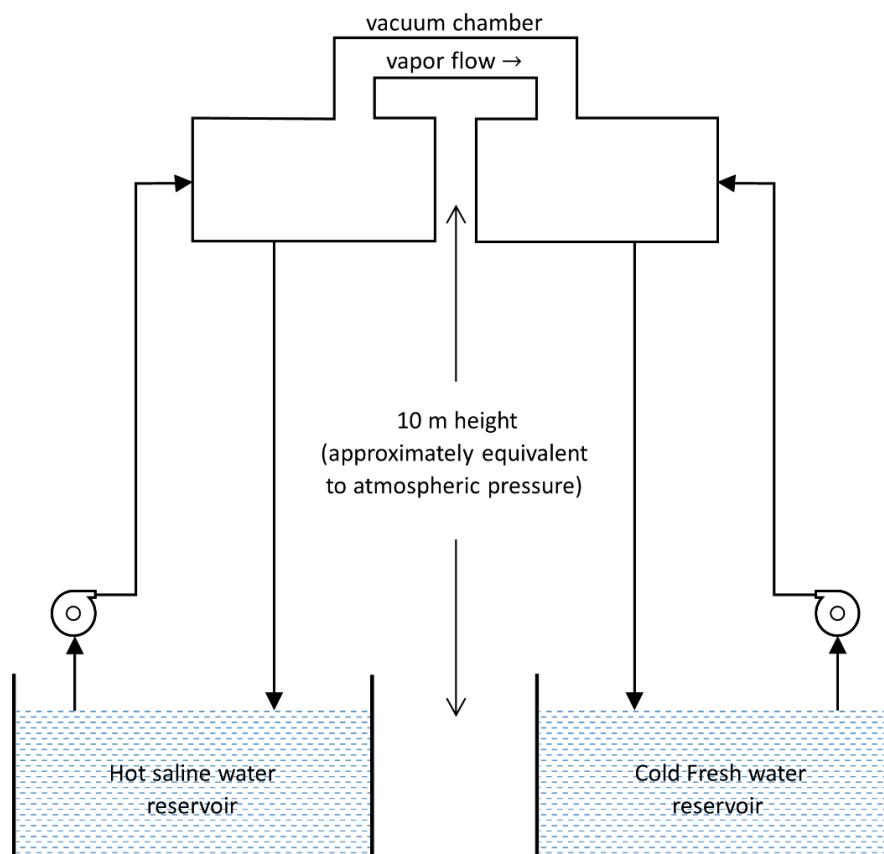


Figure 2.7: Schematic representation of NVD system adapted from Moore et al. (2008)

sides. Calculations done by the authors show that if 100 g/s is pumped on either side, a product rate of 1 g/s is achieved which will cause the saline output to be cooled by 5.4 °C and the distilled water to be heated by 5.4 °C. Huge amounts of water needs to be stored on both sides to ensure that the streams flowing back to the tanks do not change the system conditions. The authors claimed that the system is almost 40% more efficient than RO for a similar production rate. However, this calculation is based only on the pumping power and does not take into account the energy required to maintain the temperature difference between the distilled and saline water reservoirs.

A recent theoretical and experimental study on an NVD desalination system is presented by Ambarita (Ambarita, 2016). The system was similar in design to those that have been previously presented in the literature (see Fig. 2.4 and Fig. 2.5). Experimental runs were carried out for 8-hours duration using an electrical heater. These experiments showed an accumulated distilled water amount of 1.2 L for a heater temperature of 50 °C. The evaporator area was 0.2 m². The author claimed that his numerical investigation shows that an amount of 5.705 L of fresh water is possible for a 9-hour operation time with an evaporation temperature of 80 °C (Ambarita, 2017). One important thing worth mentioning in this study is that the ambient temperatures reported are almost 30 °C and condenser temperature is also in the range of 25-35 °C during the operation.

A theoretical investigation for an NVD desalination system was also presented by Attia (Attia, 2016). He presented a thermodynamic analysis for an NVD system that uses a heat pump to cycle a working fluid to heat the seawater and condense the water vapor after passing through a throttling valve. From the four different working fluids studied, he concluded that refrigerant R-718 was the best option. He calculated a cost

of \$ 0.62/m³ for freshwater production, which is comparable to the popular desalination technologies. The author reported no system design, size, operating conditions or production amount.

2.3. Multi-Stage NVD Systems

In order to increase the efficiency of the NVD system, many researchers have proposed the use of a multi-stage process. A simple representation of a multi-stage configuration is shown in Fig. 2.8. Similar to any conventional multi-effect evaporator, the concentrated liquid (brine) moves from one stage to the other. This will result in an increased overall production rate.

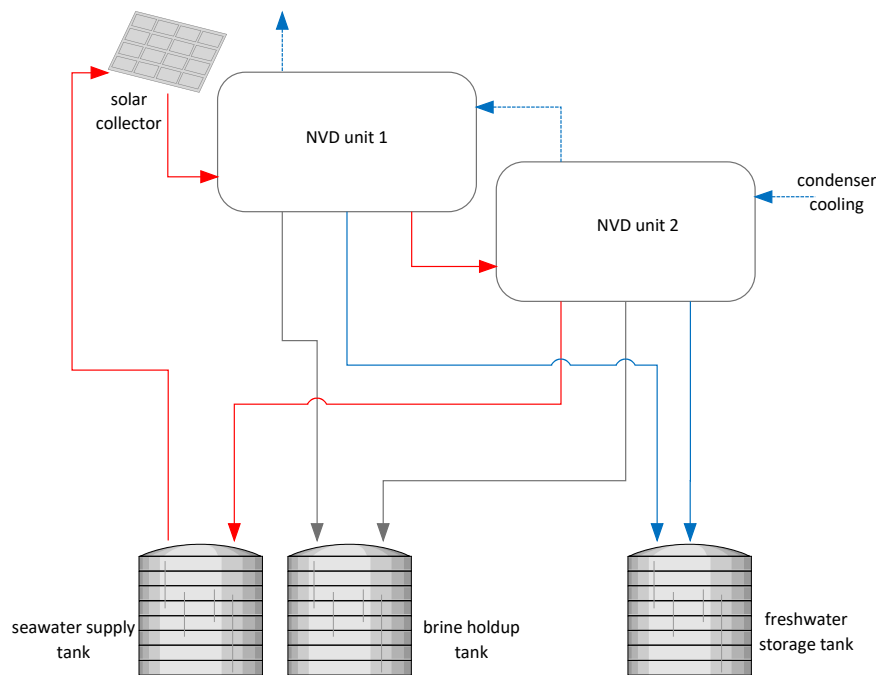


Figure 2.8: Schematic representation of a multistage NVD system

A two-stage NVD system was proposed by Reali in a series of studies wherein he presented different configurations for the process (Reali, 2006, 2007a, 2007b). This was based on a single-stage design presented in an earlier work (Reali et al., 2004). Calculations on a two-stage system showed that the higher the number of stages, the better the performance.

The energy consumption, for a production rate of 36 m³/day with a 10-hour per day operation, was reported as 1.0 kWh/m³, which is less than half of the 2.1 kWh/m³ that was reported in the single-stage design. No prototype plant or experimental study was done to verify the calculated values.

A multi-stage flash (MSF) desalination system based on NVD was also presented by Abutayeh and Goswami (2009, 2010a, 2010b). This was based on recommendations suggested by an earlier study (Al-Kharabsheh & Goswami, 2003). The system was similar to an MSF process. The saline water was heated using a solar heater and then passed on to the NVD flash chamber. Using a multi-stage system allowed the vapor produced in one stage to be used to evaporate part of the water in the next stage. This helped in better energy utilization. Experimental production rate of almost 1.6 kg/h was achieved using saline water feed temperature of 80 °C.

In a later study, the authors carried out theoretical and experimental investigation on the same system (Abutayeh et al., 2013). For a solar photovoltaic collector with a 2.48 m² area and a feed flow rate of 23.3 kg/h, a maximum production rate of 1.72 kg/h was obtained from the simulation of the process. The evaporator had a cross-section area of just under 0.1 m². The system was run for a 3-hour period with an initial pressure of 16 kPa. It is important to note that although the theoretical model was based on steady-state assumption, the experimental results showed that the system was far from exhibiting steady-state behavior after the three-hour test period.

A theoretical study to compare the operation of a single-stage and two-stage configuration for a solar driven NVD flash desalination system was presented by Maroo and Goswami (2009). For a 12-hour operation time, they calculated production amount of 11.31 kg of distillate for a single-stage system, while the two-stage system produced 13.9 kg under

the same initial conditions (an increase of 23% in the production rate). They also studied the use of a solar collector to heat the water. The solar energy driven system had a distillate output of 5.54 kg in 7.83 h for the single-stage system and 8.66 kg in 7.7 h for the two-stage system (an increase of 56% in the production rate). The authors attributed this enhancement of the production rate to the almost negligible pressure increase due to accumulation of non-condensable gases when a solar collector was used instead of the electrical heater.

A similar approach of using multi-stage system to enhance the product rate and performance was also presented by Gude et al. (Gude et al., 2012). Building on their earlier works, they proposed a new two-stage low-temperature phase-change desalination process. The system consists of two evaporation chambers, two horizontal tube condensers (inside tube vapor condensation), one heat exchanger and three barometric columns for each stage. These three columns were the saline water, the brine withdrawal and the distilled water columns, each with its own holding tank maintained under atmospheric pressure. Although the barometric height of water was used to sustain the vacuum in the evaporation chambers, a vacuum pump was employed to remove the non-condensable gases. Saline water was heated using a low-grade heat source and the vapor generated from the first stage was used to heat the feed water in the next stage. The final stage condenser used cold saline water for condensing the incoming vapor from the second stage. A combined production rate of almost 0.35 L/min was achieved from the two stages with an energy consumption of 1500 kJ/kg of distilled water. This study was based on a two-stage process, but the authors mentioned that more stages could be added to improve the performance of the system. They also suggested using a vertical arrangement of the stages to allow lesser land utilization.

The use of the NVD technology allows to generate a low-pressure environment in the headspace of barometric water column with a height of 10.3 m. This allows the evaporation of water at relatively lower temperatures. The lowered boiling point of saline water means a decrease in the thermal energy requirement for the desalination process. Additionally, NVD works without a vacuum pump and does not require pumping power for operation. The low operating temperatures used in NVD systems inhibit the scaling and corrosion problems. One problem that has yet to be resolved is the accumulation of non-condensable gases, which could cause the evaporation process to slow down or completely stop. The issue of the transport of water vapor from the evaporator side to the condenser side also needs to be resolved. In addition, theoretical investigation of the NVD process still needs further improvement because the models used for these studies have been simple.

As a relatively new desalination technique, NVD has shown a lot of promise. For a single-stage setup, experimental freshwater production rate of up to 200 L/day have been obtained (Gude et al., 2012). The production amount can be increased using a multi-stage configuration. Experimental production rates of up to 500 L/day have been reported for such a system (Gude et al., 2011).

2.4. Thermal Energy Storage

An NVD desalination system using renewable energy will need some sort of energy storage to ensure uninterrupted operation of the process. Depending on the type of renewable energy used with the desalination process, different storage methodologies have been outlined by Gude (Gude, 2015) as shown in Fig. 2.9. Desalination processes can be broadly classified into two categories based on the type of energy that drives the process, electrical and thermal. Electrical energy can be obtained from

solar photovoltaic, wind, tidal and geothermal energy. Such systems will require some sort of batteries for electrical energy storage. Thermal desalination systems can be driven by solar thermal energy, solar photovoltaic-based thermal energy, geothermal energy or waste heat.

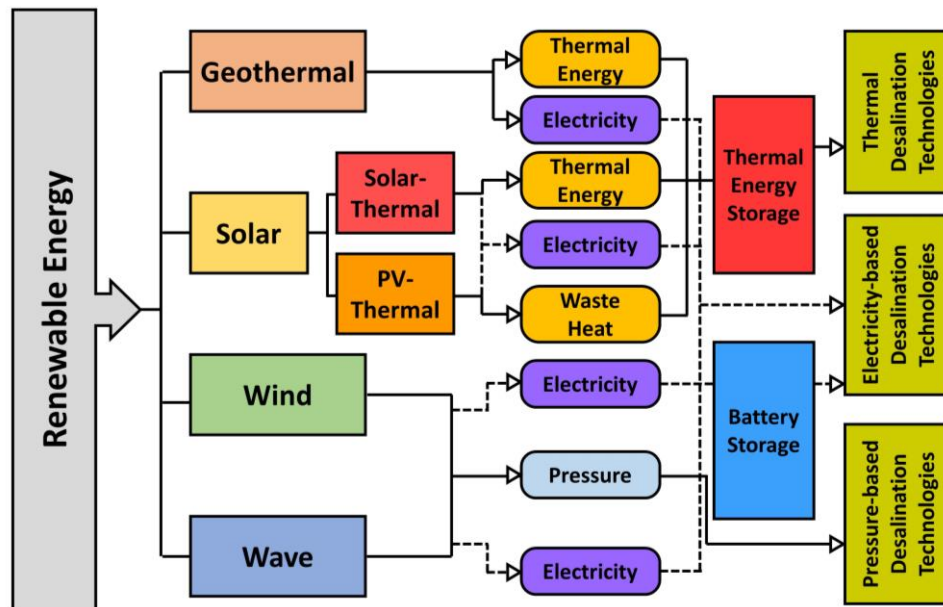


Figure 2.9: Renewable energy and energy storage options for desalination technologies (Gude, 2015)

Thermal solar energy is considered the most suitable renewable energy source option for driving desalination processes that use NVD (Gude et al., 2012). For such cases, thermal energy storage (TES) is required. A TES device transfers heat to storage media during the charging phase, and discharges it to the system as and when required. Sensible, latent and thermo-chemical media store heat in materials that change temperature, phase or chemical composition, respectively. Sensible heat storage is well-documented (Fernandes et al., 2012, Zhang et al., 2016). Latent heat storage uses phase change materials to store the energy. Liquid-solid transition to store latent heat are preferred. They allow a more compact, efficient and therefore economical system to operate (Nomura & Akiyama, 2017). Thermo-chemical heat storage is still at an early stage of

laboratory and pilot research despite its attractive application for long-term energy storage (Aydin et al., 2015). It has a distinct advantage as compared to the other two methods of thermal storage that it allows much higher energy storage with lower heat losses (Kerskes, 2016).

TES systems are of growing importance within the energy awareness: TES can reduce the levelized cost of electricity (LCOE) of renewable energy processes, with the temperature of the storage medium being the most important parameter. A detailed calculation of optimum TES for a desalination plant driven by waste heat is given by Gadhamshetty et al. (2014). The study showed that a 2950 m³ of TES tank was needed to ensure continuous operation of a 950-1600 m³/day MED plant, which uses waste heat from an absorption refrigeration system used to cool a 500 MW power plant. Gude et al. have also studied using TES for NVD desalination systems, as has been discussed earlier in Sec. 2.2.

2.5. Benefits of NVD Systems

The NVD process allows carrying out water desalination at very low pressures. Due to this low-pressure environment, low temperatures can be used for the process. Benefits of such a process are:

- 1) Low Temperature: As could be seen in the previous section, having a low evaporation temperature allows achieving a higher recovery ratio. These low temperatures can be achieved using any low-grade heat source or a renewable energy source.
- 2) Low Energy Costs: Also, as discussed in the previous section, the specific energy requirements for a given recovery ratio are lower at the lower temperatures. A low specific energy requirement means that lesser energy costs are required for processes with low temperatures for evaporation.

- 3) Low Corrosion Rates: Saline water is less corrosive at lower temperatures. These low temperatures thus allow using suitable heat transfer material in the system without much risk of corrosion.
- 4) Low Maintenance: Due to low corrosion rates, maintenance costs will also be reduced. This also allows for an extended life for the process (Kronenberg & Lokiec, 2001).
- 5) Minimal Scaling Rates: Problematic scalants like CaSO_4 have a higher solubility at lower temperatures. Hence, low temperature processes will have minimal or insignificant scaling issues (Ophir & Lokiec, 2005).
- 6) Removal of Toxic Gases: Due to the low-pressure environment, many toxic and harmful gases in the saline water are almost completely removed from the saline water in the desalination process. These non-condensable gases can be vented from the process using an appropriate venting mechanism.
- 7) Reduced Treatment Costs: As the water is subjected to a sudden pressure drop from atmospheric pressure to almost total vacuum, the DNA of most of the living organisms in the water is destroyed thus rendering added safety to the distilled drinking water (Bayer, 1967).

Chapter 3

Theoretical Analysis

Table of Contents

Subject	Page No.
Chapter Three Theoretical Analysis	36
3.1 The proposed NVD system	37
3.2 Mathematical Model	38
3.2.1 Lumped Parameter Model	38
3.2.2 Distributed Model	41
3.3 Modeling with COMSOL Multiphysics ®	46
3.3.1 Overview of COMSOL Multiphysics ®	46
3.3.2 NVD modeling in COMSOL Multiphysics ®	49
3.4 Modeling of the Parabolic Solar Collector	54
3.4.1 Geometrical Aspects of a PTC	57
3.4.2 Thermal Analysis of a PTC	59

Chapter 3: Theoretical Analysis

This chapter discusses the theoretical analysis of the proposed seawater desalination system proposed to operate on the NVD principle. The modeling of the solar collector to be used with the system is also presented.

3.1. The Proposed NVD System

The proposed system is designed as an inverted conical shape with a cylindrical base. The hot seawater is fed into the center of the evacuated chamber where it flashes. The vapor then rises to meet the cold conical roof and then under the influence of the gravity, falls along the sloping walls into the outer cylindrical portion of the base where freshwater is collected.

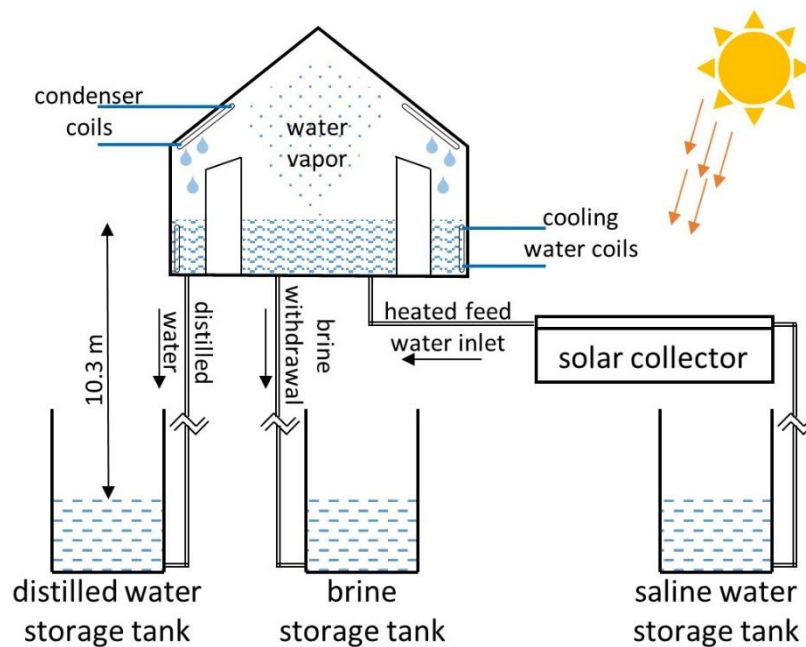


Figure 3.1: Schematic representation of the proposed NVD system

The roof is cooled by using cold seawater. This geometry allows solving the problem of vapor transfer from the evaporator to the condenser side. Additionally, the configuration allows the heat to be transferred from the solar collector outlet line to the feed water to conserve more energy. The system also includes a condenser using cold seawater, pumped using solar

power, to make the system feasible in the hot climate regions. A schematic of this configuration is shown in Fig 3.1.

3.2. Mathematical Model

The main component of an NVD system is the evaporation chamber. This is the chamber which is under the “natural vacuum” and evaporation takes place here as a result of a heat source. This heat source may add thermal energy by direct heating of the water in the evaporation chamber or may be used to pre-heat the water before it enters the evaporation chamber. Assuming that the entire vapor produced in the evaporation chamber condenses, and that it does so entirely on to the freshwater chamber side, it can be safely stated that the evaporation chamber is the most essential component of such a system. Hence, the mathematical modeling discussed below is only for the evaporation chamber in an NVD system.

Two modeling techniques were used with the NVD process. The first is a transient model, which describes the system behavior with time. This is a lumped parameter model. The second model is a more detailed one that allows spatial analysis of the process along with its dynamic behavior. Both models are discussed below.

3.2.1. Lumped Parameter Model

A complete mathematical description of an NVD process requires equations for mass, energy and solute balances. In this model, it was assumed that no temperature stratification occurs in the system and the heat capacity of the evaporator material is neglected. Such assumption of a uniform temperature requires that either the system has some sort of mixing technique employed or the system is of a small size. The assumptions may not be valid if there is no mixing and the volume of the water under study

is large. Neglecting the heat capacity of the evaporator material will cause slight errors in the heat balance. A simplified flowchart of the process is shown in Fig. 3.2.

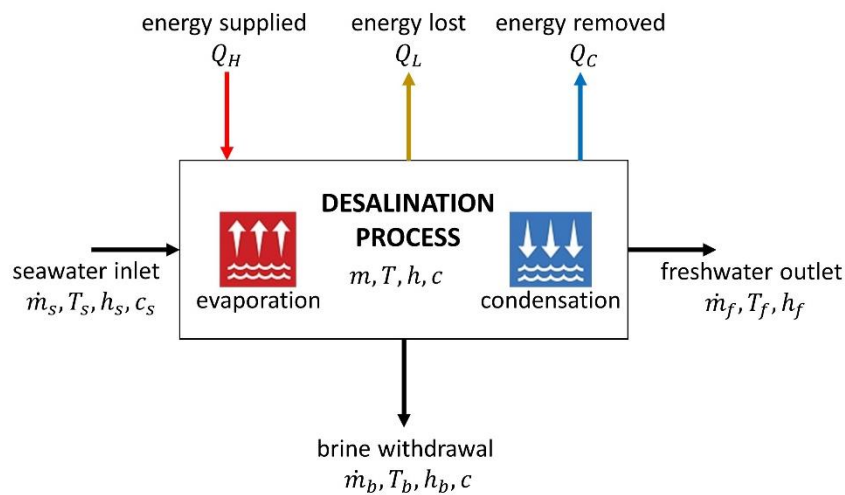


Figure 3.2: The NVD process with the inlet/outlet streams

Referring to Fig. 3.2, the balances are written as:

$$\frac{dm}{dt} = \dot{m}_s - \dot{m}_b - \dot{m}_f \quad (3.1)$$

$$\frac{dh}{dt} = \dot{m}_s h_s - \dot{m}_b h_b - \dot{m}_f h_{fg} + Q_H - Q_L \quad (3.2)$$

$$\frac{d(mc)}{dt} = \dot{m}_s c_s - \dot{m}_b c_b \quad (3.3)$$

These three equations are the mass, energy and solute balance, respectively. $\dot{m}$ refers to the mass flow rate, h to the enthalpy and c to the concentration. The subscripts s , b and f refer to the saline water, brine and freshwater, respectively. Q_H and Q_L are the heat added to and lost from the system. The system properties (temperature, density, concentration, etc.)

are assumed uniform in the evaporation chamber. Hence, the brine concentration and temperature will be the same as the average values in the evaporation chamber.

Many equations have been presented in the literature for calculating the evaporation rate of water. Most of these are for evaporation of water into air or evaporation of water into its own vapor. The most suitable equation for predicting the evaporation rate of water in an NVD system was provided by Bemporad (1995). For a detailed discussion of the evaporation rate equations, see Appendix A1. Bemporad's equation is:

$$\dot{m}_f = A_c \times \alpha_m \times \left(f(C_s) \frac{p(T_s)}{\sqrt{T_s + 273}} - \frac{p(T_f)}{\sqrt{T_f + 273}} \right) \quad (3.4)$$

where

A_c is the cross-sectional area of the evaporation section,

α_m is the mass accommodation factor,

$f(C_s)$ is a correction factor for salt concentration,

T_s is the temperature of the salt-side (i.e. evaporation section temperature),

T_f is the temperature of the cooled freshwater side,

$p(T_s)$ and $p(T_f)$ are the saturation pressures of water at the salt water and freshwater temperatures, respectively

The solution of these equations will require discretization of Eq. 3.1 to 3.3. A backward finite difference scheme can be used with Euler's method to solve the equations simultaneously. A code was developed in MATLAB® to solve these equations. This simplified model has been used by all researchers in the earlier studies reported in the literature. It will give

a general idea of the process behavior, product rate and temporal distribution of temperature and concentration in the evaporation chamber. However, since it is a lumped parameter model, the temperature and other properties are assumed constant throughout the chamber. This affects the calculation of the evaporation rate, as we know that there will be some temperature gradient in the evaporation chamber with height. Hence, the surface of the water, where evaporation takes place, will not be the same as the inlet temperature.

3.2.2. Distributed Model

The lumped parameter model developed in the previous section does not allow for spatial analysis of the system behavior. A more comprehensive model is discussed in this section. As stated earlier, the mathematical modeling of the evaporation chamber will require equations for mass, energy and salt balances. These equations are written in the general form as follows (Bird et al., 2002):

$$\frac{\partial \rho}{\partial t} + \nabla \cdot (\rho \mathbf{u}) = 0 \quad (3.5)$$

$$\rho \frac{\partial \mathbf{u}}{\partial t} + \rho (\mathbf{u} \cdot \nabla) \mathbf{u} = \nabla \cdot [-P\mathbf{I} + \boldsymbol{\tau}] + \mathbf{F} \quad (3.6)$$

$$\begin{aligned} \rho C_p \left(\frac{\partial T}{\partial t} + (\mathbf{u} \cdot \nabla) T \right) \\ = -(\nabla \cdot \mathbf{q}) + \boldsymbol{\tau} : \mathbf{S} - \frac{T}{\rho} \frac{\partial \rho}{\partial T} \bigg|_p \left(\frac{\partial \rho}{\partial t} + (\mathbf{u} \cdot \nabla) \rho \right) + \mathbf{Q} \end{aligned} \quad (3.7)$$

where

ρ is the density (kg/m³)

$\mathbf{u}$ is the velocity vector (m/s)

P is pressure (Pa)

$\boldsymbol{\tau}$ is the viscous stress tensor (Pa), which for non-Newtonian fluids like water is given by:

$$\boldsymbol{\tau} = 2\mu\mathbf{S} - \frac{2}{3}\mu(\nabla \cdot \mathbf{u})\mathbf{I} \quad (3.8)$$

$\mathbf{F}$ is the volume force vector (N/m³)

C_p is the specific heat capacity at constant pressure (J/kg·K)

T is the absolute temperature (K)

$\mathbf{q}$ is the heat flux vector (W/m²)

$\mathbf{Q}$ contains the heat sources (W/m³)

$\mathbf{I}$ is the identity matrix

μ is the viscosity (Pa.s)

$\mathbf{S}$ is the strain-rate tensor given by:

$$\mathbf{S} = \frac{1}{2}(\nabla \mathbf{u} + (\nabla \mathbf{u})^T) \quad (3.9)$$

The operation “ $\cdot$ ” (also sometimes referred to as the double dot product) denotes a contraction between tensors defined by:

$$\mathbf{a} : \mathbf{b} = \sum_n \sum_m a_{nm} b_{mn} \quad (3.10)$$

These equations are the mathematical representations of the conservation laws. Eq. 3.5 is the continuity equation and is a mathematical representation of the conservation of mass. Eq. 3.6 is a vector equation, which represents the conservation of momentum. The first term on the left hand side of the equation is the accumulation term, followed by the convective transport of momentum term. On the right hand side is the net pressure acting on the fluid and the diffusive transport of momentum. The last term in the momentum equation is the body force. Eq. 3.7 describes the conservation of energy, formulated in terms of temperature. The first term in this equation is the accumulation of energy and the transport of energy by convection. On the right hand side, the first term is the heat flux due to conduction, followed by the viscous dissipation of energy and the work done due to pressure acting on the fluid element. The last term is the sum of all heat sources and sinks.

The concentration of salt in the evaporator can be investigated using the species conservation equation:

$$\frac{\partial c_i}{\partial t} + \nabla \cdot (-D \nabla c_i) + \mathbf{u} \cdot \nabla c_i = R_i \quad (3.11)$$

where

c_i is the molar concentration of the species i (mol/m³)

D_i denotes the diffusion coefficient for species i (m²/s)

R_i is a reaction rate expression for the species i (mol/m³.s)

The first term on the left hand of Eq. 3.11 is the accumulation term, while the second term accounts for the diffusive transport of the solute species in the solvent. The third term on the left hand side represents the convective transport of the species i due to the velocity vector, $\mathbf{u}$. The term

on the right hand side of the equation is the sink term (typically due to a reaction or desorption of species i), which will be zero in this case.

Evaporation rates in an NVD system are given by the equation presented by Bemporad (Bemporad, 1995):

$$\dot{m}_{ev} = A_c \times \alpha_m \times \left(f(C_s) \frac{p(T_s)}{\sqrt{T_s + 273}} - \frac{p(T_f)}{\sqrt{T_f + 273}} \right) \quad (3.12)$$

where

A_c is the cross-sectional area of the evaporation section,

α_m is the mass accommodation factor, $\text{kg.m}^{-2}.\text{Pa}^{-1}.\text{s}^{-1}.\text{K}^{0.5}$

$f(C_s)$ is a correction factor for salt concentration,

T_s is the temperature of the salt-side (i.e. evaporation section temperature),

T_f is the temperature of the cooled freshwater side,

$p(T_s)$ and $p(T_f)$ are the saturation pressures of water at the salt water and freshwater temperatures, respectively

It is important to couple the evaporation rate equation and the balance equations because the evaporation rate has an impact on the mass balance as well as the surface temperature. The thermophysical properties of seawater have been found using the correlations provided by Sharqawy et al. (2010) and Nayar et al. (2016).

Density of seawater (kg/m^3) is given as a function of temperature, T ($^{\circ}\text{C}$) and salinity, S (g/kg) by the following relation:

$$\rho_{sw} = (a_1 + a_2T + a_3T^2 + a_4T^3 + a_5T^4) + (b_1S + b_2ST + b_3ST^2 + b_4ST^3 + b_5S^2T^2) \quad (3.13)$$

where $a_1 = 9.999 \times 10^2$, $a_2 = 2.034 \times 10^{-2}$, $a_3 = -6.162 \times 10^{-3}$, $a_4 = 2.261 \times 10^{-5}$, $a_5 = -4.657 \times 10^{-8}$, $b_1 = 8.020 \times 10^2$, $b_2 = -2.001$, $b_3 = 1.677 \times 10^{-2}$, $b_4 = -3.060 \times 10^{-5}$, $b_5 = -1.613 \times 10^{-5}$.

Saturation Pressure of seawater (in Pa) was similarly calculated from the following relation:

$$\frac{p_{v,w}}{p_{v,sw}} = 1 + 0.57357 \left(\frac{S}{1000 - S} \right) \quad (3.14)$$

where $p_{v,w}$ is given by:

$$\ln p_{v,w} = \frac{a_1}{T} + a_2 + a_3T + a_4T^2 + a_5T^3 + a_6 \ln T \quad (3.15)$$

where $a_1 = -5.800206 \times 10^3$, $a_2 = 1.3914993$, $a_3 = -4.8640239 \times 10^{-2}$, $a_4 = 4.1764768 \times 10^{-5}$, $a_5 = -1.4452093 \times 10^{-8}$, $a_6 = 6.5459673$.

Enthalpy (J/kg) of seawater, h_{sw} is given by:

$$h_{sw} = h_w - S(a_1 + a_2S + a_3S^2 + a_4S^3 + a_5T + a_6T^2 + a_7T^3 + a_8ST + a_9S^2T + a_{10}ST^2) \quad (3.16)$$

where h_w is given by:

$$h_w = 141.355 + 4202.07T - 0.535T^2 + 0.004T^3 \quad (3.17)$$

and $a_1 = -2.348 \times 10^4$, $a_2 = 3.152 \times 10^5$, $a_3 = 2.803 \times 10^6$, $a_4 = -1.446 \times 10^7$,

$a_5 = 7.826 \times 10^3$, $a_6 = -44.17$, $a_7 = 2.139 \times 10^{-1}$, $a_8 = -1.991 \times 10^4$, $a_9 = 2.778 \times 10^4$, $a_{10} = 9.728 \times 10^1$.

The latent heat of vaporization (J/kg) of seawater is given by:

$$h_{fg,sw} = h_{fg,w} \times (1 - S/1000) \quad (3.18)$$

where $h_{fg,w}$ is given by:

$$h_{fg,w} = a_1 + a_2T + a_3T^2 + a_4T^3 + a_5T^4 \quad (3.19)$$

where $a_1 = 2.501 \times 10^6$, $a_2 = -2.369 \times 10^3$, $a_3 = 2.678 \times 10^{-1}$, $a_4 = -8.103 \times 10^{-3}$, $a_5 = -2.080 \times 10^{-5}$.

3.3. Modeling with COMSOL Multiphysics®

3.3.1. Overview of COMSOL Multiphysics®

Computer simulation has become an indispensable part of research in engineering today. Computer analysis of components, in particular, is very important when new products are developed or optimization of existing designs is required. Many different options are available to researchers nowadays for simulation. When considering what option is the best for any given problem, it is very important to consider that the modeling software is able to accurately depict what happens in the real world. A computer simulation environment is simply a translation of real-world physical laws into their virtual form. How much simplification takes place in the translation process helps to determine the accuracy of the resulting model.

A simulation environment that allows the possibility to add any physical effect to a given model should thus be the most ideal. That is what COMSOL is all about. COMSOL Multiphysics® is a general-purpose

software platform, based on advanced numerical methods, for modeling and simulating physics-based problems (COMSOL Inc., 2016). It comes with many physics modules that are of great help in modeling and simulation. A quick look at the different modules offered by COMSOL is given in Fig. 3.3.

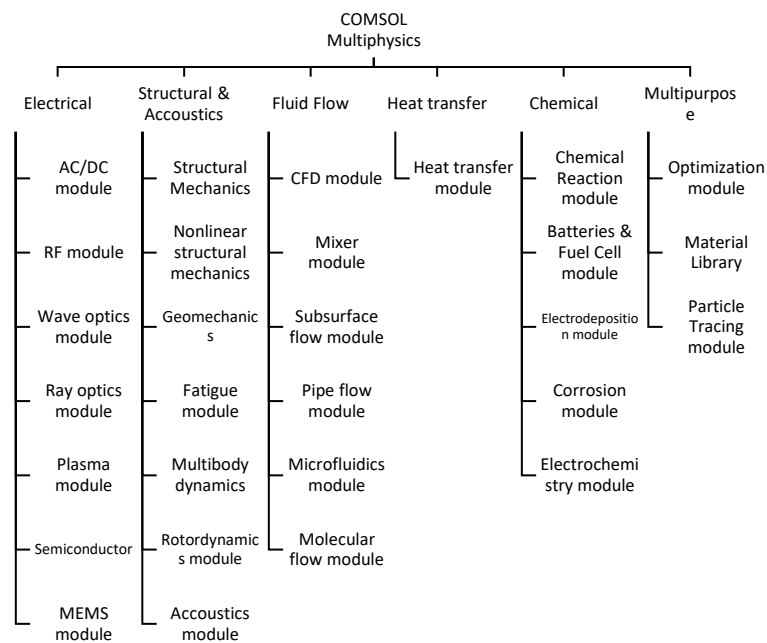


Figure 3.3: Different physics modules in COMSOL®

With this kind of all-inclusive modeling environment, COMSOL gives the confidence to build a model with real-world precision. These modules can be coupled and/or solved separately for steady state or transient solutions using a variety of built-in solving techniques. COMSOL Multiphysics also comes with a built-in library of materials with pre-defined properties. It also allows a great deal of flexibility in both the material properties and the physics modules. Not only does COMSOL allow all these different physics modules to be linked to each other, it also has options to import models from other leading software like AutoCAD, Solidworks, CATIA, Revit, Inventor and PTC Pro etc. and also allows

linking with MATLAB and Excel. It allows to arbitrarily include user-defined equations that may describe a material property, boundary, and source or sink term, or even a unique set of partial differential equations (PDEs). New physics interfaces may be created from these equations or they could simply be coupled with existing physics. In brief, COMSOL Multiphysics has a very user-friendly and powerful Graphical User Interface, which makes the modeling and “what if” scenarios very easy without investing a lot of programming time (Zimmerman, 2006).

COMSOL Multiphysics uses the Finite Element Method (FEM) to solve the partial differential equations (PDE) that define the various physics. COMSOL offers automatic and semi-automatic meshing tools. The finite element mesh serves two purposes. It first subdivides the CAD geometry being modeled into smaller pieces, or elements, over which it is possible to write a set of equations describing the solution to the governing equation. The mesh is also used to represent the solution field to the physics being solved. For the two-dimensional models, triangular and quadrilateral elements are used. There are four different element types for the three-dimensional problems – tetrahedral, bricks, prisms and pyramids, as seen in Fig. 3.4.

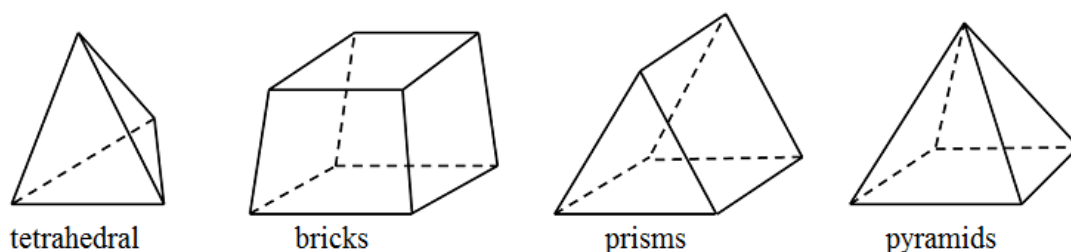


Figure 3.4: Different 3D meshing element types

These four elements can be used, in various combinations, to mesh any 3D model. By default, automatic tetrahedral meshing is used for physics defined in solids, and a combination of tetrahedral and boundary-

layer meshing for fluids. Tetrahedral elements are also known as a simplex, which simply means that any 3D volume, regardless of shape or topology, can be meshed with these elements. The other three element types (bricks, prisms and pyramids) should be used only when it is motivated to do so. The primary motivation to use bricks and prisms is that they can reduce the number of elements used for meshing. Pyramids are generally used to bridge different shaped elements. It should be noted that these elements will not always be able to mesh a particular geometry. However, the user may take full control over the sequence of operations used for creating the mesh by defining a “mesh sequence”.

3.3.2. NVD modeling in COMSOL Multiphysics®

Modeling of any problem in COMSOL Multiphysics starts with the definition of the geometry. Depending on the type of study required, a one-dimensional, two-dimensional or a three-dimensional geometry may be defined. COMSOL offers very adept drawing tools to define the geometry with all essential features. Once the component is drawn in the required geometry, a material can be attached to it. The most common material required for NVD modeling is water, which is already defined as a built-in material with the common properties. Since these properties are for pure water, saline water properties can be defined as functions of temperature and concentration using Equations 3.13 to 3.19. COMSOL allows changes to any pre-defined material properties. If it is desired to include the walls of the evaporator into the study, appropriate material can also be added to the walls. However, this will only add to the computation time and will not make a significant impact on the study.

Next comes the attachment of the physics modules to the component. The two modules that were used in the modeling and simulation of the NVD system are the Computational Fluid Dynamics (CFD) module and

the Heat-Transfer module. Each module allows different boundary conditions to be defined for the component. A laminar flow model can be used for the CFD physics because the velocities in the evaporator are low and not much turbulence is expected. For the CFD module, the initial conditions need to be defined for the velocity field as well as the pressure. The velocity can be entered as zero for all the three components of the velocity vector. For the pressure the pressure can be defined as $p = p_0 + \rho g \Delta h$ where Δh is the depth from the top surface of the fluid and p_0 is the pressure at the surface. A “pressure point constraint” to set the pressure to p_0 at any point on the surface helps the solution to converge and avoids the divergence problem that may arise due to the infinite number of solutions while solving the momentum equation.

For the boundaries of the evaporator across which there is no mass transfer, a “no slip” wall should be added. This helps the solver to know that the velocity is zero at these boundaries. Inlet of the saline water and the outlet of the brine can be defined using the “inlet” and “outlet” boundary conditions. If it is desired to define mass flow rates at the inlet and outlet boundaries, it is preferred to use “inlet” boundary condition with negative flows for the outlets. COMSOL allows defining mass flow rates in the “inlet” boundary conditions. This option sets the integrated normal mass flow across the entire boundary to the defined value. The mass flow is assumed normal to the boundary surface, and the tangential flow velocity is set to zero. A similar mass outlet condition can be defined for the top surface of the water in the evaporator. Generally, the brine flow rate, $\dot{m}_b$, is decided upon and the evaporation rate can be calculated using Eq. 3.12. A volume force should be added to the domain that contains water to account for the buoyancy force, ($\mathbf{F}$ in Eq. 3.6). COMSOL allows three approaches for defining the volume force:

1. Using the full Navier-Stokes equation where the volume force is defined as $\mathbf{F} = \rho \mathbf{g}$
2. Using the Navier-Stokes equation a pressure shift, the volume force is defined as $\mathbf{F} = (\rho - \rho_0) \mathbf{g}$, where ρ_0 is the water density defined at a reference temperature and pressure.
3. Using the Boussinesq approximation (Schmitt, 2007), where the volume force is defined as $\mathbf{F} = -\rho_0 \left(\frac{T - T_0}{T_0} \right) \mathbf{g}$. The term ρ_0 is the reference density defined at the reference temperature, T_0 and the reference pressure. The Boussinesq approximation states that the density is constant except in the buoyancy term.

For better accuracy, it is advised to use the first approach, which considers the full Navier-Stokes equation.

In the Heat Transfer module, the “heat transfer in solids” and “heat transfer in fluids” conditions should be added to the solid and fluid domains, respectively. COMSOL will use the thermophysical properties from the materials defined earlier. As with the CFD module, initial conditions should be specified for the temperature. The ambient temperature can be used as the initial condition. The inlet temperature should also be defined at the seawater inlet. “Thermal insulation” boundary condition should be added to all walls that are insulated and no heat transfer is expected over them. For the walls that could have heat losses, “heat flux” condition should be used. COMSOL allows to add a specific value for the heat flux (q in Eq. 3.10) or a convective heat loss based on the external temperature, T_{ext} , given by $q = h(T_{ext} - T)$ where h is the convective heat transfer coefficient. This heat transfer coefficient can be defined as a constant value or can be calculated by COMSOL using the built-in options of forced or natural convection. A heat flux boundary should also be added

to the top surface of the water where heat is lost due to the latent heat of evaporation. This is calculated using Eq. 3.18 and 3.19.

COMSOL uses the finite element method (FEM) to compute the numerical solutions of the partial differential equations (PDEs) that are used in the modeling of such systems. Discretization techniques using polynomial approximations of different orders are available for use in COMSOL. Since the flow is expected to be laminar, a first-order upwind discretization is acceptable. Higher order polynomials for discretization of the velocity and pressure may be used for more accuracy, but this will add to the time required for solving the problem. Also, linear discretization has a better convergence than the higher order schemes. For laminar flow, the numerical diffusion will be naturally low, hence first-order discretization can be used without any worry of significant errors. To avoid convergence errors, it is advisable to use the same discretization schemes for all physics. Hence, the heat transfer module should also be set to use the linear discretization scheme. COMSOL also allows a very versatile meshing technique. The mesh can be built by the user with full control over the shape and size of the elements. However, it is preferred to use the built-in meshing techniques, which are designed for best results for the different physics. The “physics-controlled mesh” for a 3D problem in CFD is based on tetrahedral elements. Since a volume force is used in the problem, it is better to use a fine mesh because course meshes usually do not work well with buoyancy.

For the concentration distribution in the evaporator, the built-in module for the transport of species caused convergence problem. To overcome the issue, a lumped model can be used to determine the concentration of the seawater in the evaporator. The following species material balance equation can be written:

$$\frac{d}{dt}(\rho c V) = (\rho c)_{in} \dot{V}_{in} - (\rho c)_s \dot{V}_b \quad (3.20)$$

where the subscript *in* refers to the inlet, *s* to the saline and *b* the brine withdrawal, $\dot{V}$ is the volumetric flowrate of the mentioned flows. Salt does not leave the surface where evaporation takes place. This assumption is a reasonable one, because the concentration is not expected to change by much in the entire system. The maximum expected concentration is at the surface, which will not differ by more than 3% from the inlet concentration. As noted by earlier researchers (Al-Kharabsheh & Goswami, 2004, Gude & Nirmalakhandan, 2008), and also reported by Keren et al. (1993), the evaporation rate drops by 1% if the salt concentration increases by 1%. Hence, not much effect of the concentration is expected on the evaporation rate calculations if a lumped model is assumed. This can be seen from the results of a 2D model shown in Fig. 3.5 and 3.6 below.

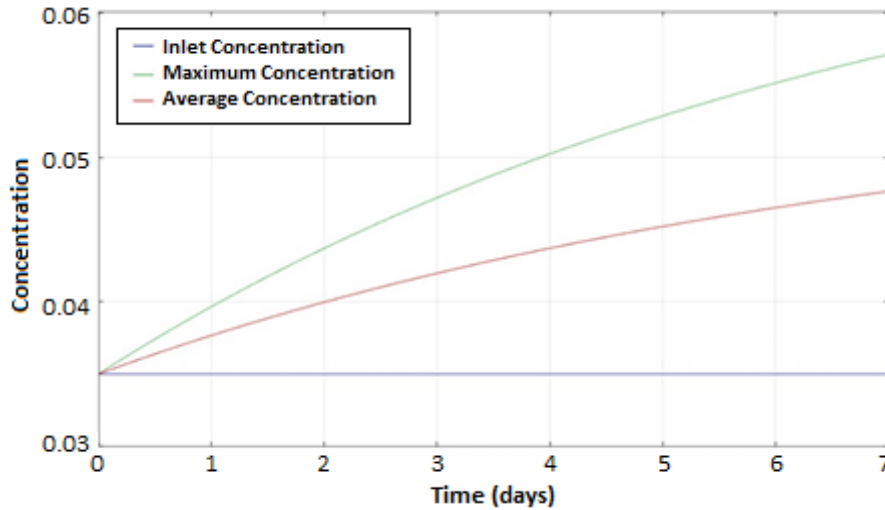


Figure 3.5: Concentration change with time in the evaporation chamber

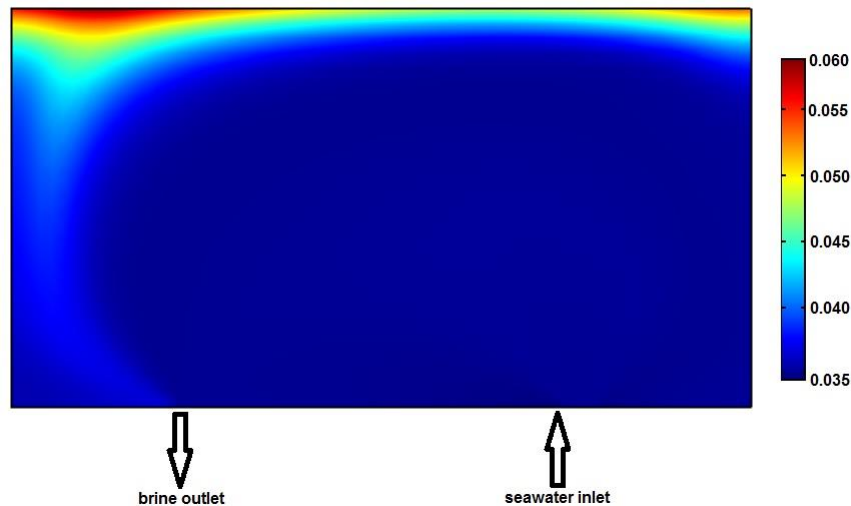


Figure 3.6: Concentration distribution in the evaporation chamber

It can be seen from Fig. 3.5 that the maximum concentration (at the surface of the water), reaches around 6% in 7 days. The average concentration at the evaporator is just below 5%. Hence, if this average value is used, the error in the calculation of the evaporation rate will be very small. It can also be seen from Fig. 3.6 that the major portion of the evaporator has almost a uniform concentration. A slight increase in the concentration can be seen near the top surface, due to the evaporation.

3.4. Modeling of the Parabolic Solar Collector

Solar energy is the most readily available renewable energy source. Due to the increasing demand of energy and the environmental hazards associated with the combustion of fossil fuels, solar energy is gaining more and more attention as an alternative energy source (Delyannis & El-Nashar, 2010). A major advantage of solar energy is that it is environmentally clean (Kalogirou, 2004). It can be utilized through conversion to electricity by means of photovoltaics (PV) or directly as solar thermal energy (STE). A recent review of the different solar energy technologies has shown that there is no clear advantage of PV over STE or vice versa (Chu & Meisen, 2011). However, the study did mention that STE is more predictable and

allows cheaper and easier energy storage than PV. Solar thermal energy is considered more economic and has found more acceptance in the industrial use (Mekhilef et al., 2011). Solar thermal energy is harnessed by using solar collectors, which convert the energy from incident solar radiation to useful heat energy.

Solar collectors can be classified into two main categories: non-concentrating and concentrating collectors. Non-concentrating collectors have a concentration ratio (CR) of one or less, while the concentrating types have much higher values. CR is the ratio of the absorber area to the intercepting area of the collector. Another advantage of the concentrating collectors is that they can be designed for solar tracking.

Several designs for the concentrating collectors are available. They include the cylindrical trough collectors (CTC), parabolic trough collectors (PTC), compound parabolic collectors (CPC), heliostat field collectors (HFC), parabolic dish collectors (PDC) and the linear fresnel collectors (LFC). They can be designed for single-axis solar tracking, like the CTC, PTC, CPC and LFC, which have a CR values up to 85 or for two-axis tracking, like the PDC and the HFC, which have CR values of up to 2000 (Kalogirou, 2013). Among the single-axis tracking, concentrating collectors, the PTC has the highest CR values and is simple in design. It is a high-performing solar collector with high heat efficiency for operating temperatures of up to 400 °C (Jebasingh & Joselin Herbert, 2016).

A PTC has the geometrical benefit of its shape that allows it to achieve high concentrating ratios. By the virtue of its parabolic shape, a PTC focuses direct solar radiation onto its focal line. A receiver tube, made from a high thermal conductivity material and with a working fluid flowing inside, is installed in this focal line. The fluid absorbs the thermal energy from the concentrated radiation and raises its enthalpy. The PTC can easily

be set up for single-axis solar tracking to ensure that the solar beams intercepted by the collector are always normal to the collector surface. A PTC makes use only of direct solar radiation, called beam radiation or Direct Normal Irradiance (DNI). This is the fraction of solar radiation that reaches the Earth's surface as a parallel beam and is not diverted by the presence of clouds or other particles in the air (Fernandez-Garcia et al., 2010). Among the renewable energy sources, solar energy is the most popular because it is abundant, free and clean. Solar collectors are special types of heat exchangers that convert the solar radiation energy to internal energy of a transport medium. Different types of solar collectors are available which are shown in Fig. 3.7 (Kalogirou, 2013).

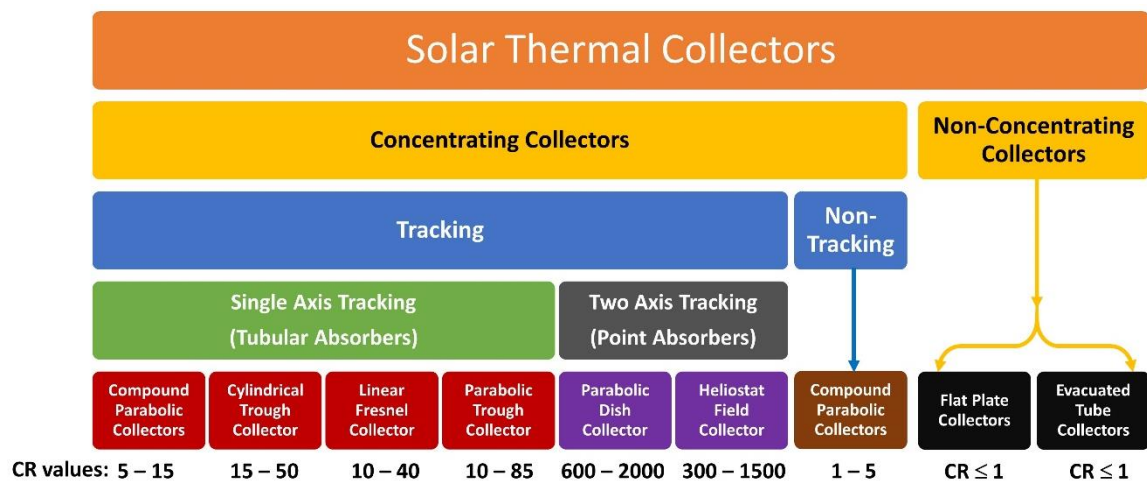


Figure 3.7: Types of solar collectors

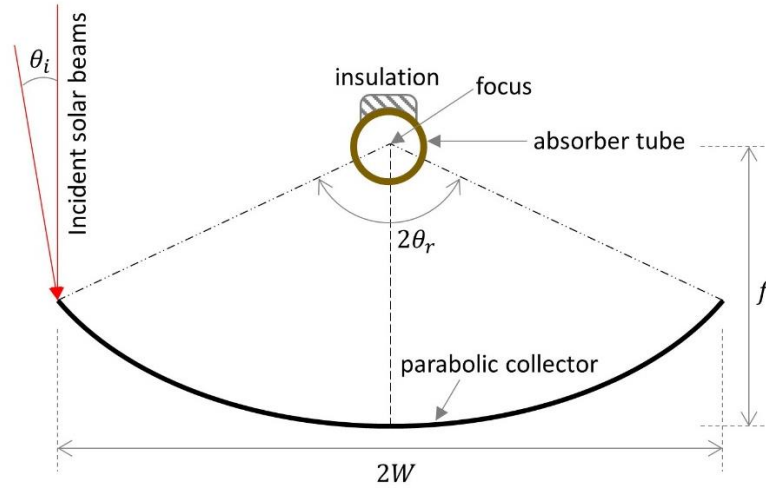


Figure 3.8: Geometrical aspects of a parabolic trough collector

3.4.1. Geometrical Aspects of a PTC

The Parabolic Trough Collector (PTC) has been selected because it is the simplest of the concentrating collectors and can be used effectively with a single axis solar tracking. The geometrical aspects of a PTC are illustrated in Fig. 3.8. The most important aspect of a PTC is the concentration ratio (CR), which is defined as the ratio between the effective aperture area and the absorber tube area. A relation between the CR, the rim angle (θ_r) and the solar beam incident angle (θ_i) is given as (Kandlikar & Vij, 1978):

$$CR = \frac{1 - \sin \theta_i / \sin \theta_r}{\frac{\sin \theta_i}{\sin \theta_r} \left(\pi \frac{\theta_r + 90 - \theta_i}{180} \right)} \quad (3.21)$$

According to this equation, it can be seen that the concentration ratio has the highest value at a rim angle of about 70° for all incident angles (see Fig. 3.9). Hence, this rim angle was selected for the PTC to be used in the setup. The width (W) of the PTC was chosen to be 0.8 m. The focal length, f , of the PTC is calculated as (Macedo-Valencia et al., 2014):

$$f = \frac{W}{4 \tan(\theta_r/2)} \quad (3.22)$$

For a width of 0.8 m and a rim angle of 70°, this is calculated to be 0.286 m. The equation of the parabola is then given by :

$$y = \frac{x^2}{4f} \quad (3.23)$$

where the x values range from -0.4 m to +0.4 m. The minimum radius (R_{min}) for the absorber tube, which will be able to intercept all the rays focused to the focal point, for a given solar beam incident angle and rim angle is given by the equation (Macedo-Valencia et al., 2014):

$$R_{min} = \frac{2f \sin \theta_i}{1 + \cos \theta_r} \quad (3.24)$$

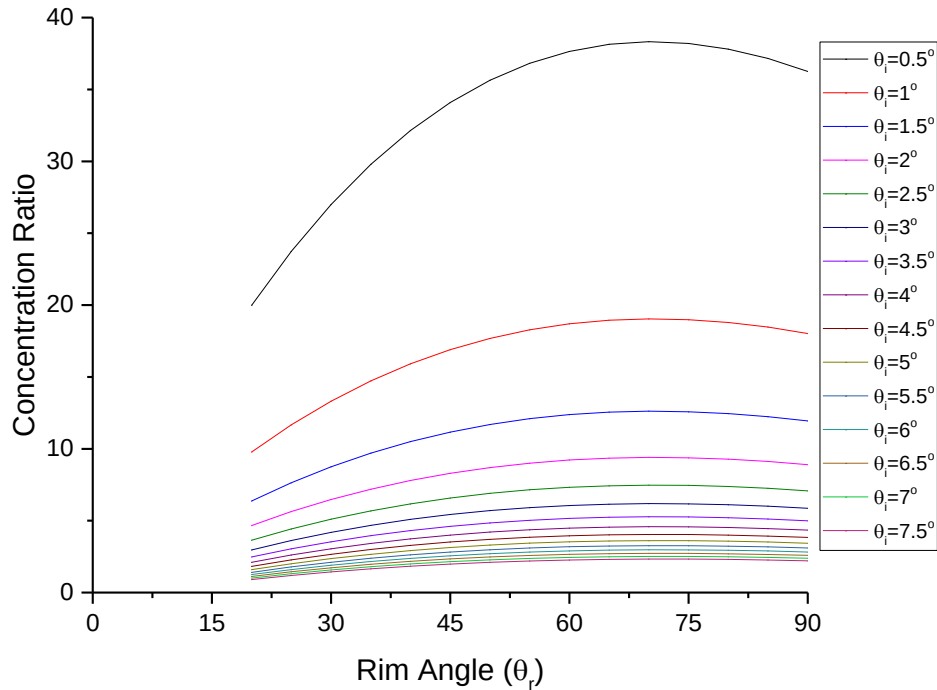


Figure 3.9: Concentration ratio as a function of rim angle for different incident angles

Simonovic et al. (2016) suggested that the incident angle for the sun's beams should be between 0.835° and 1.219° , otherwise the resulting calculations would be erroneous. For an incident angle of 1° and a rim angle of 70° , this is calculated as 0.0037 m (i.e. 3.7 mm) or a diameter of 7.4 mm. The selected absorber tube diameter was, therefore, 12.7 mm (which is the inside diameter for a $\frac{1}{2}$ " nominal size copper tube). The absorber tube can be shielded by an evacuated glass envelope to minimize convection heat losses from the absorber tube to the environment. The thickness of this glass envelope tube should be enough to withstand the atmospheric pressure, and at the same time, it should not be thick to cause optical losses of the incident solar radiation.

3.4.2. Thermal Analysis of a PTC

The energy components in the absorber tube are shown in Fig. 3.10. The red arrows indicate the solar energy absorbed by the glass tube and

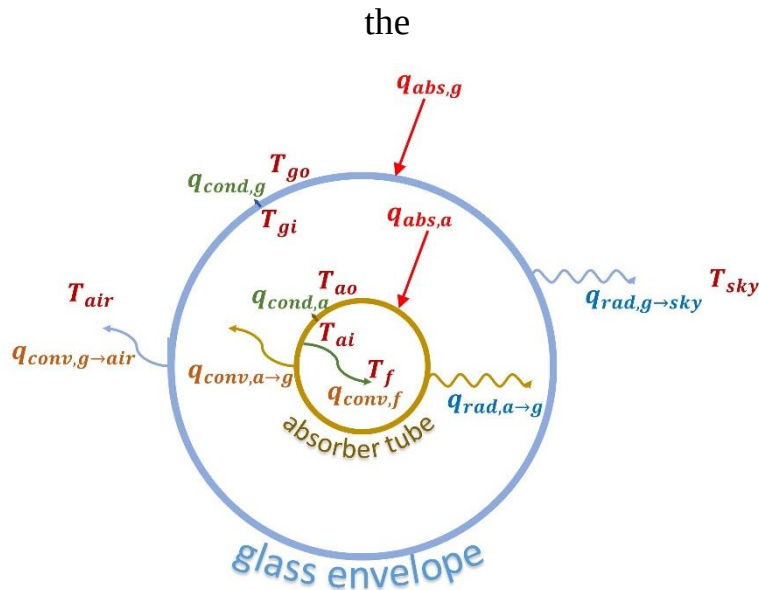


Figure 3.10: Flow of energy in the PTC absorber tube

absorber tube. The other components of energy flowing in the absorber tube are:

- the conduction heat transfer ($q_{cond,a}$ and $q_{cond,g}$ through the absorber tube and the glass envelope walls, respectively),
- the convection heat transfer ($q_{conv,f}$ for the fluid in the absorber tube, $q_{conv,a \rightarrow g}$ in the air trapped between the absorber tube and the glass envelop, and $q_{conv,g \rightarrow air}$ for the convection losses from the glass envelop to the surrounding air), and
- the radiation losses ($q_{rad,a \rightarrow g}$ for the radiation losses from the absorber tube surface to the glass envelope and $q_{rad,g \rightarrow sky}$ for the radiation losses from the surface of the glass envelope to the sky).

The losses from the system can be calculated by the equations shown below using the thermal network shown in Fig. 3.11.

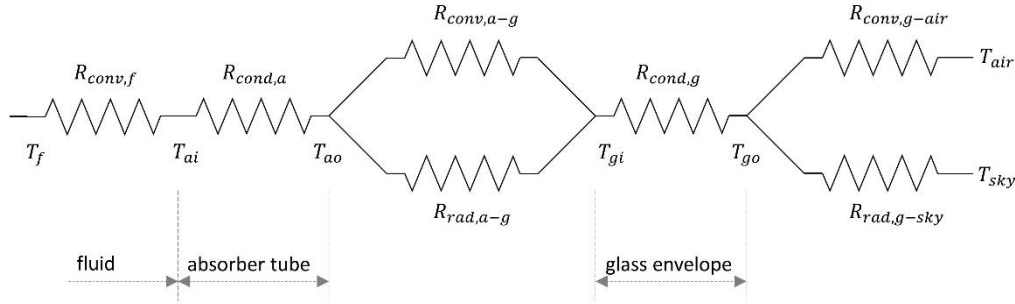


Figure 3.11: Thermal resistance network showing the heat losses in the PTC

$$\dot{q}'_{focus} = \frac{I_{dn} \cdot r \cdot R \cdot \kappa \cdot A_a}{L} \quad (3.25)$$

$$\dot{q}'_{abs_g} = \dot{q}'_{focus} \alpha_g \quad (3.26)$$

$$\dot{q}'_{abs_a} = \dot{q}'_{focus} \tau_g \alpha_a \quad (3.27)$$

$$\dot{q}'_{abs_g} + \dot{q}'_{rad,a \rightarrow g} + \dot{q}'_{conv,a \rightarrow g} = \dot{q}'_{rad,g \rightarrow sky} + \dot{q}'_{conv,g \rightarrow air} \quad (3.28)$$

$$\dot{q}'_{abs_a} = \dot{q}'_{rad,a \rightarrow g} + \dot{q}'_{conv,a \rightarrow g} + \dot{q}'_{conv,f} \quad (3.29)$$

$$\dot{q}'_{Abs} = \dot{q}'_{abs_g} + \dot{q}'_{abs_a} \quad (3.30)$$

$$\dot{q}'_{ThermalLoss} = \dot{q}'_{rad,g \rightarrow sky} + \dot{q}'_{conv,g \rightarrow air} \quad (3.31)$$

$$\dot{q}'_u = \dot{q}'_{Abs} - \dot{q}'_{ThermalLoss} \quad (3.32)$$

$$\dot{q}'_{cond,a} = \dot{q}'_{conv,f} \quad (3.33)$$

$$\dot{q}'_{cond,g} = \dot{q}'_{rad,a \rightarrow g} + \dot{q}'_{conv,a \rightarrow g} \quad (3.34)$$

where

I_{dn} = direct normal solar radiation, (W/m²)

r = reflector mirror's reflectance, dimensionless

R = absorber interception factor, dimensionless

κ = factor for IAM (incident angle modifier), dimensionless

L = length of the PTSC, (m)

α_g = glass envelope absorptivity, dimensionless

τ_g = glass envelope transmittance, dimensionless

A_a = absorber pipe absorptivity, dimensionless

$\dot{q}'_{focus}$ = direct solar radiation on the outer glass envelope surface, (W/m)

$\dot{q}'_{abs_g}$ = solar radiation absorbed by glass envelope, (W/m)

$\dot{q}'_{abs_a}$ = solar radiation absorbed by absorber pipe, (W/m)

$\dot{q}'_{rad,a \rightarrow g}$ = radiation heat transfer between the absorber pipe and the glass envelope, (W/m)

$\dot{q}'_{conv,a \rightarrow g}$ = convection heat transfer between the absorber pipe and the glass envelope, (W/m)

$\dot{q}'_{rad,g \rightarrow sky}$ = radiation heat transfer from the glass envelope to the sky, (W/m)

$\dot{q}'_{conv,g \rightarrow air}$ = convection heat transfer from the glass envelope to the ambient air, (W/m)

$\dot{q}'_{conv,f}$ = convection heat transfer from the absorber pipe to the fluid, (W/m)

$\dot{q}'_{Abs}$ = direct solar radiation absorbed by the PTC, (W/m)

$\dot{q}'_u$ = useful solar energy delivered by the PTC, (W/m)

$\dot{q}'_{ThermalLoss}$ = heat losses from the to the surrounding environment, (W/m)

Each of the energy terms mentioned above can be calculated from equations found in reference books on heat transfer (Holman, 2010, Kreith et al., 2011, Incropera et al., 2012, Duffie & Beckman, 2013). The conduction heat transfer equations are:

$$\dot{q}'_{cond,a} = 2\pi k_a \frac{(T_{a1} - T_{a2})}{\ln(r_{a2}/r_{a1})} \quad (3.35)$$

$$\dot{q}'_{cond,g} = 2\pi k_g \frac{(T_{g1} - T_{g2})}{\ln(r_{g2}/r_{g1})} \quad (3.36)$$

where k is the thermal conductivity, T_1 and T_2 represent the inner and outer tube temperatures, r_1 and r_2 represent the inner and outer tube radius and the subscripts a and g represent the absorber tube and the glass tube, respectively. The convective heat transfer equations are given as:

$$\dot{q}'_{conv,f} = \pi d_{a1} h (T_{a1} - T_f) \quad (3.37)$$

$$\dot{q}'_{conv,a \rightarrow g} = \pi d_{a2} h (T_{a2} - T_{g1}) \quad (3.38)$$

$$\dot{q}'_{conv,g \rightarrow air} = \pi d_{g2} h (T_{g2} - T_\infty) \quad (3.39)$$

where h is the convective heat transfer coefficient and can be determined based on the properties of the fluid, d_1 and d_2 are the inner and outer tube

diameters, and T_{∞} refers to the ambient temperature. The radiative heat transfer equations are:

$$\dot{q}'_{rad,a \rightarrow g} = \sigma \pi d_{a2} \left[\frac{T_{a2}^4 - T_{g1}^4}{\frac{1}{\varepsilon_{a2}} + \frac{d_{a2}}{d_{g1}} \left(\frac{1}{\varepsilon_{g1}} - 1 \right)} \right] \quad (3.40)$$

$$\dot{q}'_{rad,g \rightarrow sky} = \sigma \pi d_{g2} \varepsilon_{g2} (T_{g2}^4 - T_{sky}^4) \quad (3.41)$$

where σ is the Stefan-Boltzmann constant ($= 5.670367 \times 10^{-8} \text{ W/m}^2\text{K}^4$), ε is the surface emissivity and T_{sky} is an equivalent temperature for sky (assuming it to behave as a blackbody) and is equal to $0.0552T_{\infty}^{1.5}$. Other symbols and subscripts have the same meaning as in the previous equations.

The thermal efficiency (η) of a PTC is defined as:

$$\eta = \frac{Q_u}{A_{ap} \cdot I_{dn}} \quad (3.42)$$

where

A_{ap} = aperture area, m^2 ,

Q_u = useful heat absorbed by the absorber tube, (W/m^2) given by:

$$Q_u = \dot{m}_f C_{p,f} (T_{f,out} - T_{f,in}) \quad (3.43)$$

where

$\dot{m}$ = mass flow rate (kg/s),

C_p = specific heat of the fluid, (J/kg.K),

T_{out} = outlet temperature of the heat transfer fluid, (K),

T_{in} = inlet temperature of the heat transfer fluid, (K),

and the subscript f refers to the heat transfer fluid.

Chapter 4

Experimental Setup and Procedure

Table of Contents

Subject	Page No.
Chapter Four Experimental Study	64
4.1 Experimental Setup	65
4.1.1 Evaporation Chamber	68
4.1.2 Freshwater Chamber	69
4.1.3 Condenser Space	69
4.1.4 Piping and Storage Tanks	71
4.1.5 Instrumentation	73
4.2 Experimental Procedure	78
4.2.1 Batch Operation	79
4.2.2 Continuous Operation with Heating in Evaporation Chamber	81
4.2.3 Continuous Operation with Heating in Feed Water Storage Tank	82
4.3 Uncertainty Analysis	83
4.3.1 Uncertainty and Confidence Intervals	83
4.3.2 Propagation of Errors	86
4.3.3 Uncertainty in the Measured Values	88

Chapter 4: Experimental Setup and Procedure

In order to investigate NVD desalination experimentally, a conical roof system was designed and built. This chapter describes this experimental setup in detail. The procedure for the operation of the experimental setup is explained and the different modes of operation that have been investigated are also detailed.

4.1. Experimental Setup

The experimental setup used in this study is shown in Fig. 4.1. It is composed of a cone-shaped tank with an evaporation chamber in the inside and a cooled freshwater chamber on the outside. Stainless steel sheets with 3 mm thickness have been used for the construction of the tank. The tank is placed on the roof of a building while the storage tanks for the freshwater and seawater are placed on the ground level in order to achieve the required height for the water column to balance the atmospheric pressure (almost 10.3 m) as shown in the schematic diagram of Fig. 4.2. Since the system is to be operated under vacuum, care was taken to ensure that all connections are properly sealed. The cylindrical base and the conical roof of the tank were joined by a flanged connection. The flange was made using 4 mm stainless steel sheet. A 4 mm rubber gasket was used to seal the joint and 50 ASTM-A307 bolts have been used to ensure a tight joint. For additional safety, the flange was enveloped by a 10 cm diameter plastic pipe enforced with stainless steel wire that was completely filled with silicone. The pipe connections from the main tank to the storage tanks on the ground level were made using PPR (polypropylene random) pipes with thermal connections. The pipes had a pressure rating of 25 bar and can handle temperatures up to 90 °C. These pipes were used due to the low pressure in the system.

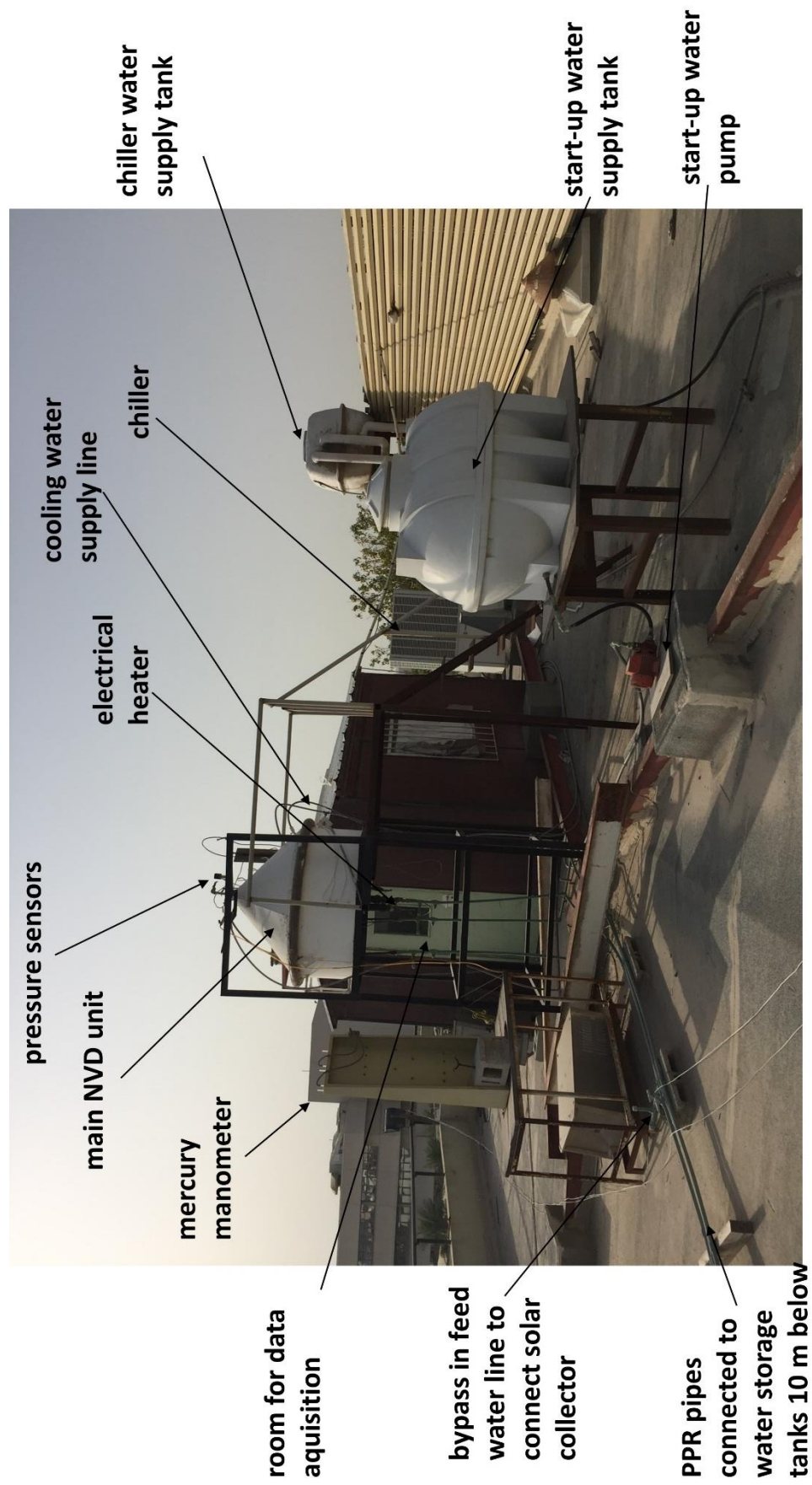


Figure 4.1: Photo of the experimental setup

The main components of the experimental setup are explained in the following subsections.

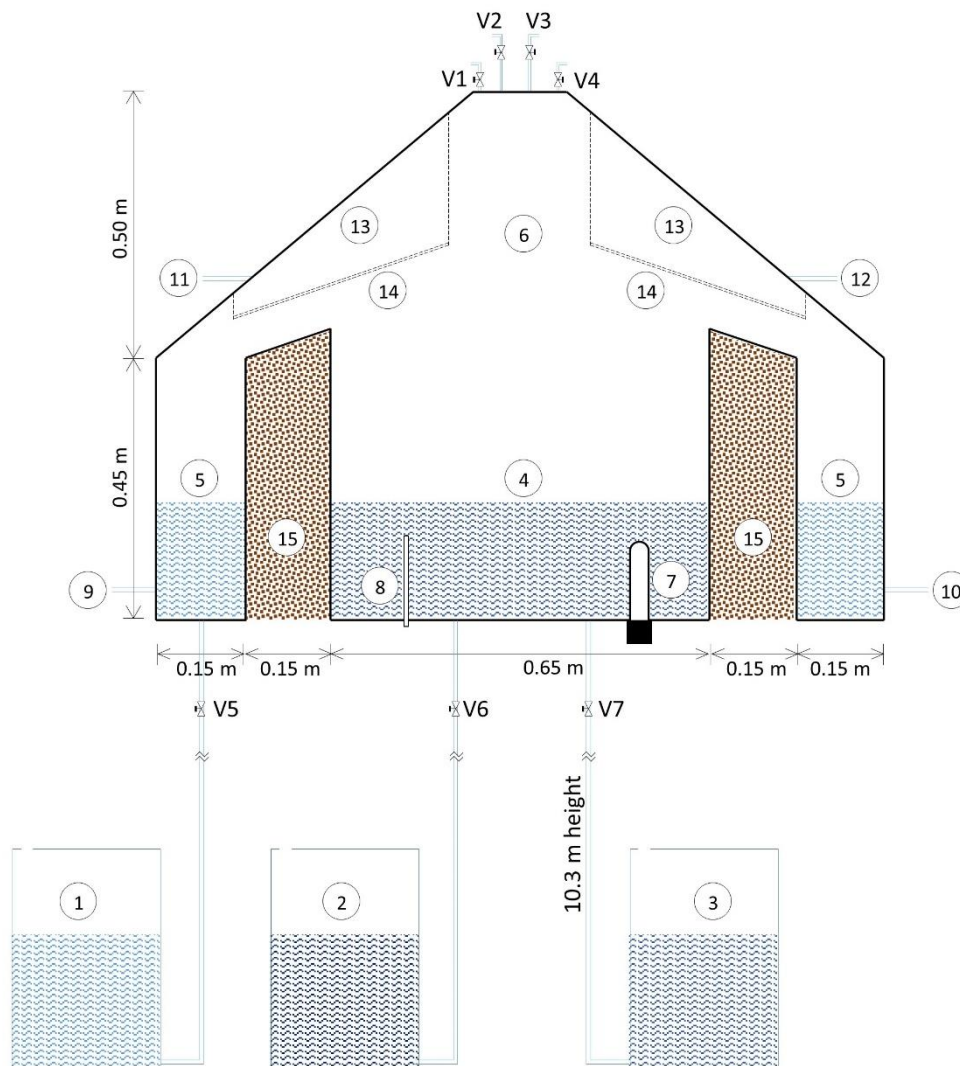


Figure 4.2: Schematic representation of the proposed NVD system
 1) freshwater storage tank 2) brine storage tank 3) saline water storage tank 4) evaporation chamber 5) freshwater chamber 6) conical shaped condenser space 7) electrical heater 8) temperature probe 9) freshwater chamber cooling coil inlet 10) freshwater chamber cooling coil outlet 11) conical tank roof cooling coil inlet 12) conical tank roof cooling coil outlet 13) condensation fins 14) ledge at bottom of fin 15) insulated annular space

4.1.1. Evaporation Chamber

The evaporation chamber is the internal cylinder of the conical tank. It is a 0.65 m diameter tank with a height of 0.50 m. The evaporation chamber has a volume of 0.166 m³. It has four openings on its bottom side. Two of these are used to connect the saline water inlet and the brine withdrawal pipes. A third opening is used for insertion of a temperature probe. This probe is used to measure temperature variation of the water in the evaporation chamber with water height, as shown by (8) in Fig 4.2. The fourth opening allows for connection of an electrical heater or a heat exchanger pipe for heating the water in the evaporation chamber, as shown by (7) in Fig 4.2. For the experimental purposes, an electrical heater with a power rating of 1 kW was used. The heater was also fitted with a Multicomp® Control Thermostat switch to control the temperature of the heater. Additionally, it was fitted with a safety level switch to turn off the heater in case of a drop in the water level of the evaporation chamber. The inside view and the bottom view of the evaporation chamber can be seen in Fig. 4.3.



Figure 4.3: Photos of the evaporation chamber (a) top view of the evaporation chamber (without roof), and (b) view from the bottom of the evaporation chamber showing the heater and pipe connections

4.1.2. Freshwater Chamber

The freshwater chamber is the outer cylinder of the conical tank. This is an annular tank with the outer diameter of 1.25 m and the inner diameter of 0.95 m. This chamber has a height of 0.45 m. The freshwater chamber has a volume of 0.233 m³. The inner walls of the freshwater chamber are connected to the walls of the evaporation chamber at the top to have an inclined surface that allows the condensed water droplets to flow down into the freshwater chamber. The annular space between the freshwater chamber and the evaporation chamber is insulated in order to minimize the heat losses on either side. The freshwater chamber needs to be at a lower temperature than the evaporation chamber and thus has a cooling coil in it. The cooling coil is a 0.0127 m copper tube through which cooling water flows. The coil has three turns along the circumference of the freshwater chamber. Based on estimated water temperature of 60 °C and a cooling water temperature of 20 °C, the length of the cooling coil tube was determined to be 9 m.

4.1.3. Condenser Space

One of the novelty of the design of this setup, as compared to earlier studies, is the absence of an isolated condenser. The conical roof of the tank is cooled using a cooling coil that forms three turns around its inner surface. To enhance the condensation process, the roof is fitted with condensation fins that are connected to the cold roof and extending over the cooling coil. The lower edge of the fin is connected to a ledge that collects the condensed droplets that fall along the surface of the fin. The ledge is sloped downwards to let the collected water flow in to the freshwater chamber. A total of 14 equally spaced fins are attached to the roof, Fig. 4.4. The conical section is 0.50 m in height and has an angle of 40°. At the center of the roof, are four pipe connections. These are used for

the connection to the pump for initial filling of the tank, the standby vacuum pump, vacuum pressure sensors and an opening for overflow.



(a)



(b)

Figure 4.4: Photos of the condenser space (a) the condenser fin and the thermocouple wells are shown (b) the cooling coil can be seen spiraling along the inner surface of the conical roof

The cooling coil used is a 0.0127 m (half-inch) copper tube. It spirals along the conical roof in the condenser space and makes three turns in the roof. The length of the conical coil was determined assuming an average vapor temperature of 60 °C and a cooling water temperature of 20 °C. It was estimated to be around 11 meters. Hence, three turns around the circumference were used, which gives a cooling coil of 11.8 m. For experimental purposes, a Trane ® Air-cooled water chiller was used to provide cooling water. It had a cooling capacity of 19.5 kW, with a 6.2 kW compressor and 0.72 kW pump which could deliver cooling water at a rate of 55 L/min.

A vacuum pump was fitted to one of the four connections at the top of the conical roof. A water-ring vacuum pump with a power rating of 1.6

kW was connected to the system to compensate for any pressure losses during the experiments. The water-ring vacuum pump was also connected to a direct condensation tank as shown in Fig. 4.5. As the non-condensable gases are sucked out by the vacuum pump, the water vapor is also carried along. Using such a vacuum pump, this water vapor can be condensed to provide extra production. Using a regular vacuum pump will cause this water vapor to be lost.

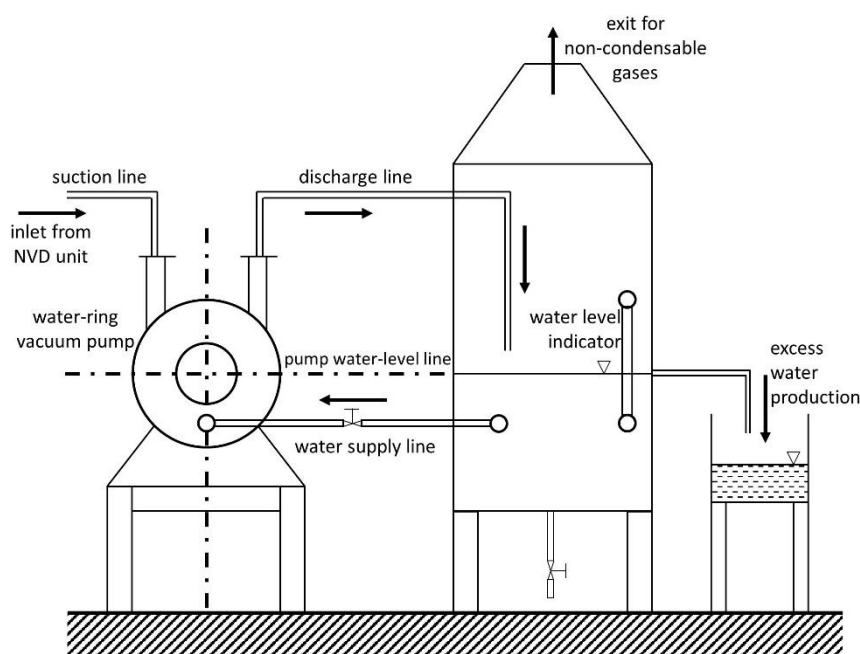


Figure 4.5: Water-ring vacuum pump allows removal of non-condensable gases and recovers the water vapor

4.1.4. Piping and Storage Tanks

The storage tanks are PVC (polyvinyl chloride) drums that are open to atmosphere. They are placed on the ground level. The tanks need to be open to atmosphere; hence, small holes were made in the top of each tank. Other than these holes, the tanks were covered to keep dust and other contaminants from entering. PPR (polypropylene random) pipes with 0.0127 m diameter are used to connect the bottom of the storage tanks with the conical tank, with an elevation of 10.3 m. The outlets from the

evaporation chamber and the freshwater chamber are both fitted with valves. Special connectors were used to provide a leak free PPR to copper pipe interface to connect to the copper pipes coming from the evaporation and freshwater chamber. Provisions area also made in the pipe connecting the feed water storage tank to the evaporator to connect a solar collector. This allows the water fed from the storage tank to be heated by the solar collector before entering the evaporation chamber. The storage tanks and the PPR pipes used in the experimental setup are shown in Fig. 4.6.



(a)



(b)

Figure 4.6: Photos of the (a) the storage tanks placed on the ground floor to provide the barometric height of water, and (b) the thermal fittings on the PPR piping

Initially the pipes were connected to the tanks from the top of the tanks. However, this connection caused air bubbles to enter the pipes if the level of water went below the submerged pipes. To avoid this issue, the pipes were connected to the bottom of the storage tanks. The feed water

tank needs to be higher than the brine and distilled water tank, so that water can flow in the right direction for each tank. To make this possible, the feed water tank was kept on an adjustable elevation control platform. The tanks were also fitted with transparent acrylic tubes to observe the water level. A mercury thermometer was used to measure the temperature of the water in the storage tanks.

4.1.5. Instrumentation

To measure the temperature, pressure and other variables in the experimental setup, appropriate measurement instruments were used. Fig. 4.7 shows the location of these instruments. The various instruments are discussed in the following sub-sections.

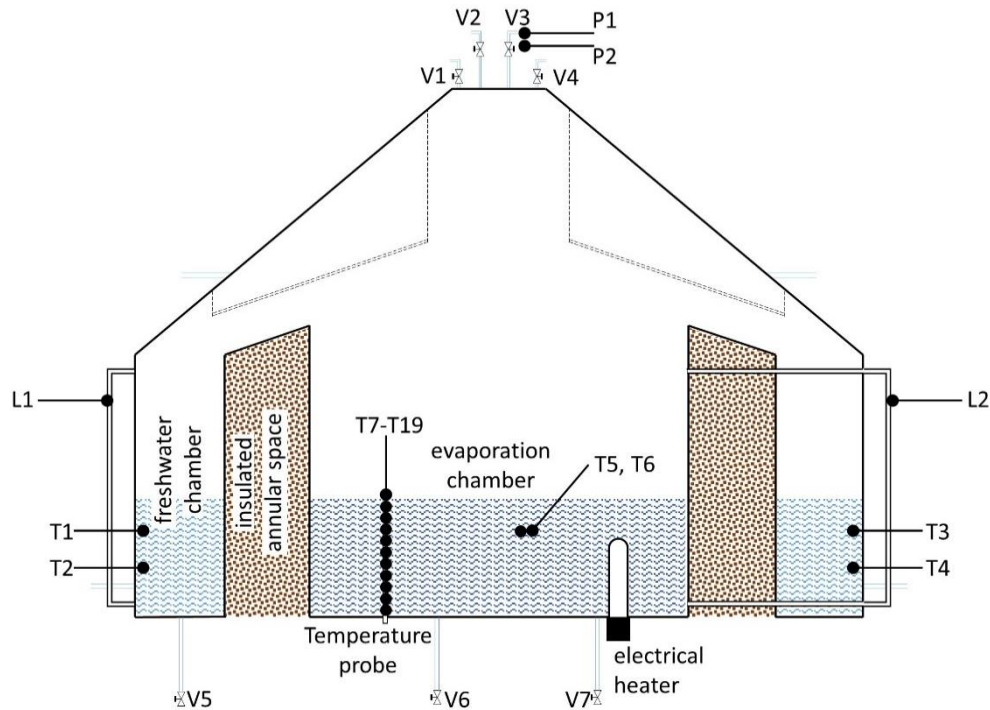


Figure 4.7: Measuring instrumentation locations

Temperature measurement:

The temperatures were measured using calibrated K-type thermocouples. All thermocouples were prepared in the lab and were

calibrated using freezing and boiling water. Calibration curves were prepared for all thermocouples and the readings were adjusted accordingly. The thermocouples were placed in thermal wells prepared by using copper tubes for high conductivity. The wells were filled with a high conductivity oil to ensure precise temperature measurements. They were connected to a Digi Sense Thermocouple Scanning Thermometer with 12-Channels. All thermocouple wires were shielded to avoid electromagnetic interference that could cause disturbance in the temperature measurements. The thermocouples were placed at various points in the experimental setup to measure temperatures at required areas:

- six thermocouples in the freshwater chamber (3 different heights on either side of the chamber),
- four thermocouples in the center of the evaporation chamber (near the heat source),
- six thermocouples in the condensation space to measure temperatures of the water vapor.

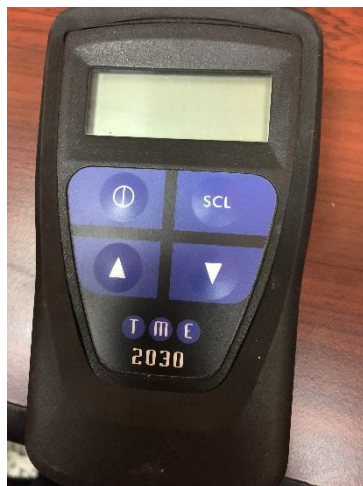
In addition to the above-mentioned thermocouples, a probe was prepared to measure the thermal stratification in the evaporation chamber. A 35-cm copper tube was fitted with 13 thermocouples that were located at an equal distance of 2.5 cm apart. The first thermocouple was 1 cm above the bottom of the evaporation chamber. Successive thermocouples were kept on alternating sides of the tube to avoid contact of the measurement ends. The top end of the tube and each of the small holes for the thermocouples were sealed with epoxy resin to avoid leaking of water through these holes. For further protection, the lower end (where the thermocouple wires exited) was also sealed with epoxy resin after filling the tube with silicon.



(a)



(b)



(c)



(d)

Figure 4.8: Photos of the temperature measurement instruments (a) Digi-Sense Scanning Thermometer (b) Digi-Sense hand-held thermometer (c) TMC 2030 calibrator (d) Fluke infrared thermometer

The thermocouples were also calibrated from time to time using a TMC thermocouple Calibrator. A Fluke® 62 Mini iR Thermometer was used to measure the surface temperature of the tank. Fig 4.8 shows all the instruments used for temperature measurement.

Pressure measurement:

The pressure in the tank was measured using two Teledyne Hastings 760s vacuum sensors connected to Teledyne Hastings HPS-760 display meters (see Fig 4.9). These sensors were selected because they can measure vacuum pressures in the range of 0 to 1000 mm Hg absolute. The compatible display allows direct display of the pressure in mm of Hg and also allows interface with a computer. The sensors were connected to a valve at the top of the conical roof. A specially designed stainless steel Tee-connector was used to connect the two sensors to the tank. Additionally, a mercury manometer was also used to check the pressure. This manometer was used only to ensure that the sensors were giving proper readings.



(a)



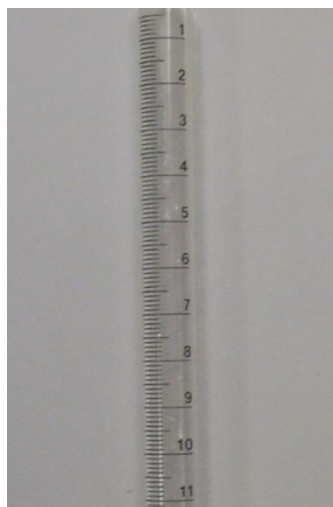
(b)

Figure 4.9: Photos of the pressure measurement instruments (a) Teledyne Hastings 760s Vacuum Sensor (b) Teledyne Hastings HPS-760 display meter

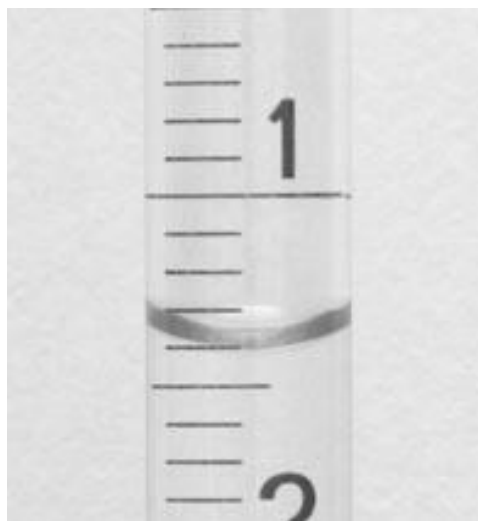
Level measurement:

The level in the evaporation chamber and the freshwater chamber is observed using scaled transparent acrylic level indicator tubes connected to the outside of the experimental setup with a resolution of 0.001 m (1

mm) as shown in Fig 4.10. A high-resolution camera was used to record the level readings to remove any parallax errors caused by level observation using the naked eye.



(a)



(b)

Figure 4.10: Photos of the level measurement tubes (a) scaled transparent acrylic tube (b) high-resolution photo of level measurement to get accurate readings

Flow measurement:

The flowrate of the cooling water used in the condenser and the freshwater chamber was measured using a rotameter. To avoid the problem of pressure leakage, flowmeters were not installed in the water feed and withdrawal lines. The fresh water accumulation was measured by noting the change of water level in the storage tank. The brine withdrawal rate was measured on regular intervals of 1 hour using a graduated beaker and a stopwatch.

Measurement of ambient conditions:

The solar intensity was measured using a Kimo® SL100 solarimeter. This meter was used to measure hourly solar radiation. The meter has a

range of up to 1300 W/m^2 , which is within the range of the solar irradiation of the Kingdom of Bahrain. A Testo® 625 Thermohygrometer was used to measure the ambient temperature and air quality. These instruments are shown in Fig 4.11.



(a)



(b)

Figure 4.11: Photos of the (a) Kimo® SL-100 solarimeter, and (b) Testo® 625 Thermohygrometer

4.2. Experimental Procedure

Since the system is designed to run under almost vacuum conditions, it is important to make sure that there are no pressure leaks in the main tank and the connections. As explained earlier, the main tank was sealed properly using a gasket and silicon ring around the main joint between the cylindrical bottom and conical roof. For other connections, epoxy resin was used to ensure that no leaks were present.

Before operation, the system was tested for leaks. For the testing, the system was filled with water using a pump. After being completely filled, water was allowed to drain into the storage tanks 10 m below the main NVD tank. When the atmospheric pressure balances the water column height, the system is under a natural vacuum. It was then allowed to remain

under natural vacuum to monitor changes in the pressure to detect any possible leakages. From an initial absolute pressure of 60 mm Hg, the pressure was observed to rise to around 90 mm Hg after 24 hours. This rise in pressure is due to the water vapor.

The experiments were conducted in three different ways. The first mode of operation was the batch mode, in which there was no water entering or leaving the unit through the feed line or the brine withdrawal line. The primary objective of the batch process was to study the system without too many disturbances to the operating conditions. It also allows the simultaneously measurement of evaporation and condensation amounts. The other two modes were continuous in nature and varied only in the way the water was heated. In the first process, the water was heated directly in the evaporation chamber. This procedure was used to simulate processes where a heat exchanger or any other low-grade heat source is used to heat the water in the evaporation chamber. In the second continuous operation, the water was heated in the feed tank so that the water is already heated before entering the evaporation chamber. This experiment was used to simulate setups where water is heated using solar thermal collectors or any other heating preheating process. The procedure for all these different modes of operation are described in the following sub-sections.

4.2.1. Batch Operation

The conical tank is first filled with water using a pump. All valves are closed except valves V1, through which water enters the tank, and V2, which is left open to let the air escape. Pumping of water is stopped when water overflows through this outlet. Once the tank is full, the pump is switched off and valves V1 and V2 are also closed. Care is needed to ensure that no air bubbles are left in the pipes. The storage tanks on the ground level should have a water level above the pipe connections so that air does

not enter the pipes. Similarly, all valves at the top of the conical tank should be closed. Valves V5, V6 and V7 are then opened through which water drains down under the influence of gravity. As the water level drops in the conical tank, a vacuum is created in the space above the evaporation chamber and the freshwater chamber. The difference of the height between the conical tank and the storage tanks is adjusted so that the water drops down to a visible level in the level indicator tubes. On attaining the required level valves V5, V6 and V7 are closed (for continuous operation of the process, these valves are left open to allow continuous withdrawal of freshwater and brine and for continuous makeup of seawater to the evaporation chamber).

Once the vacuum is created, the water in the evaporation chamber is heated, either through an electrical heater or through heat exchange with a hot liquid stream heated using solar energy or any other low-grade heat source. Since the process is running at very low pressures, water boils at low temperatures and high-energy inputs are not required. As the water vapor rises in the conical tank, it will condense upon contact with the cooled conical roof or the condensation fins. Both the roof and the fins are sloping downwards to let the water droplets slide in to the freshwater chamber. To ensure that the vapor pressure over the freshwater chamber is lower than the evaporator section, the freshwater chamber is cooled by using cooling water running through the cooling coils in the chamber. A vacuum pressure of around 8 kPa was recorded initially. Levels in both the evaporation chamber and the freshwater chamber, as well as the pressure in the tank, were recorded at regular intervals. Temperature readings were recorded by the computer using the ScanLink software connected to the Scanning Thermometers. For testing procedure, water in the evaporation

chamber was heated using an electrical heater with different operating temperatures.

4.2.2. Continuous Operation with Heating in Evaporation Chamber

The conical tank is first filled with water using a pump and the water is then allowed to drain under the influence of the gravity just as in the batch operation. The only difference is that the valves V5, V6 and V7 are left open to allow for withdrawal of freshwater and brine and for inlet of feed seawater. The feed water makes up for the amount of water evaporated and the amount that is withdrawn as brine. The level in the tank is thus maintained constant to maintain the barometric height of the water column.

The feed water tank is then raised to a level higher than the brine storage tank and the freshwater storage tank. The elevated feed water tank makes the water from the feed water tank to move up to the evaporation chamber under the influence of the atmospheric pressure. No mechanical energy is needed to pump the water up to the evaporation chamber. On the other hand, the brine storage tank is at a lower level and hence water flows down through the brine withdrawal line into the brine storage tank. This downward movement is under the influence of gravity and no mechanical energy is required for this flow as well.

The top of the conical tank is connected with a vacuum pump to allow for removal of incondensable gases that may add to the pressure in the tank. A drop in the water level in the evaporation chamber indicates an increase in the pressure. It was also noted that some bubbles formed in the brine withdrawal line could cause blockage of the withdrawal line, hence causing a rise in the water level in the evaporation chamber. A pump was fitted at the bottom of the withdrawal line to get rid of any such blockage. The water

in the evaporation chamber was heated using the electrical heater connected to the evaporation chamber.

4.2.3. Continuous Operation with Heating in Feed Water Storage Tank

This is same as the previous continuous operation, with the difference only in the way the water is heated. In the previous operation, water was heated using an electrical heater in the evaporation chamber. In this operation, water was heated using a 1.5 kW electrical heater connected in the feed water storage tank. A schematic representation of the heater connected in this tank is shown in Fig. 4.12.

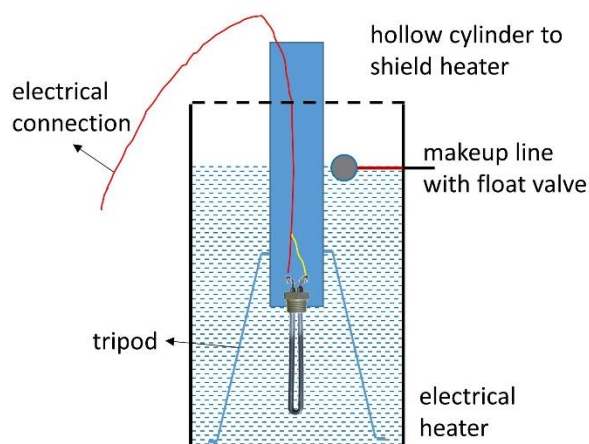


Figure 4.12: Water heater in the feed water tank

A vertical stainless steel pipe of one-meter length was fitted with a tripod stand. The bottom end of this pipe, which was submerged into the water, was sealed using epoxy resin to prevent leakage of water into the pipe where the electrical connections were present. A makeup line was also connected to the feed water tank to compensate for water, which was feed to the evaporation chamber by the atmospheric pressure. A float-valve was used to control the level in the feed water tank. This is important because if the level rises in this tank, the level of water in the evaporation chamber

will also rise and there could be a danger of overflow of saline water into the distilled water chamber.

4.3. Uncertainty Analysis

The data obtained from measurements done during the experiments is used to gain knowledge of the actual physical process. This knowledge will depend greatly on the accuracy of the measured values and the ability to express the “uncertainty” with regards to that these. The measured quantity must thus have two essential components: (1) a numerical value giving the best estimate of the measured quantity, and (2) the degree of uncertainty associated with this estimated value.

4.3.1. Uncertainty and Confidence Intervals

The accuracy and measurement ranges of the instruments used is shown in Table 4.1.

Table 4.1: Accuracy, measurement and error for the measuring instruments used

Location	Instrument	Manufacturer	Range	Error (%)
T1-T29	Thermocouple	Labfacility	–50 to 150 °C	1.5%
P1-P2	Pressure gauge	Teledyne Hastings	0 to 1000 mm Hg	0.25%
L1-L2	Level indicators	made-in-house	0 to 400 mm	0.25 %
	Rotameter	Mobrey	0.015 – 0.14 L/s	5%
offline	Conductivity meter	WTW	0 to 200 μ S/cm 0 to 200 mS/cm	0.5%
offline	Solar intensity meter	Kimo Instruments	1 to 1300 W/m ²	5%

To understand this uncertainty, we look at the basis statistical analysis regarding the measurements. The error, E , in a measured quantity is

defined as the difference between the true value of the measured quantity and the measured value. Since the true value is not available in most cases, this quantity is estimated by using a certain mean value. The most common estimate used in experimental measurements is the arithmetic mean. For a certain measured quantity, X , the arithmetic mean, $\bar{X}$, is defined as:

$$\bar{X} = \frac{1}{N} \sum_{i=1}^N X_i \quad (4.1)$$

where N represents the number of measurements.

The error in the measured value may then be given by:

$$E = X - \bar{X} \quad (4.2)$$

The error obtained by this method is called the absolute error. A shortcoming of this definition is that it does not take into account the order of magnitude of the measured value. One way to account for the magnitude of the measured quantity, is to normalize the error using the true value (or its estimated by using the mean). This is called the relative error, and is given by:

$$\varepsilon = \frac{X - \bar{X}}{\bar{X}} = \frac{E}{\bar{X}} \quad (4.3)$$

The uncertainty in a measured value can be related to the resolution of the measurement instrument (Type B uncertainty). If the resolution is given as $\pm U$, then the measured value should be reported as $X \pm U$. This means that the true value can lie anywhere in the interval from $X - U$ to $X + U$. The width of the uncertainty, in this case, will be $2U$. This is a simplistic approach to uncertainty. To assign a more accurate value to the uncertainty, statistical analysis approach is required.

A more rigorous definition of uncertainty (Type A) relies on the statistical notion of confidence intervals and the Central Limit Theorem (Patience, 2013). The confidence interval is based on calculation of the standard deviation of the mean, $S_{\bar{X}}$. The standard deviation of a sample is calculated as:

$$S_X = \sqrt{\frac{\sum_{i=1}^N (X_i - \bar{X})^2}{N - 1}} \quad (4.4)$$

The Central Limit Theorem then states that the standard error of the mean equals the sample standard deviation divided by the square root of the number of samples:

$$S_N = \frac{S_X}{\sqrt{n}} \quad (4.5)$$

The uncertainty is given as the product of the standard error of the mean and a confidence interval (arbitrarily defined by the experimenter). If the measurements for a certain variable are repeated many times, under the same conditions, then the sample of the measurements follows the normal “Gaussian” distribution. If a single measurement is made, it will be unlikely equal to the mean value, but it should not differ from the mean value significantly, otherwise it has to be rejected. The confidence intervals for a normal distribution are shown in Fig 4.13.

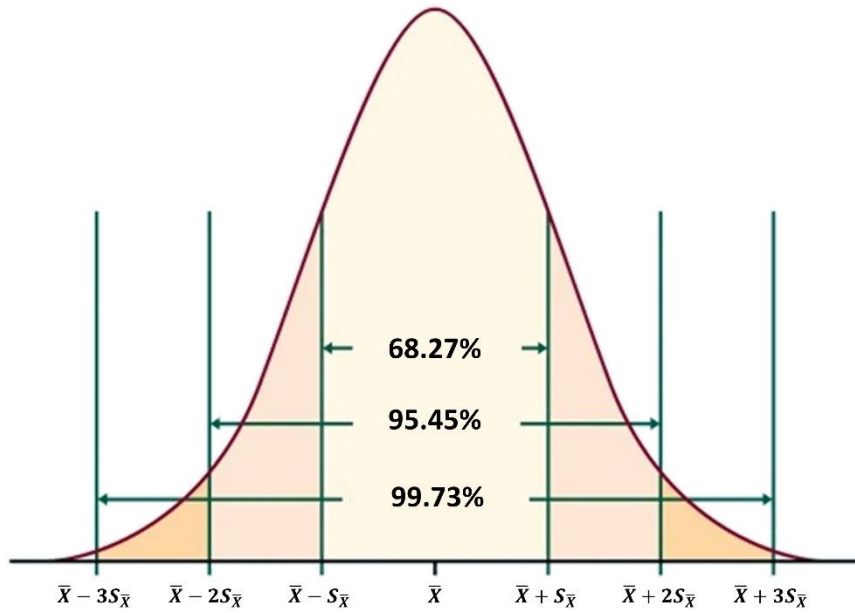


Figure 4.13: Standard deviation and confidence intervals

As can be seen from Fig. 4.13, the confidence level for the range of an uncertainty $U = \pm S_{\bar{X}}$ is only 68.27%, while it is 95.45% for an interval of $U = \pm 2S_{\bar{X}}$ and 99.73% for $U = \pm 3S_{\bar{X}}$. If we desire to use a confidence interval of 95%, then the required uncertainty is given as:

$$\delta = \frac{1}{\sqrt{n}} 1.96 S_{\bar{X}} \quad (4.6)$$

4.3.2. Propagation of Errors

It is very rare to find measurements that rely on a single factor. Volumetric flowrate, for example, depends on measurement of pressure drop, fluid density and geometry, which adds up to three or more factors. All these factors will have some uncertainty associated with them, and these uncertainties will contribute to the overall uncertainty in the measured variable. We can write the measured variable, R , as:

$$R = R(X_1, X_2, X_3 \dots X_N) \quad (4.7)$$

The total uncertainty associated with R could be estimated with good accuracy using a root-sum square combination of the effects of each of the individual inputs and that this operation preserved the odds (Kline & McClintock, 1953). This can be written mathematically as:

$$\delta R = \sqrt{\sum_{i=1}^N \left(\frac{\partial R}{\partial X_i} \delta X_i \right)^2} \quad (4.8)$$

This is the basic equation of uncertainty analysis. Each term represents the contribution made by the uncertainty in one variable, δX_i , to the overall uncertainty in the result, δR . Each term has the same form: the partial derivative of R with respect to X_i multiplied by the uncertainty interval for that variable. The estimated uncertainty in the result has the same probability of encompassing the true value of the result as the uncertainties in the individual measurements have of encompassing their true values.

In many applications, it is desired to express the uncertainty estimate as a fraction of the measurement, rather than in some specified units. While this can always be calculated, starting from the results of the general form in Eq. 4.8, it is sometimes possible to do the calculation of relative uncertainty directly. In particular, whenever the equation describing the result is a pure "product form" like:

$$R = X_1^a \cdot X_2^b \cdot X_3^c \dots X_M^m \quad (4.9)$$

then the relative uncertainty can be found directly as:

$$\frac{\delta R}{R} = \sqrt{\left(a \frac{\delta X_1}{X_1} \right)^2 + \left(b \frac{\delta X_2}{X_2} \right)^2 + \dots + \left(m \frac{\delta X_M}{X_M} \right)^2} \quad (4.10)$$

This is a natural and convenient approach in situations where the uncertainties of the component measurements are described in terms of

percent of reading and the result is needed in the same terms (Moffat, 1988).

The general expression for the uncertainty of a function that is a product of N factors is:

$$\frac{\delta R}{R} = \sqrt{\sum_{i=1}^N \left(a_i \frac{\delta X_i}{X_i} \right)^2} \quad (4.11)$$

The general expression for the uncertainty of a function that involves addition and subtraction becomes:

$$\frac{\delta R}{R} = \sqrt{\sum_{i=1}^N (a_i \delta X_i)^2} \quad (4.12)$$

4.3.3. Uncertainty in the Measured Values

The variables that were measured for the experiment are the temperature, pressure, flow rate and level. The errors associated with each of these measurands is discussed in the following sub-sections.

Error in temperature measurement:

The thermocouples were calibrated using the freezing and boiling points of water, 0 and 100 °C, respectively. Since, the temperature in the experimental study are not expected to be beyond this range, and thermocouples are known to have a linear relation with temperature, linear interpolation can be used to calculate the temperature at any point between the two reference temperatures used for calibration. The actual temperature, T_{actual} , is thus given as:

$$T_{actual} = T_{fc} + \frac{T_{bc} - T_{fc}}{T_{bm} - T_{fm}} (T_m - T_{fm}) \quad (4.13)$$

where the subscript f refers to the freezing point, b refers to the boiling point, c refers to calibration and m refers to measured.

Applying the uncertainty analysis equation (Eq. 4.10) to find the uncertainty of temperature measurements, we can write:

$$\delta_{temp} = \sqrt{\left(\frac{\partial T_{actual}}{\partial T_{bc}} \delta T_{bc}\right)^2 + \left(\frac{\partial T_{actual}}{\partial T_{fc}} \delta T_{fc}\right)^2 + \left(\frac{\partial T_{actual}}{\partial T_{bm}} \delta T_{bm}\right)^2 + \left(\frac{\partial T_{actual}}{\partial T_{fm}} \delta T_{fm}\right)^2 + \left(\frac{\partial T_{actual}}{\partial T_m} \delta T_m\right)^2} \quad (4.14)$$

Knowing that the freezing and boiling points of the water used for calibration were 0 and 100 °C, respectively and assuming that no errors are associated with these values, we can reduce the equation to:

$$\delta_{temp} = \sqrt{\left(\frac{\partial T_{actual}}{\partial T_{bm}} \delta T_{bm}\right)^2 + \left(\frac{\partial T_{actual}}{\partial T_{fm}} \delta T_{fm}\right)^2 + \left(\frac{\partial T_{actual}}{\partial T_m} \delta T_m\right)^2} \quad (4.15)$$

The partial derivatives can be found by differentiating Eq. 4.13 with respect to each of the variables. Noting that δT_{bm} , δT_{fm} and δT_m will be same for all the thermocouples, we can substitute the derivatives in Eq. 4.15 to give:

$$\delta_{temp} = \delta T_m \sqrt{2 \left(\frac{T_{bc} - T_{fc}}{T_{bm} - T_{fm}}\right)^2 \left\{ \left(\frac{T_m - T_{fm}}{T_{bm} - T_{fm}}\right)^2 - \left(\frac{T_m - T_{fm}}{T_{bm} - T_{fm}}\right) + 1 \right\}} \quad (4.16)$$

Statistically, a sample size of 30 is considered large enough (Hogg et al., 2015), hence a samples size of 100 measurements was used to calculate the uncertainty in the temperature measurement. The uncertainty for the thermocouples was calculated using Eq. 4.5 and found to be 0.05 °C. Substituting this value in Eq. 4.16, gives the uncertainty in the temperature measurements which is almost 0.08 °C.

Error in pressure measurement:

The pressure sensors were calibrated using the atmospheric pressure and the maximum attainable vacuum inside the system at the start of the experiments. Like the temperature, the actual pressure in the system is given as:

$$P_{actual} = P_{vc} + \frac{P_{ac} - P_{vc}}{P_{am} - P_{vm}} (P_m - P_{vm}) \quad (4.17)$$

where the subscript v refers to the vacuum, a refers to the atmospheric pressure, c refers to calibration and m refers to measured.

If we assumed that the atmospheric pressure is constant (760 mm Hg) and is measured accurately with no errors, then in a procedure similar to the temperature measurement, we can derive the following equation for the uncertainty in the pressure measurement:

$$\delta_{pressure} = \sqrt{\delta P_{vc}^2 \left(1 - \frac{P_m - P_{vm}}{P_{am} - P_{vm}}\right)^2 + 2\delta P_m^2 \left(\frac{P_{ac} - P_{vc}}{P_{am} - P_{vm}}\right)^2 \left(\left(\frac{P_m - P_{vm}}{P_{am} - P_{vm}}\right)^2 - \frac{P_m - P_{vm}}{P_{am} - P_{vm}} + 1\right)} \quad (4.18)$$

As in the case of the temperature measurement, using a sample size of 100 measurements, we find the uncertainty of the sensors to be 1.1 mm Hg and the uncertainty in all the pressure measurements is calculated to be 5.5 mm Hg.

Error in flowrate measurement:

The volumetric flowrate was measured using a rotameter. The rotameter was calibrated at a water temperature of 20 °C. The actual flowrate can be calculated by:

$$\dot{V}_{actual} = \dot{V}_m \sqrt{\frac{\rho_c}{\rho}} \quad (4.19)$$

where ρ_c is the water density at calibration conditions (i.e. 998.2 kg/m³) and ρ is the density at the operating conditions.

The flowmeter was used to measure the cooling water flow. A flow of around 5 L/min (0.083 L/s) was used for the cooling water. A standard deviation of 0.15 L/min was obtained from 50 measurements. A total error in the flow rate measurement was calculated as 0.71 L/min.

Error in level measurement:

The level was measured in the evaporator chamber and in the freshwater chamber. Both tanks were fitted with graduated acrylic tubes which had a resolution of 1 mm. The level was measured to calculate the evaporation and condensed amounts during the experiments. The evaporation chamber has a cross section area of 0.3318 m² and the freshwater chamber has an annular area of 0.5184 m². Hence, the resolution of the volume determined using the resolution of the level is 0.3318 and 0.5184 L for the evaporation chamber and the freshwater chamber, respectively.

To improve the readings obtained from this configuration, a high-resolution camera was used to take the photos of the level indicators and the reading was taken using interpolation from the digitized image. An accuracy enhancement of five times was achieved using this procedure, so that the resolution of the volume readings was improved to be 0.07 and 0.10 L for the evaporation chamber and the freshwater chamber, respectively.

Chapter 5

Results and Discussion

Table of Contents

Subject	Page No.
Chapter Five Results and Discussion	92
5.1 Experimental Results	93
5.1.1 Batch Operation	93
5.1.1.1 Effect of Heating Temperature	99
5.1.1.2 Effect of Water Salinity	102
5.1.2 Continuous Operation with Heating in Evaporation Chamber	103
5.1.3 Continuous Operation with Heating in Feed Water Storage Tank	108
5.2 COMSOL Model	112
5.3 Model Validation	116
5.3.1 Batch Operation	116
5.3.2 Continuous Operation	118
5.3.3 Simulation Results	120
5.4 Parametric Studies using COMSOL	134
5.4.1 Batch Operation	134
5.4.1.1 Effect of Evaporation Temperature	135
5.4.1.2 Effect of Freshwater Side Temperature	136
5.4.1.1 Effect of Evaporator Size	137
5.4.2 Continuous Operation	137
5.4.2.1 Effect of Evaporation Temperature	138
5.4.2.2 Effect of Freshwater Side Temperature	139
5.4.2.3 Effect of Brine Withdrawal Rate	142
5.4.2.4 Effect of Water Height in Evaporation Chamber	144
5.5 NVD using Solar Parabolic Trough Collector	145
5.6 Cost Analysis	151

Chapter 5: Results and Discussion

This chapter presents the results of the different experimental configurations studied followed by the validation of the COMSOL model using the experimental results. Results of the COMSOL simulation and the parametric studies done using COMSOL are also presented.

5.1. Experimental Results

The experiments were conducted in three different modes of operation. They are batch operation, continuous operation with heater in evaporation chamber and continuous operation with heater in seawater storage tank. As mentioned in chapter 4, these different modes of operations were used to simulate different types of processes. The batch process was used to simulate a desalination still, while the continuous processes were used to simulate continuous desalination processes that could represent active and passive solar stills. Each of these modes of operation are discussed in the coming sections.

5.1.1. Batch Operation

A schematic representation of the experimental procedure has been previously presented in Fig. 4.2. In the batch process, the water was pumped up to fill the NVD tank completely with the air outlet valve (V2) open. Once full, all the valves on the top of the tank (V1-V4) are closed, and the bottom outlet valves (V5-V7) are opened to allow water to drain under the effect of gravity, causing a natural vacuum inside the tank. Once the water level equilibrates with the atmospheric pressure, all the valves are closed. The heater is then turned on and readings from the different sensors are noted. The measured values for the hourly variations of temperature, pressure and accumulated water production amounts are shown in Figs. 5.1-5.3. Fig. 5.1 shows the variation of temperature in water

inside the evaporation chamber. Thermocouples were located in a probe at equal distances of 2.5 cm from 0 to 30 cm, to measure the temperature at different heights. There is sharp temperature rise in the beginning of the heating process to the use of an electric heater. As time passes, the temperature can be seen to settle around a certain steady-state value.

A temperature gradient can be seen with the surface having the higher temperature and the bottom with the lowest temperature values. This difference reaches a maximum value of 4.7 °C between the top and bottom temperature readings. As the height increased the difference in the temperature values reduced, hence for the sake of clarity, only one of the upper thermocouple data has been shown. The temperature difference is due to the buoyancy effect, which causes hot water to rise and the relatively cooler water to move down. In addition, the bottom is not insulated to allow access to the connections of pipes, valves, measurement probes and electrical heater. Due to the absence of insulation at the bottom, heat losses here cause the temperature to be lowered. Assuming an inside temperature of 45 °C and an outside temperature of 30 °C, an average convection heat transfer coefficient of approximately 4.7 W/m².K can be calculated for air which amounts to heat losses of around 23 W from the bottom surface. The temperature difference increases during the nighttime when the ambient temperatures are lower and reduces significantly during the daytime when the temperatures are higher.

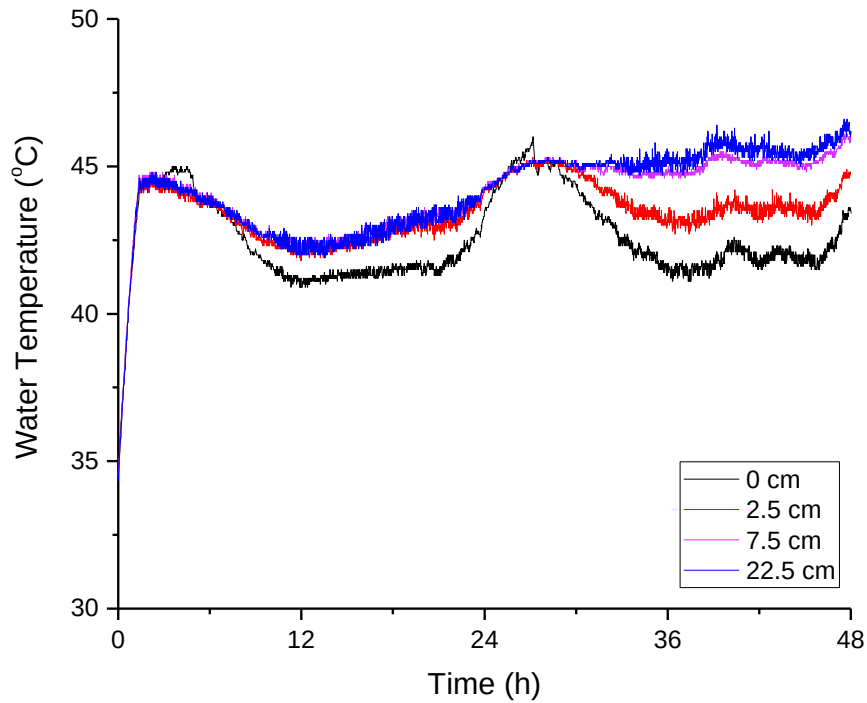


Figure 5.1: Temperatures recorded at different depths in the evaporation chamber

The behavior of the system temperature in response to the changes of the set temperature are shown in Fig. 5.2. Four different heating temperature values, 45, 50, 65 and 70 °C, were used. As shown in Fig 5.1, the temperature readings in the layers near the surface are almost the same. As the water evaporates, the level of water in the evaporation chamber decreases and some thermocouples are no longer under water. The temperature shown here are the temperature of the thermocouple at the highest elevation, which remained under water throughout the experiment. The temperature will not be the temperature of the surface of water, but it is measured close to the surface, hence it is called “near surface water temperature”. It can be seen that the near-surface water temperature does not reach the temperature of the set point of the heater (T_{set}). This difference can be attributed to the heat losses from the system as well as to

the fact that the surface temperature sees a temperature drop as the water evaporates and carries along with it the heat of vaporization.

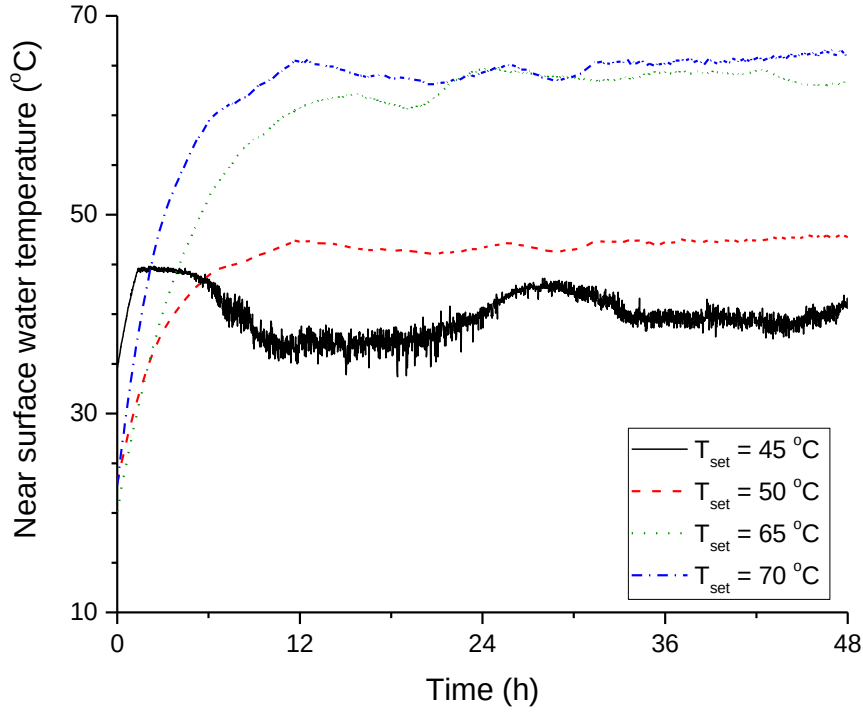


Figure 5.2: Near Surface Water Temperature variations with different heating temperatures

A notable valley can be seen after around 12 hours of operation. The temperature drops and then rises again to become almost steady. This phenomenon is due to the nucleate boiling, which causes the heat transfer coefficient to drop during the partial film-boiling zone. Once the film boiling takes control, we see the rise in temperature again. At lower temperatures, boiling point is not been reached, hence we do not see this trend. It can also be noted from Fig. 5.2 that the temperature in the evaporator section follows a first-order dynamics that can be written as:

$$T(t) = T_o + K_p(T_{set} - T_o)(1 - e^{-t/\tau}) \quad (5.1)$$

where $T(t)$ is the system temperature (in °C) as a function of time t (in hours), T_o is the initial temperature of the system (in °C), T_f is the set-temperature of the heater (in °C), K_p is the system gain and τ is the time constant (in hours) for the system. These parameters for the different set temperatures have been calculated as shown in Table 5.1. For the case of the 45 °C experiment, there was a lot of disturbance in the temperature readings, hence only the portion of the initial rise readings and the near-steady state values at the end were used for fitting. Therefore, the goodness of fit (R^2) value is not reported.

Table 5.1: Model-fitting parameters for different set temperatures

T_{set} (°C)	T_o (°C)	K_p	τ (hours)	R^2
45	34.5	0.4725	3.00	-
50	23.0	0.4805	3.00	0.9867
65	20.2	0.6706	4.72	0.9933
70	22.8	0.8924	2.99	0.9869

Fig. 5.3 shows the pressure rise in the tank with time. Since the pressure rise behavior was similar in all cases, only the readings of the experiments with the lowest and the highest temperature are shown. The pressure rise was exponential with time. The driving force in the desalination process is the pressure difference across the evaporation chamber and the freshwater chamber. Since, both these tanks are under the same conical roof, it was not possible to measure the individual vapor pressures for the two chambers. However, the fact that the two water bodies are at different temperatures is a clear indication of the difference in vapor pressures. The pressure rise in the tank is due the buildup of the water vapor and the presence of non-condensable gases. Some pressures losses may

also be attributed to any small leak left unfixed in the experimental setup, although a thorough check was done to minimize any such loss. The standby vacuum pump provision is made so that non-condensable gases could be removed if the pressure increased to a level that inhibited the evaporation rates. The pressure increase in the system was found to fit to an exponential growth given by the equation:

$$\Delta P = \Delta P_{max}(1 - e^{-t/\tau}) \quad (5.2)$$

where ΔP is the pressure difference (in mm Hg), ΔP_{max} is the maximum expected pressure difference (in mm Hg), t is the time (in hours) and τ is the time constant (in hours). The time constant for the pressure rise was determined to be around 25 hours.

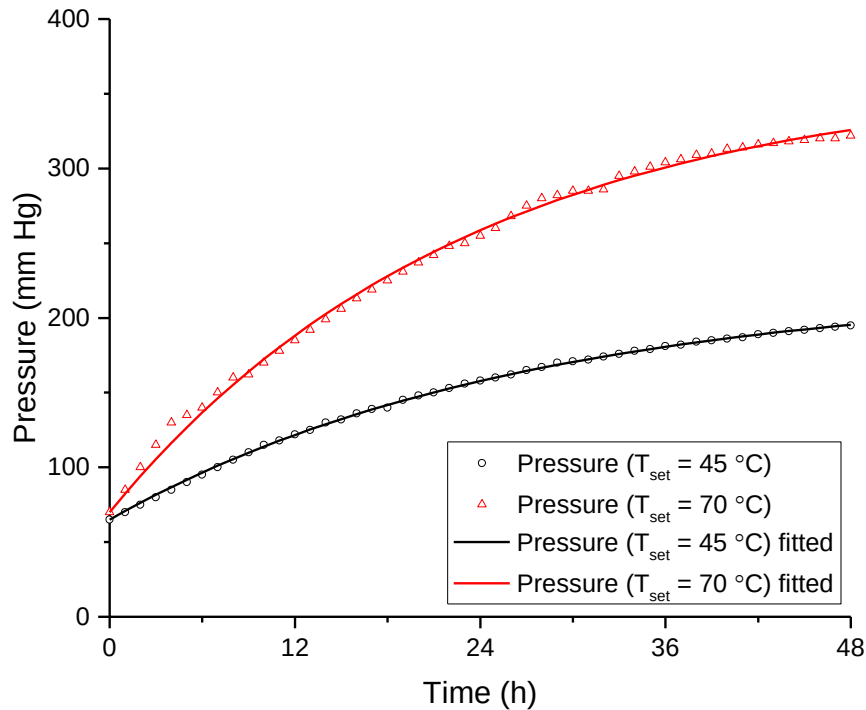


Figure 5.3: Pressure variations in the tank with time for two different operating temperatures (experimental and fitted values according to Eq.

5.2)

The effect of the change of the set point temperature of the heater and the concentration of the feed water on the freshwater production was investigated experimentally. The results are discussed in the forthcoming sub-sections.

5.1.1.1. Effect of Heating Temperature

To study the effect of the heating temperature on the freshwater production, four different temperatures were used. The amount of water evaporated was calculated by noting the change in the level of water in the evaporation chamber. The results are shown in Fig 5.4. A quick rise in the evaporation rates is observed in the beginning of the process because of the sudden rise in temperature. As the temperature gets steady, the evaporation rates also become constant and almost a linear trend is observed. For a linear fit to the data shown in Fig. 5.4 and keeping an intercept of zero, the slopes (evaporation rate) corresponding to the four temperatures are 0.4702, 0.5782, 0.9291 and 1.0599, respectively. This is a clear indication of the increase of evaporation rates with the increase in temperature.

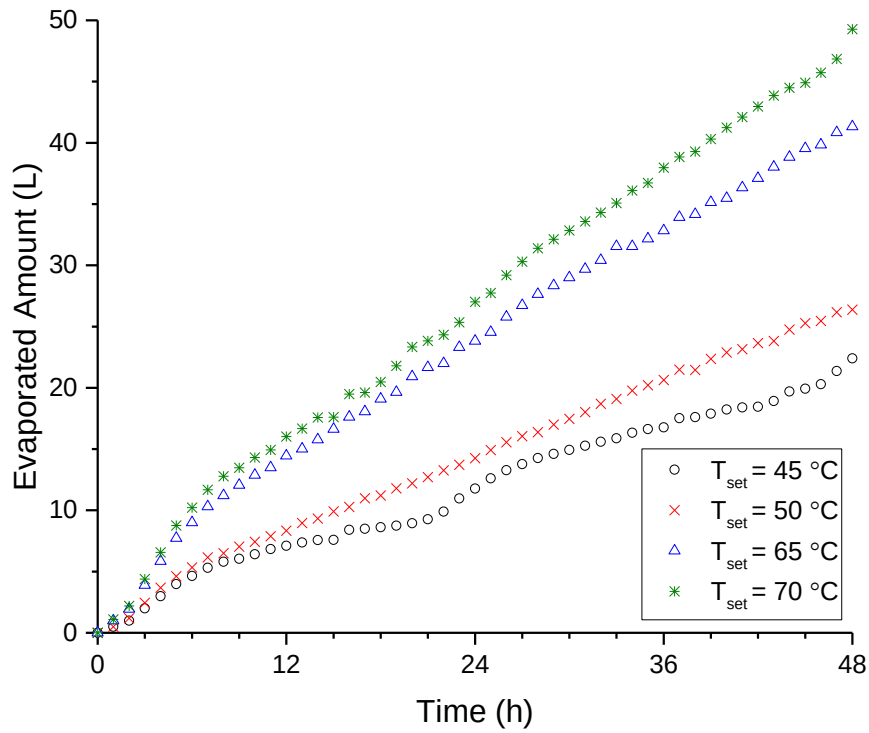


Figure 5.4: Evaporated amount with time at different operating temperatures

The total evaporated and condensed amounts of water with time are shown for two temperatures in Fig. 5.5. The amounts were measured using the drop in the level of water in the evaporation chamber and the rise of the water level in the freshwater chamber. It can be seen that the evaporated amounts are almost the same as the condensed amount, which shows that the condensation method adopted in the system was very effective. The total volume of freshwater evaporated over a period of 48 hours using a temperature of 45 °C was measured as 22.4 L while it was 49.3 L at a temperature of 70 °C. The corresponding condensed amounts were 22.1 L and 48.8 L, respectively.

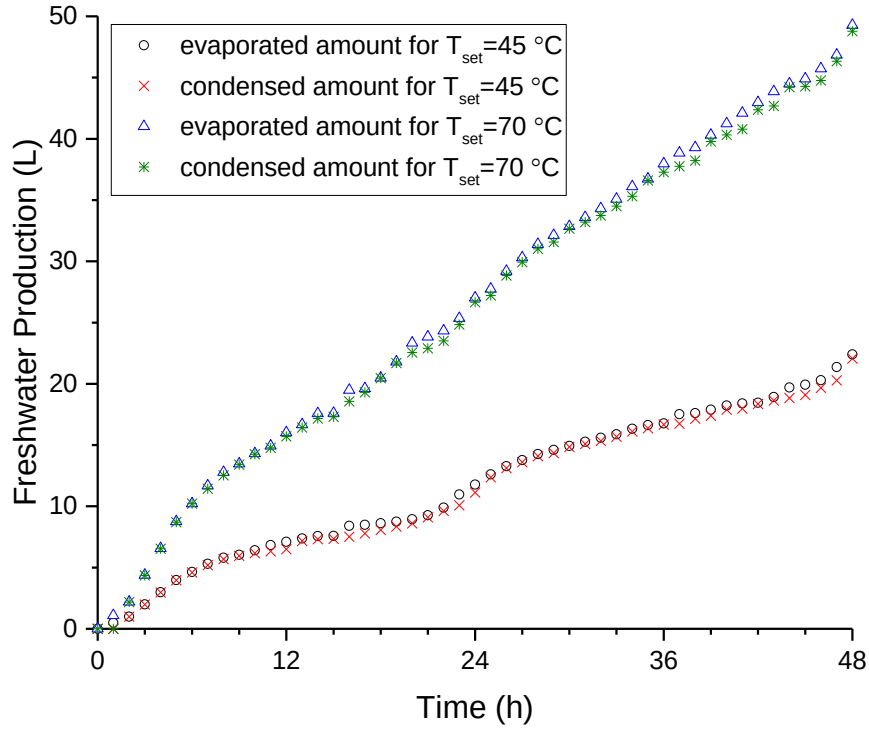


Figure 5.5: Evaporation and condensation amounts at two different operating temperatures

The effect of changing the set temperature, T_{set} , in the heater on the freshwater production amount is shown in Fig. 5.6. The total evaporated amount was noted after a period of 48 hours for the four different operating temperatures. The production rate increases in an inverse exponential growth with the increase in temperature. This is also obvious in the studies by Al-Kharabseh and Goswami (Al-Kharabsheh & Goswami, 2004) and Gude et al. (Gude & Nirmalakhandan, 2008). A general conclusion would be that higher temperatures are more suitable. However, the thermal efficiency of the process has been reported to fall with an increase in the inlet temperature (Gude et al., 2010). It is thus important to study the cost-effectiveness of raising the inlet temperature and find the optimum operating temperature.

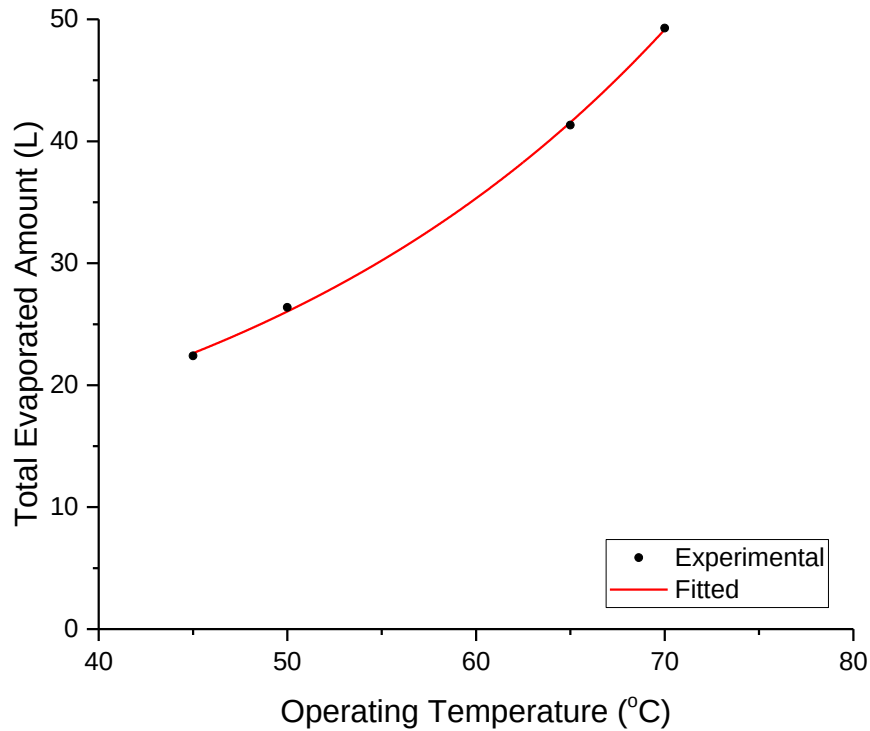


Figure 5.6: Total evaporated amount (in 48 hours) as a function of the operating temperature

5.1.1.2. Effect of Water Salinity

The effect of water salinity on the evaporation amount is also studied. Three concentrations were used to study this effect: 0.1% (tap water), 1% (brackish water) and 3.5% (seawater). The results are presented in Fig. 5.7 for an operating temperature of 50 °C. A decrease in the evaporated amounts can be seen as the salt concentration increases. Initial concentration of 3.5% (representative of seawater concentration) shows a decrease of around 5% in the evaporated amounts as compared to the tap water. The trend appears linear within the range investigated in this studied, however other studies have reported it to be non-linear (Panin & Brezgunov, 2007; Al-Shammiri, 2002). Higher values of salinity were not studied because it was intended to limit the study to salinity values found in the region.

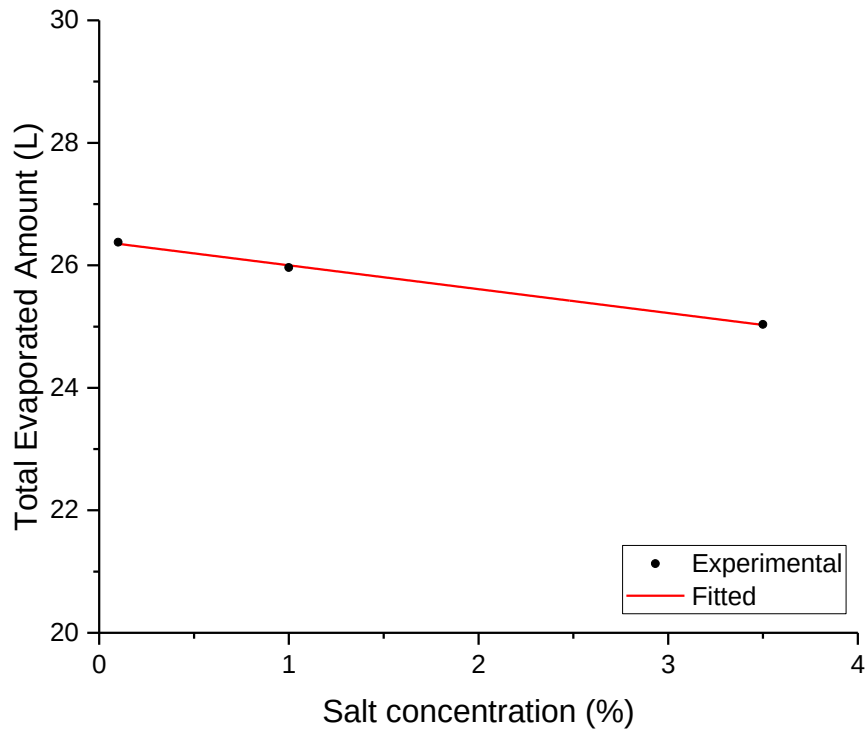


Figure 5.7: Evaporation Amount (in 48 hours) as a function of salt concentration

5.1.2. Continuous Operation with Heating in Evaporation Chamber

For the continuous operation, the NVD unit was filled with water and then all the valves were closed except for the freshwater outlet valve, brine withdrawal valve and the feed line valve. Once the water in the three lines reached the required height to balance the atmospheric pressure, the heater in the evaporation chamber was turned on. The feed water line valve and the brine withdrawal valve were left open to allow the continuous operation. However, the freshwater outlet valve was closed, to allow monitoring of the freshwater production rate. The experimental observations for the continuous operation with heating in the evaporation chamber are shown in Fig. 5.8 – 5.12. The temperatures at different heights measured by the temperature probe inserted in the evaporation chamber are

shown in Fig. 5.8 and Fig. 5.9 for set temperatures of 45 °C and 70 °C, respectively. A temperature gradient between the top and bottom surface of water in the evaporation chamber can clearly be seen. This is similar to the observation from the batch operation and can be attributed to the buoyancy in the water column. Heat losses from the bottom of the evaporation chamber could also be contributing to this behavior.

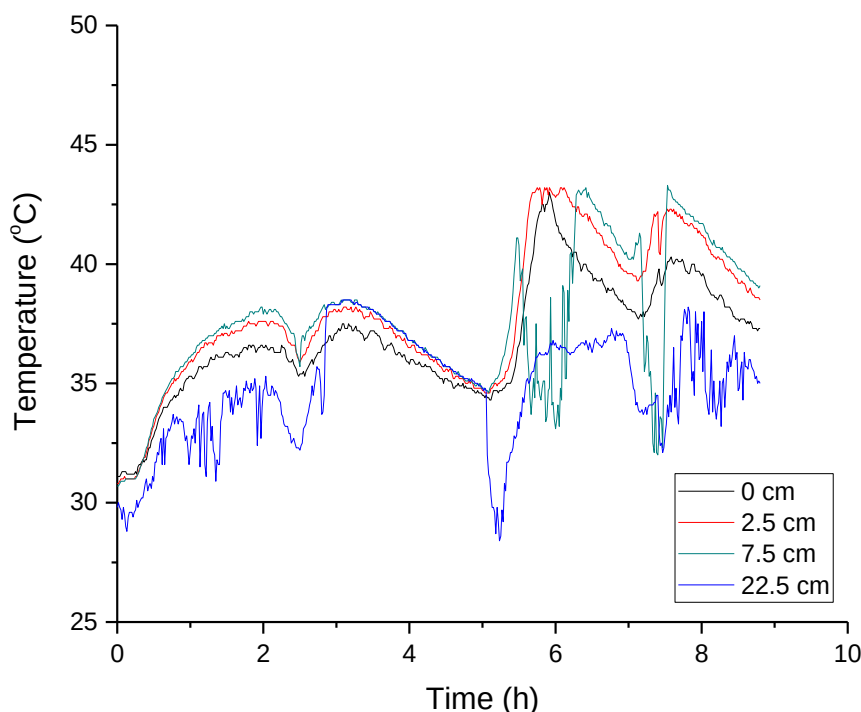


Figure 5.8: Temperatures recorded at different depths in the evaporation chamber at 45 °C set temperature

Notable drops in temperature values can be seen at certain times in Figs. 5.8 – 5.10. This is due to the operation of the withdrawal pump, which was used to withdraw water when the outlet was blocked by bubbles. The sudden withdrawal meant that the evaporation chamber was filled by cooler feed water, which causes this sudden temperature drop.

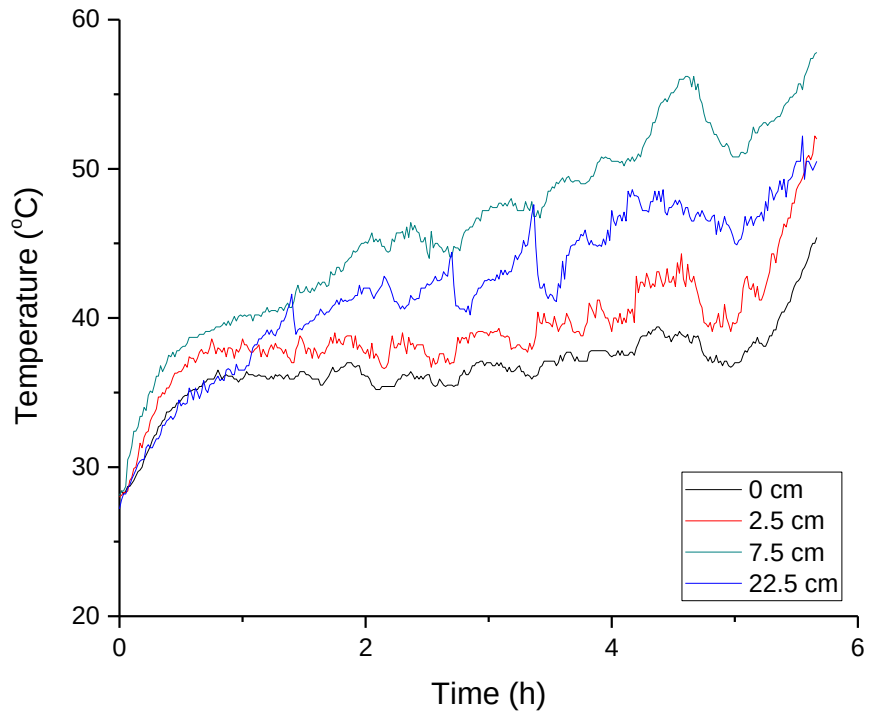


Figure 5.9: Temperatures recorded at different depths in the evaporation chamber at 70 °C set temperature

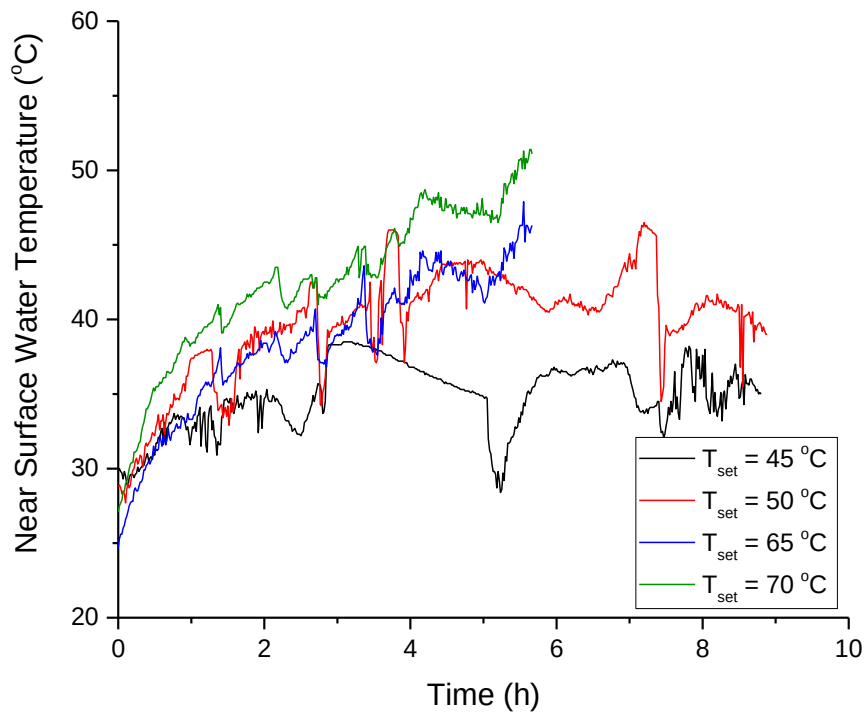


Figure 5.10: Near surface water temperature in the evaporation chamber at different set temperatures

A noticeable difference from the batch operation here was the absence of the valley that was seen in the temperatures recorded in the batch operation. This behavior was attributed to bubble formation around the heater, which caused a drop in the heat transfer rates and hence caused a lowering of the temperature. However, in the continuous process, the movement of water prevented the accumulation of these bubbles around the heater and hence no such phenomenon was observed. It was also seen that the temperature behavior was not as steady as the batch operation. The inlet/outlet of water from the evaporation chamber caused much disturbance in the temperature as can be seen from Fig. 5.10.

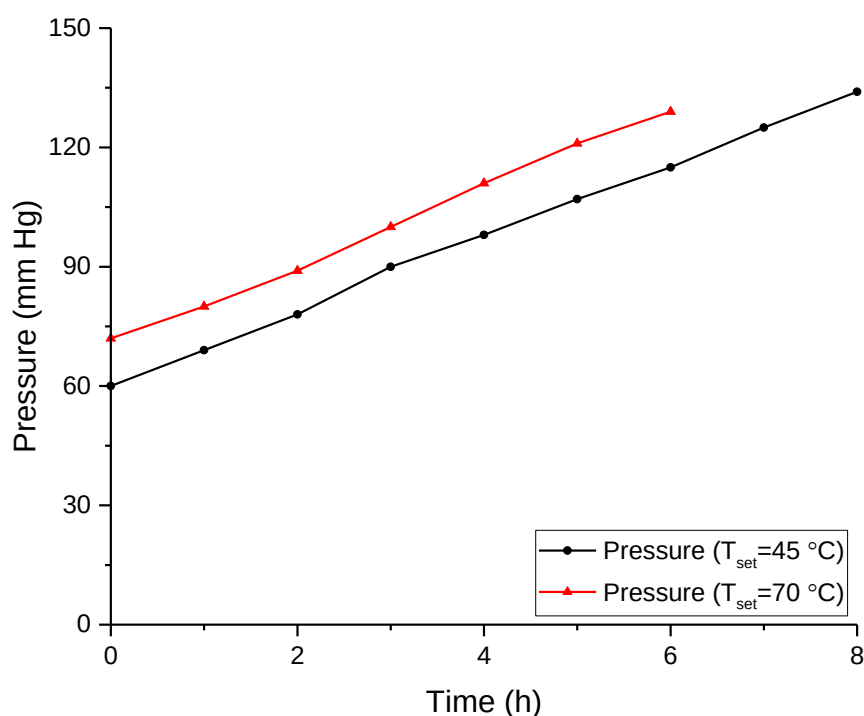


Figure 5.11: Pressure change in the evaporation chamber with time

The system pressure measurements varying with time are shown for two different temperatures in Fig. 5.11. The pressure was observed to increase at a rate of almost 10 mm Hg per hour. This increase could be attributed to the accumulation of water vapor, incondensable gases present

in the feed water and also due to air leaking in to the system. The water production amount with different temperatures is shown in Fig. 5.12. One big problem observed in this operation was that the incoming feed water carried the bubbles formed around the heater away into the brine outlet pipe. These bubbles often caused a blockage in the water flow and caused the flow of water to cease. In the NVD process, the flow of water is similar to a syphon process. Any hindrance in the flow causes the flow to stop permanently. To overcome this issue, a suction pump was added to the brine withdrawal line. The pump was used whenever the brine withdrawal appeared to pause. The pump's suction forced the bubbles to break up and flow was resumed. Abrupt changes in the temperature readings are observed when the suction pump is used, as can be seen in Fig. 5.8 and 5.10. However, this problem was seen as a major issue in the performance of the system. As can be seen from the results, the operation was not carried out for long hours. Due to the frequent interruptions by the bubbles, operation time was limited to 6-8 hours.

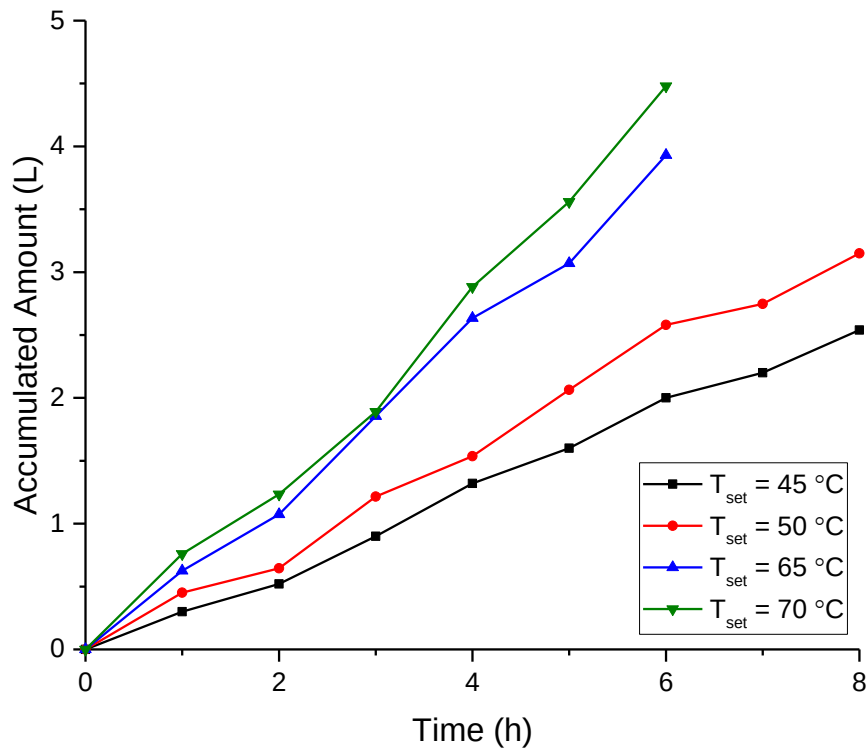


Figure 5.12: Freshwater production amounts at four different set temperatures

5.1.3. Continuous Operation with Heating in Feed Water Storage Tank

To overcome the problem of the bubbles formation discussed in the previous section, it was decided to preheat the feed water in the feed water storage drum. The process is similar to the previous operation, except that the heating is not done in the evaporation chamber. Instead, the feed water enters the evaporation chamber, pre-heated by a heater installed in the feed water drum. Temperatures recorded at different heights in the evaporation chamber are shown in Fig. 5.13. It can be seen that there is no big temperature gradient in the top and bottom of the evaporation chamber. Since, there is no heating inside the tank, the effect of buoyancy is much less than the previous two operations.

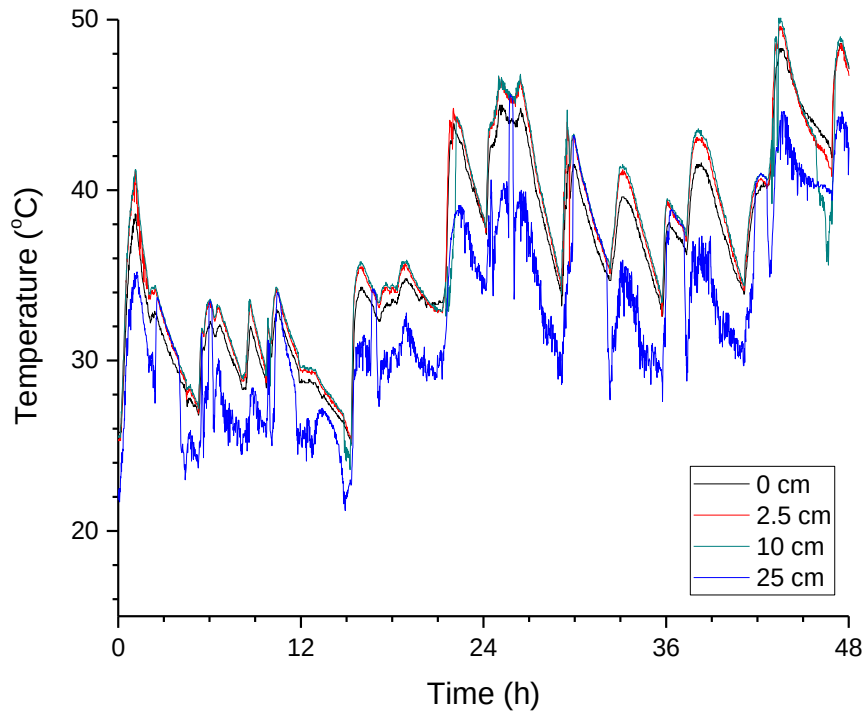


Figure 5.13: Temperatures recorded at different depths in the evaporation chamber

The average evaporator temperature and freshwater side temperature is shown in Fig. 5.14. It can be seen that the average temperature in the evaporator is significantly lower than the set point temperature in the feed water drum (50 °C). This is due to the heat losses in the storage drum as it was not well insulated. The pipes carrying the water to the evaporation tank were also not insulated. The freshwater side temperature also seems to be affected by ambient temperatures and it can clearly be seen to rise during the sunlight hours.

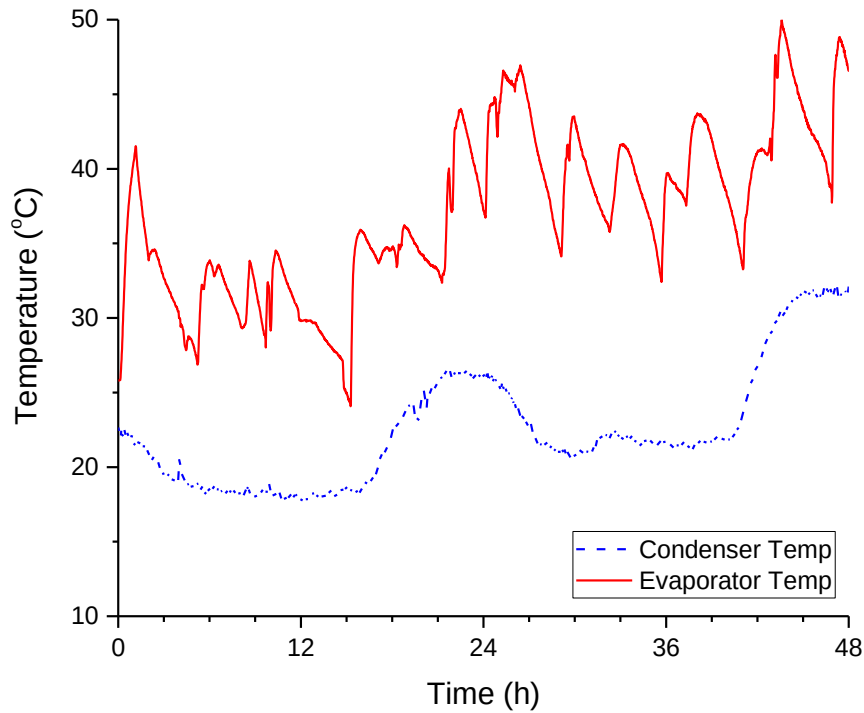


Figure 5.14: Average temperatures recorded in the Evaporator chamber and the Freshwater chamber

The system pressure measurements varying with time are shown for temperatures of 50 °C and 70 °C, respectively in Fig. 5.15. The pressure was observed to increase at a rate of almost 10 mm Hg per hour. This increase is similar to the pressure increase observed in the previous mode of operation. However, since the process was intended to run for a longer time, a vacuum pump was employed to lower the system pressure when it reached 150 mm Hg. The vacuum pump was operated for around 5-10 minutes to bring the pressure down close to the initial pressure. The accumulated amounts of freshwater are shown for different temperatures in Fig. 5.16. The average production rates were 9.05 L/day and 15.65 L/day for operating temperatures of 50 °C and 70 °C, respectively. For the continuous operation, the freshwater outlet valve was closed and only the change in the condensed amount was observed. Evaporation rates were not monitored in these cases.

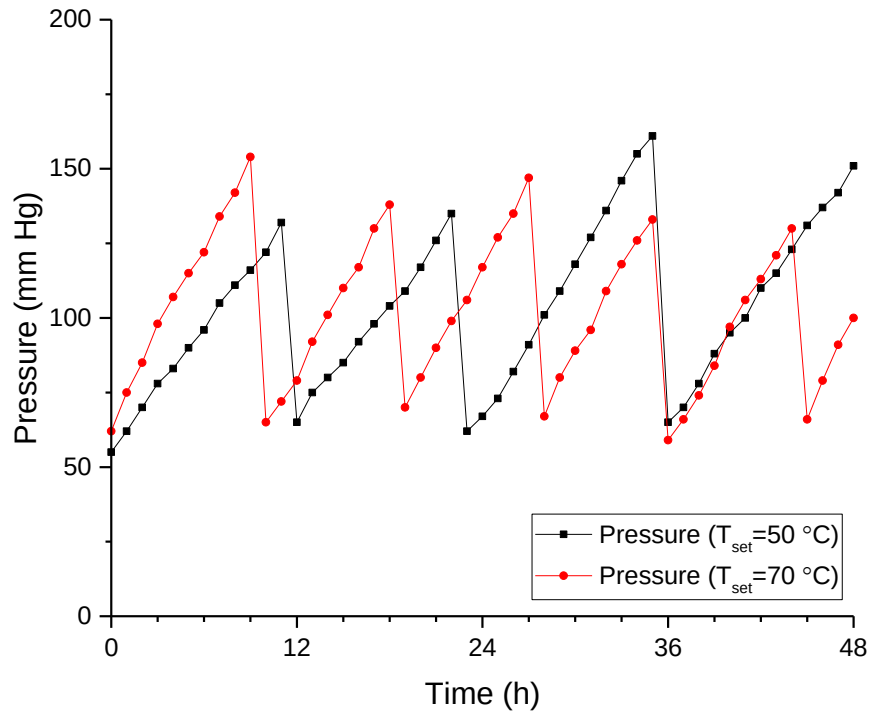


Figure 5.15: Pressure change in the evaporation chamber with time

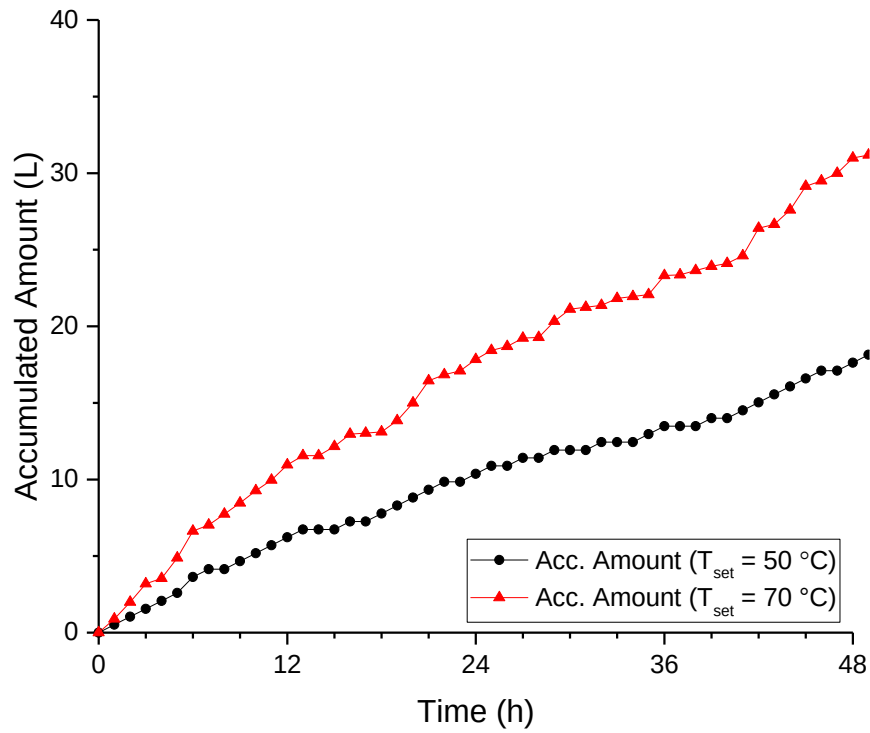


Figure 5.16: Freshwater production amounts at different set temperatures

5.2. COMSOL Model

The NVD process was modeled using COMSOL to simulate the desalination process. The general procedure for using COMSOL for NVD simulation has been presented in Section 3.3. To simulate the experimental procedure, some of the physical characteristics and parameters that were defined in the COMSOL model are shown in Table 5.2. These parameters were used in defining the geometry and other properties of the system.

Table 5.2: Values of the parameters used in the COMSOL model

Name	Expression	Value	Description
dc	65[cm]	0.65 m	Evaporation chamber diameter
dp	0.5[in]	0.0127 m	Inlet/Outlet pipe diameter
Ac	$\pi \cdot dc^2/4$	0.33183 m ²	Area of evaporation chamber
Ap	$\pi \cdot dp^2/4$	1.2668E−4 m ²	Area of pipe
Tf	20[degC]	293.15 K	Freshwater side temperature
mb	1[kg/h]	0.0002778 kg/s	Brine withdrawal rate
Tin	60[degC]	333.15 K	Saline water inlet temperature
Sp	15[cm]	0.15 m	distance between inlet and outlet pipes
hc	35[cm]	0.35 m	Height of water in the evaporation chamber
alpha	2.4e-6	2.4E−6	Mass accommodation factor

The model also requires equations for determining the thermophysical properties of water. These were defined as “variables” in the COMSOL. These variables are shown in Table 5.3. The equations for the vapor pressures and the latent heat of vaporization were used as reported by Bemporad (Bemporad, 1995), instead of the equations mentioned in

chapter 3. This was done only for the sake of simplicity after making sure that the values given by the different equations give similar values in the range of temperatures used in the system. For other thermophysical properties of seawater, the equations mentioned in chapter 3 (Sharqawy et al., 2010) were used.

Table 5.3: Definitions of the variables used in the model

Name	Expression	Unit	Description
Pvapf	$100[\text{Pa}] * \exp(63.042 - 7139.6[\text{K}]/T_f - 6.2558 * \log(T_f * 1[1/\text{K}]))$	Pa	Freshwater vapor pressure
Pvaps	$100[\text{Pa}] * \exp(63.042 - 7139.6[\text{K}]/e_{ave}(T) - 6.2558 * \log(e_{ave}(T) * 1[1/\text{K}]))$	Pa	Saline water vapor pressure
me	$\alpha * A_c * (P_{vaps}/(e_{ave}(T))^{0.5} - P_{vapf}/T_f^{0.5})$	kg/s	Evaporation rate
Hv	$1000[\text{J/kg}] * (3146 - 2.36[1/\text{K}] * e_{ave}(T))$	J/kg	Latent heat of vaporization

The evaporation chamber was modeled as a simple cylinder with a diameter of d_c and a height of h_c as specified in Table 5.2. This is shown in Fig. 5.17. As discussed earlier, the evaporation chamber is the main component of the NVD system and it is the evaporation rate that decides the product amount. It is assumed that the entire vapor produced is condensed on the freshwater side. This is also ascertained by the results of the experimental analysis.

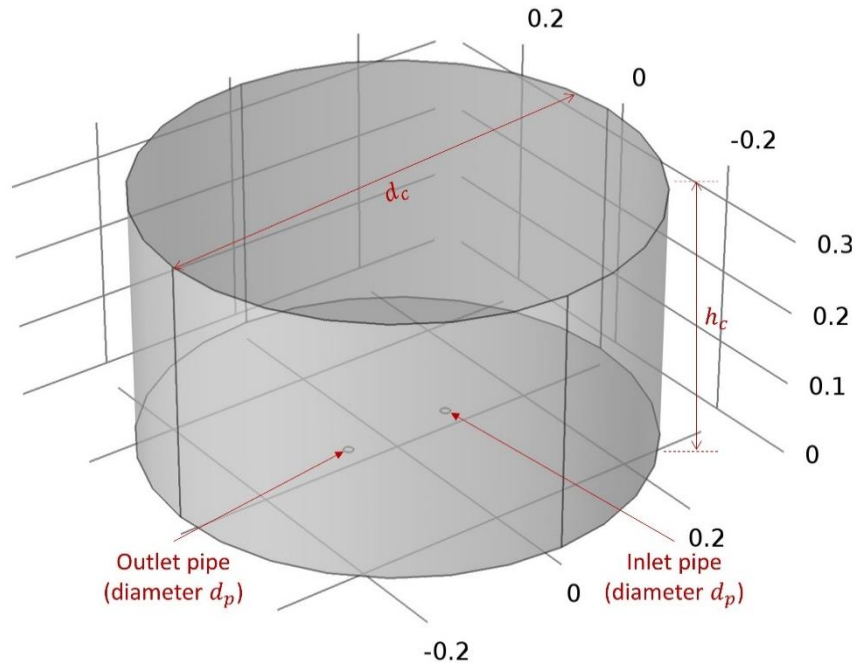


Figure 5.17: Geometry of the evaporator section defined in COMSOL

The boundary conditions were added to the geometry as has been discussed in chapter 3. The initial condition for velocity was taken as 0 m/s and the initial temperature was assumed to be at 30 °C for all simulations. The problem was solved using a free tetrahedral mesh. Due to the involvement of the volume force in the problem, a fine mesh was needed. The mesh consisted of 64217 tetrahedral, 64 pyramid, 10876 prism, 6186 triangular, 160 quadrilateral, 236 edge and 16 vertex elements. The average element quality was 0.62 with a maximum size of 0.0458 m. Boundary layer meshing refinement was used for layers close to the edges. Similarly, the inlet and outlet pipe cross-sections were refined to improve the accuracy of the results in those narrow regions. The mesh used in the analysis is shown in Fig. 5.18.

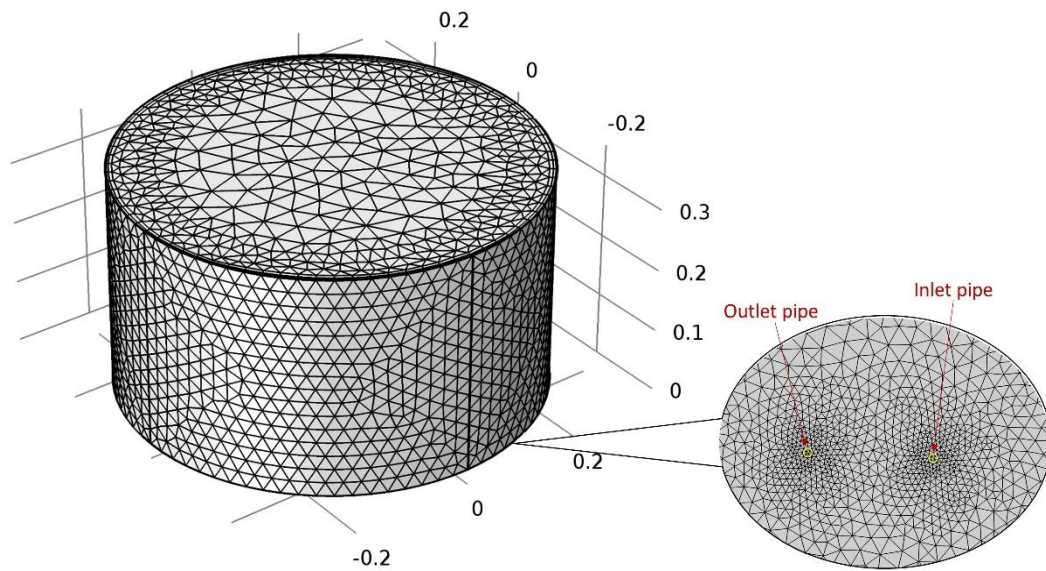


Figure 5.18: Meshing of the evaporator section showing refinements in the inlet/outlet regions

The simulation was run for a 15 days period. The computation time on a computer with an Intel(R) Core(TM) i7-4790, 3.60 GHz, 4 cores processor and 32 GB of memory was 188 hours, 29 minutes and 2 seconds (approximately 8 days). The continuity equation, energy equations and the momentum equation along with the concentration distribution equation and the evaporation rate were solved using a fully coupled solver. A backward differentiation formula (BDF) was used for the time stepping to solve the partial differential equations.

Other than the pre-defined variables that COMSOL solves for according to the physics modules used, temperature probes were added to the problem as well. These temperature probes were defined as “point probes” and located at the corresponding positions as the physical temperature probes used in the experimental system, as shown in section 4.2.4. “Surface average” couplings were also defined to calculate the

average of the variables (temperature or velocity) at the evaporation surface and the brine outlet boundary.

5.3. Model Validation

The simulation results were compared with the experimental results for validation of the model. The results of the batch operation and continuous operation were both validated using settings similar to the experimental conditions. The results are shown in Figs. 5.19 – 5.22.

5.3.1. Batch Operation

The model results were found to closely resemble the experimental results. A comparison of the experimental and simulated results for an operating temperature of 50 °C is shown in Figs. 5.19 – 5.20. The comparison of the water temperature measurements from the experimental observations and the COMSOL model is presented in Fig. 5.19. The initial steep rise in the experimental values as seen in Fig. 5.19 is due to the fact that an electrical heater was used to heat the water in the experiment. The COMSOL model uses a Temperature boundary value for the heat source hence we see a relatively slower rise in the temperature. However, the model readings settle to the experimental recorded values with passage of time. The fluctuations in the experimental readings are due to the on/off effect of the heater. During the initial phase, some the temperatures do not coincide due to the reason stated, but this does not affect the model output by much because we can see that the steady state temperatures are similar with an error of ± 1 °C.

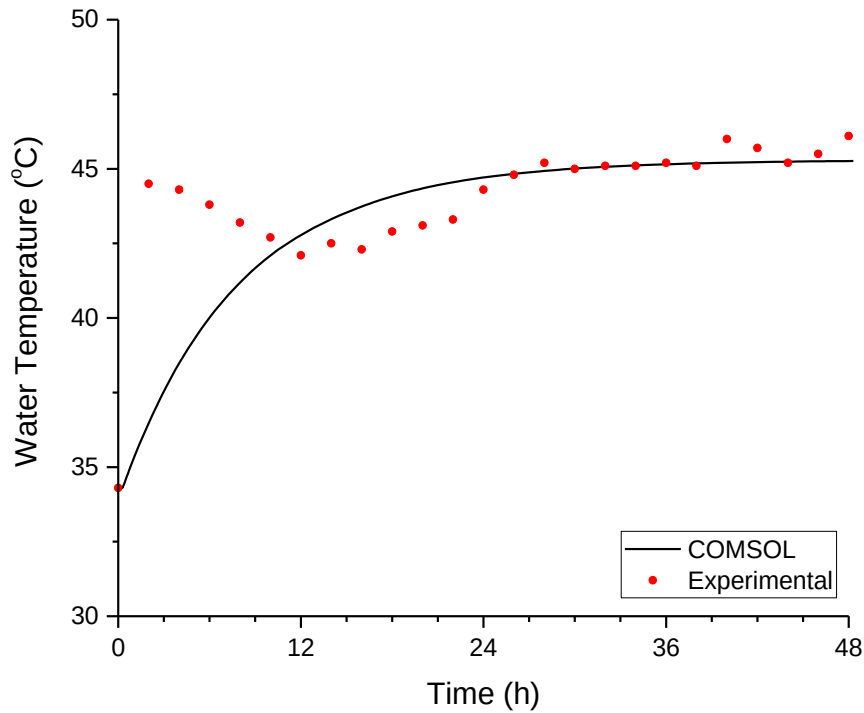


Figure 5.19: Comparison of water temperature from experimental and simulated results during batch operation

In addition, we can see that the freshwater production amounts were almost similar from experimental observations. A comparison of the accumulated amounts measured experimentally and calculated from the simulation is shown in Fig. 5.20. Due to the sudden rise in temperature in the experiment as compared to the simulation, we can see that the initial evaporation rates are higher in the experiment as compared to the simulation. Over the passage of time, the evaporation rates become similar and the rate of increase in the accumulated amounts is same but the simulated results suffer an offset that was caused due to the initial difference of temperatures. A steady state error of almost -4 kg is shown by the model. This can be corrected by modeling an electrical heater in the COMSOL simulation, but that was not done to save computational time.

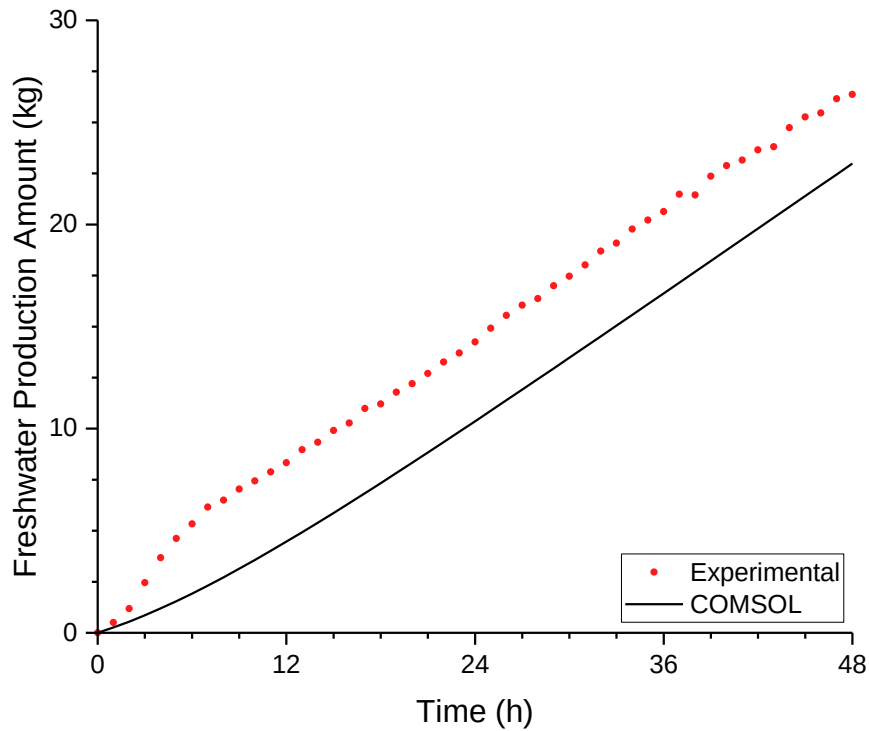


Figure 5.20: Comparison of freshwater production amounts from experimental and simulated results during batch operation

5.3.2. Continuous Operation

The simulation results for the continuous operation matched the experimental observations in a similar fashion to the batch operation. Initially, the heater used in the experimental study causes the temperature to rise quickly and the temperature rises much steeply than the prediction of the COMSOL model. The reason is the same as mentioned earlier. As the time passes, the temperature of water, as observed from the experimental results, is scattered around the predicted temperature from the COMSOL model as seen in Fig. 5.21. The points in the figure represent hourly temperature values, which had been measured continuously hence they appear as scattered data.

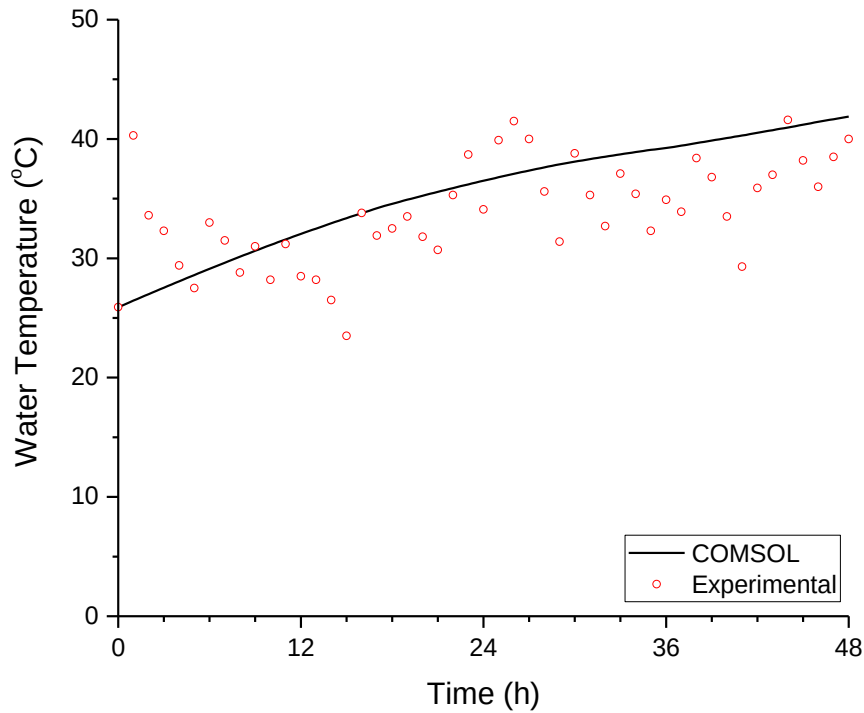


Figure 5.21: Comparison of water temperature from experimental and simulated results during continuous operation

A behavior similar to the batch operation can also be seen for the freshwater production amount in the continuous operation. As seen in Fig. 5.22, due to the use of the heater, a steep rise in the temperature in the experiment causes rapid evaporation of the water. This initial behavior is not replicated in the simulation because a temperature boundary condition has been used, which does not provide this sudden heat flux. A steady state error of almost -4 kg can again be seen here in the model predictions. However, it can be seen that as the time passes, the simulation predicts the evaporation rate more accurately. This can be seen from the similar slopes of the two lines for the experimental and simulated results.

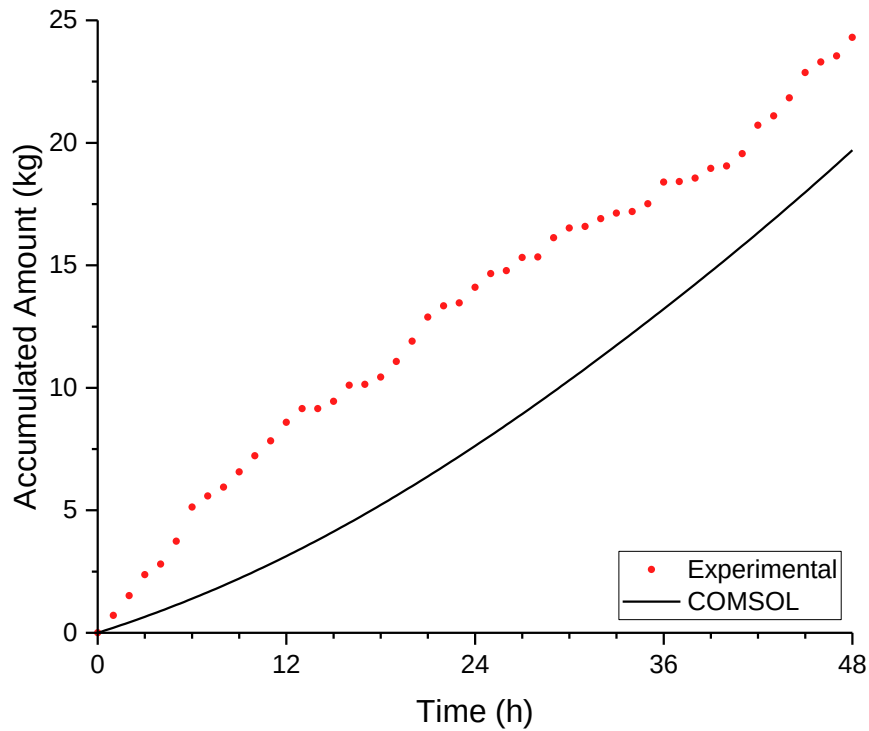


Figure 5.22: Comparison of freshwater production amounts from experimental and simulated results during continuous operation

From the above discussion, it can be concluded that the developed model was able to predict the results with reasonable accuracy for the observations tested. Hence, it can be trusted to predict the results with precision for different conditions.

5.3.3. Simulation Results

The simulation was carried out to get a better understanding of the NVD process. The results for the simulations of the batch and the continuous operation are discussed here.

The temperature variation in the evaporator section was analyzed by selecting an imaginary plane around the heater. The surface plots represent the temperature variation in a horizontal plane at a height of 0.05 m from the bottom as shown in Fig. 5.23. The arrow vectors represent the velocity

field. This plane was chosen to show how the temperature varies along a plane around the heater. Temperature variations are shown for different times (0, 0.1, 0.5, 1, 2 and 7 days, respectively) as shown in Fig. 5.24.

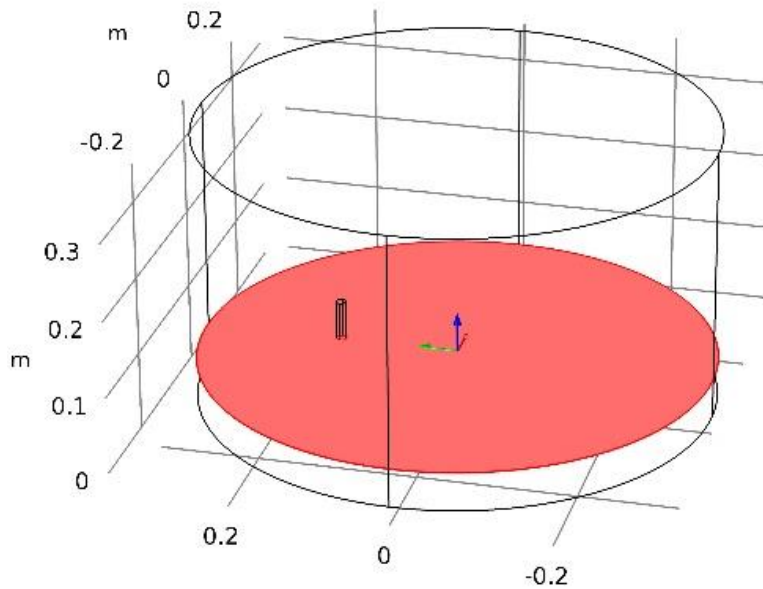


Figure 5.23: The shaded area indicates the plane selected for the temperature and velocity measurements in the evaporation chamber

The temperature rises quickly in the beginning. It then settles at a certain temperature value that is lower than the heater temperature. Important thing to note here is that the temperature is almost uniform throughout the plane except the peak seen around the heater. It can be concluded from this that the heat transfer is dependent mainly on convection here rather than conduction. If we see the velocity vectors, we can see that there is a lot of water movement in all directions due to the buoyancy forces. This causes turbulence and hence a uniform distribution of the temperature is seen. A similar phenomenon can be seen in any horizontal plane at any given height. This particular plane was selected because the heater lies within this plane and the temperature distribution

can be expected to be more aberrant here than in the planes above the heater.

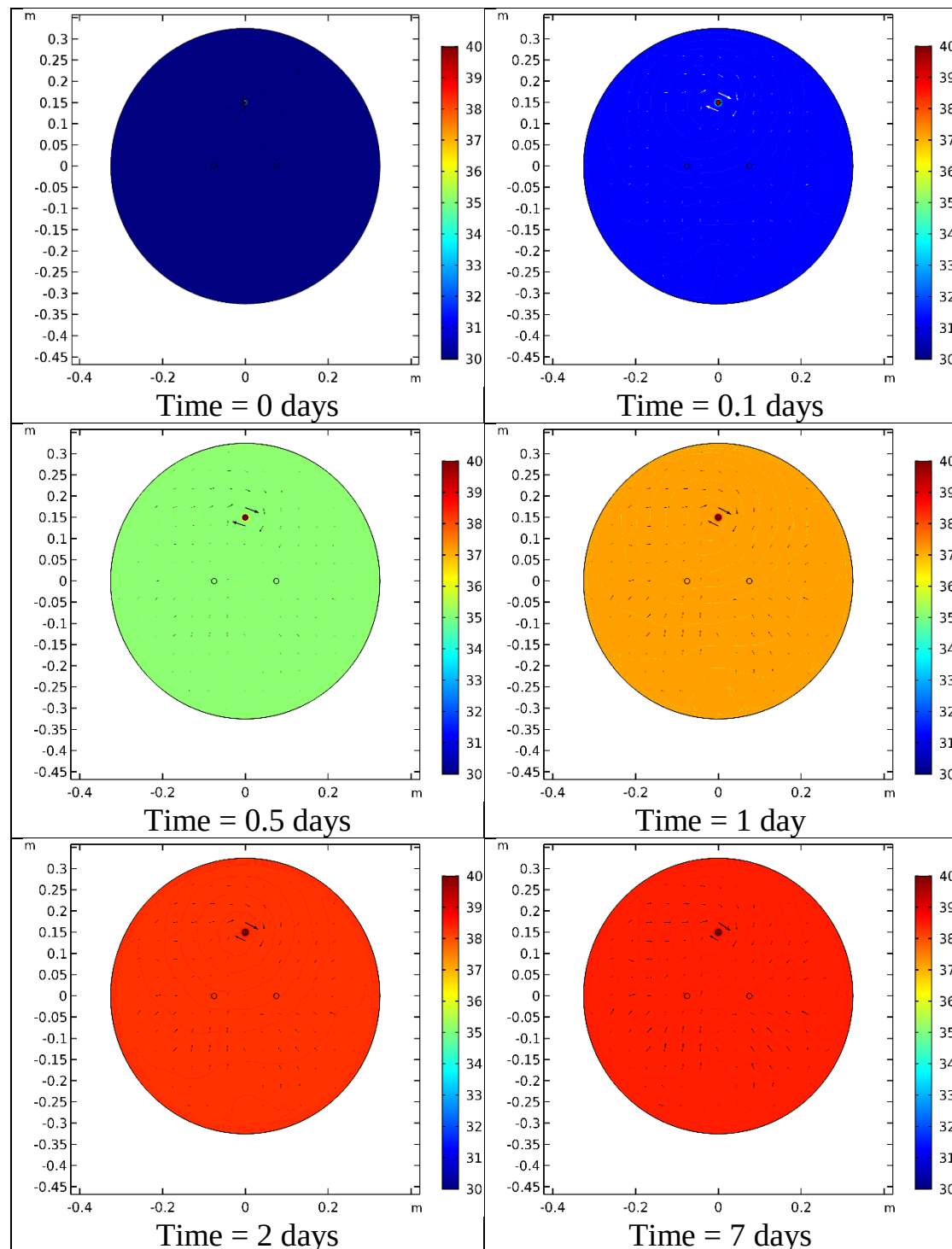


Figure 5.24: Temperature and velocity profiles at different times in a horizontal plane in the evaporator

The temperature distribution along the centerline running through the top surface of the water in the evaporation chamber is shown in Fig. 5.25. It can be seen that although the temperature appears to be uniform, there is nonetheless a slight variation in the temperature. This difference is not very prominent in Fig. 5.24, where the color range does not show much variation in the slight temperature difference that exists in the plane. The temperature can be seen to be lower at the stagnant regions of the chamber near the walls. It is higher at the areas closer to the water heater. This difference between the higher and lower temperature reduces with time.

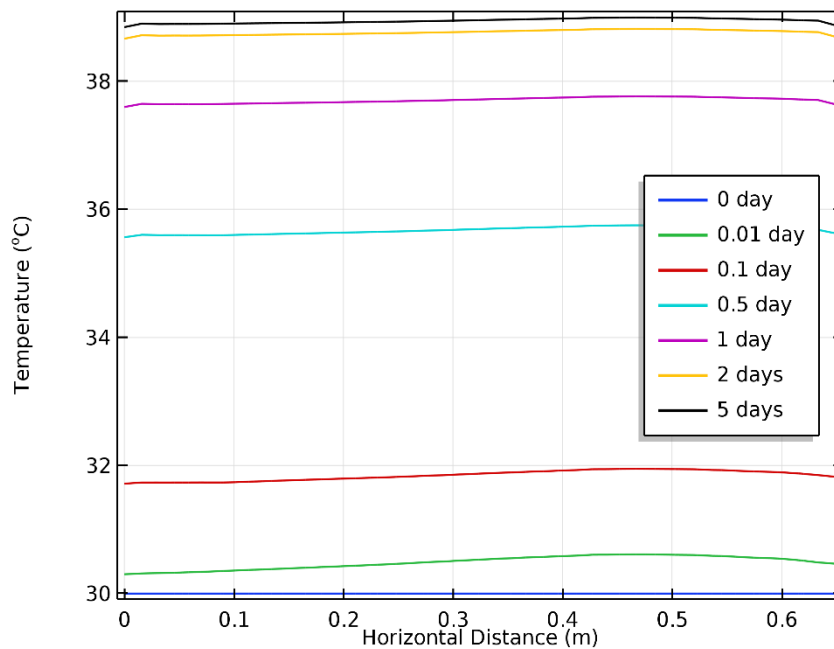


Figure 5.25: Temperature distribution along the centerline at different times on the top surface in a horizontal plane in the evaporator

The temperature distribution was also analyzed in the vertical direction by selecting a plane at the center of the evaporation chamber that passes through the heater. This plane is shown in Fig. 5.26.

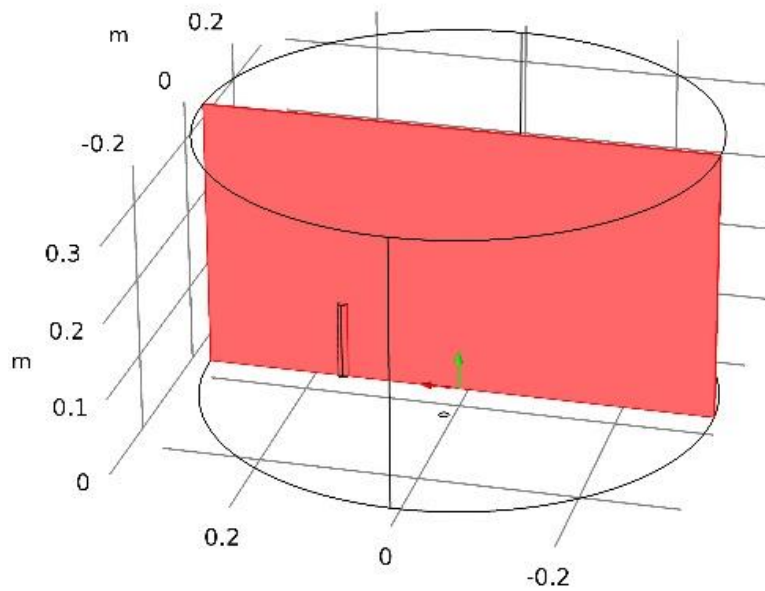


Figure 5.26: The shaded area indicates the vertical plane selected for the temperature measurement in the evaporation chamber

Just like the temperature distribution in the horizontal plane, the temperature distribution in the vertical plane is shown at different times (0, 0.1, 0.5, 1, 2 and 7 days, respectively) in Fig. 5.27. The first order behavior of the system is obvious from these figures as well. The temperature can be seen to vary quickly at first and then it settles to a certain steady state value. The temperature around and above the heater rises much more than the other areas. Unlike the horizontal plane, the temperature in the vertical plane is not uniform. A temperature gradient can be seen as the hot water rises up and the colder layers move down. This temperature gradient was also seen in the experimental observations.

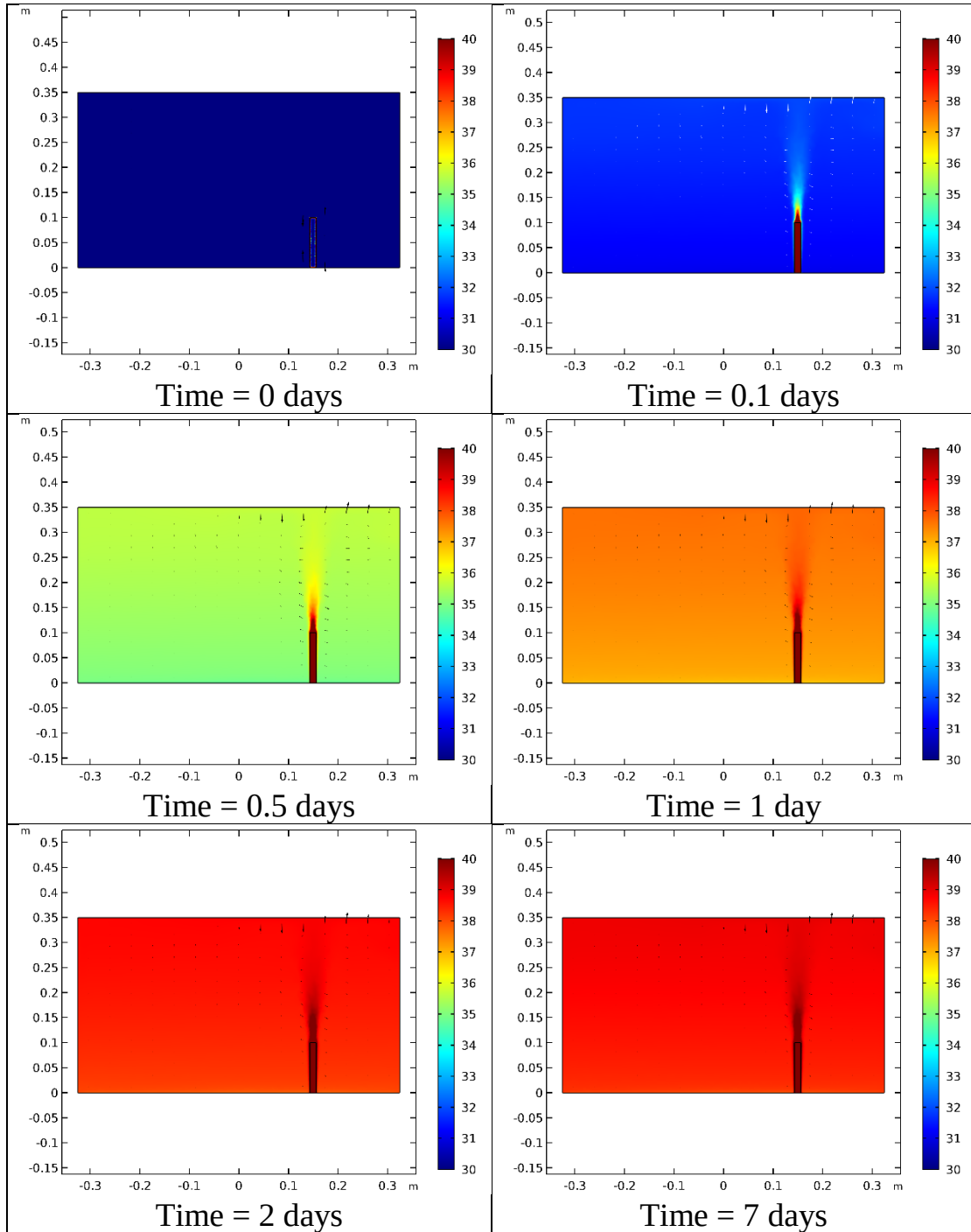


Figure 5.27: Temperature distribution at different times in a vertical plane in the evaporator

As mentioned earlier, the temperature distribution in the horizontal direction is almost uniform with a peak just around the heater. This peak becomes less obvious as we go above the heater. In the top layers, the

temperature is almost uniform throughout. This vertical temperature gradient is also shown by the plot in Fig. 5.28. This figure shows the temperature along a vertical line which runs through the center of the evaporation chamber. It can be seen that at any time there exists a clear temperature gradient in the evaporation chamber. The lower layers have a lower temperature while the top layers have a higher temperature. As explained above, this is due to the heat loss at the uninsulated bottom of the evaporation chamber as well as due to the buoyancy effect in the chamber.

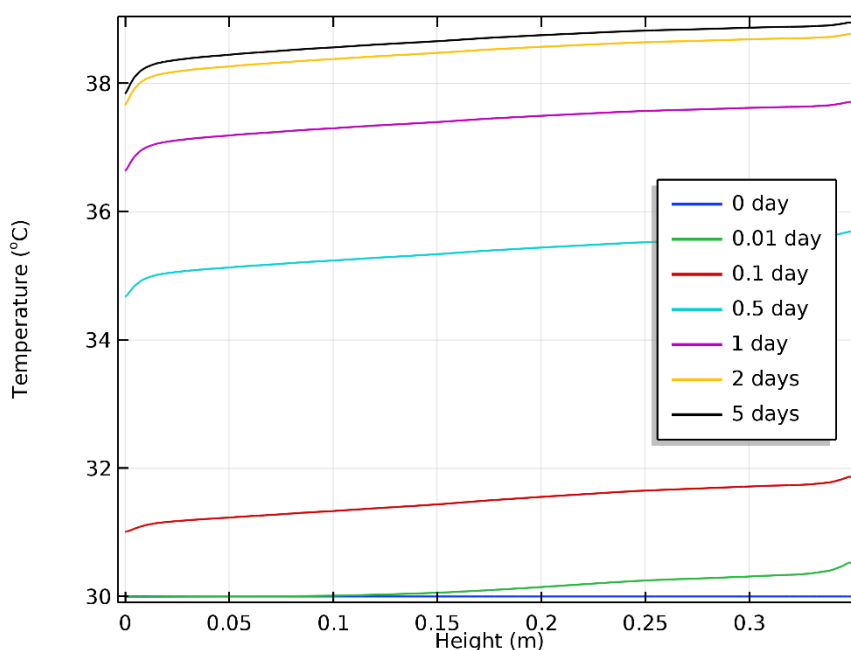


Figure 5.28: Temperature distribution at different times along a vertical line through the center of the evaporator

This temperature gradient becomes most prominent in the area just above the heater. This can be seen in Fig. 5.29. The temperature at the point just above the heater is close to the set point temperature of the heater (50 °C in this case). Initially the difference between the heater and the top layer is large but as the time passes, the water gets heated and we can see that the difference is reduced. The slope of this temperature gradient also

becomes less steep due to this reduced difference. It is also important to note that even after the steady state has been attained, the gradient still exists.

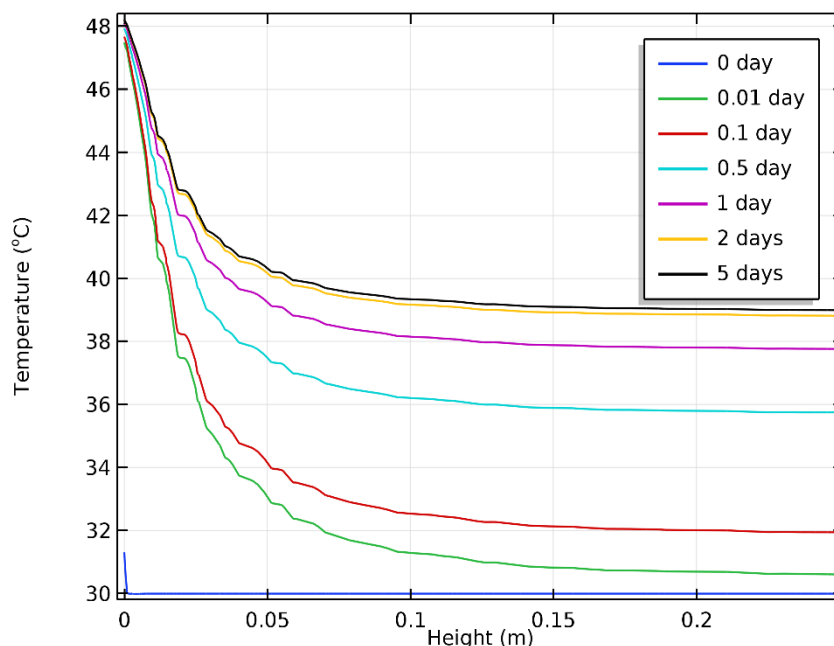


Figure 5.29: Temperature distribution at different times along a vertical line above the heater in the evaporator

The continuous process results are presented now. The continuous process took a much longer time to reach the steady state than the batch process. The temperature and evaporation rate in the continuous process are shown in Fig. 5.30 and Fig. 5.31, respectively. The system reaches a steady state in almost 10 days. The temperature shown by the different lines in Fig. 5.30 is the temperature observed by the temperature probes at various heights in the evaporation chamber. Temperature probes were placed at a distance of 2.5 cm starting from the bottom of the evaporation chamber. In the batch process, we observed a temperature gradient between the lower and higher probes. In the continuous process, we can see that the probes are almost similar to each other. Due to a more enhanced mixing in the continuous process, the temperature is well distributed in all the layers.

Hence, no temperature gradient is seen in the continuous process in the probes. A slight bump can be seen at some places on the curve, which is due to the effect of the changes in the ambient temperature.

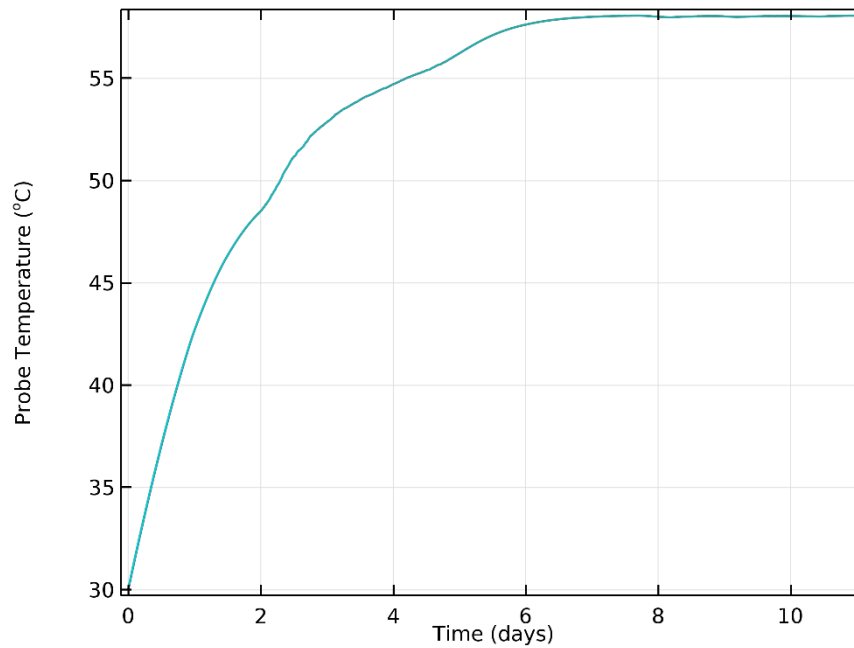


Figure 5.30: Temperature recorded by the probes at various heights in the evaporator

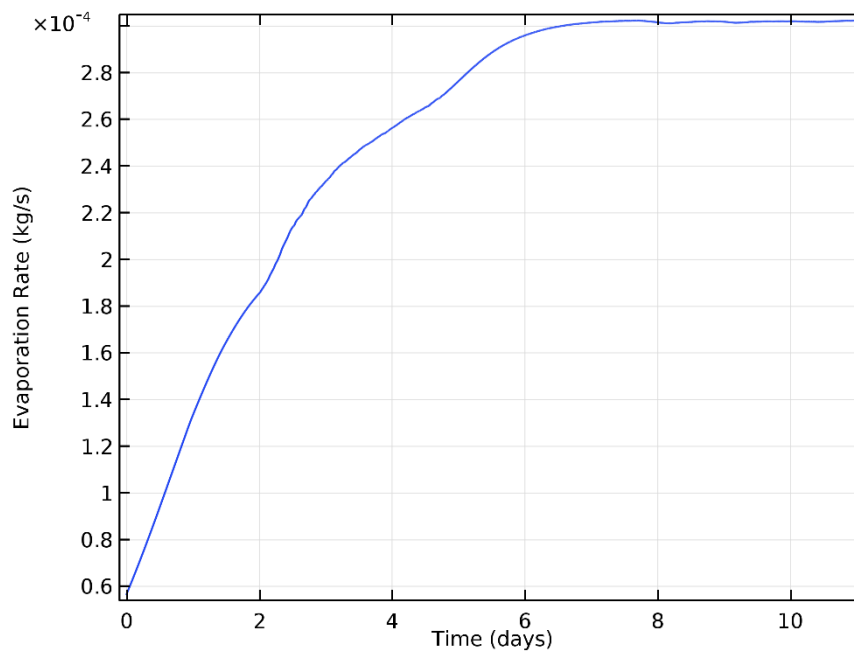


Figure 5.31: Changes in the evaporation rate with time in a continuous operation

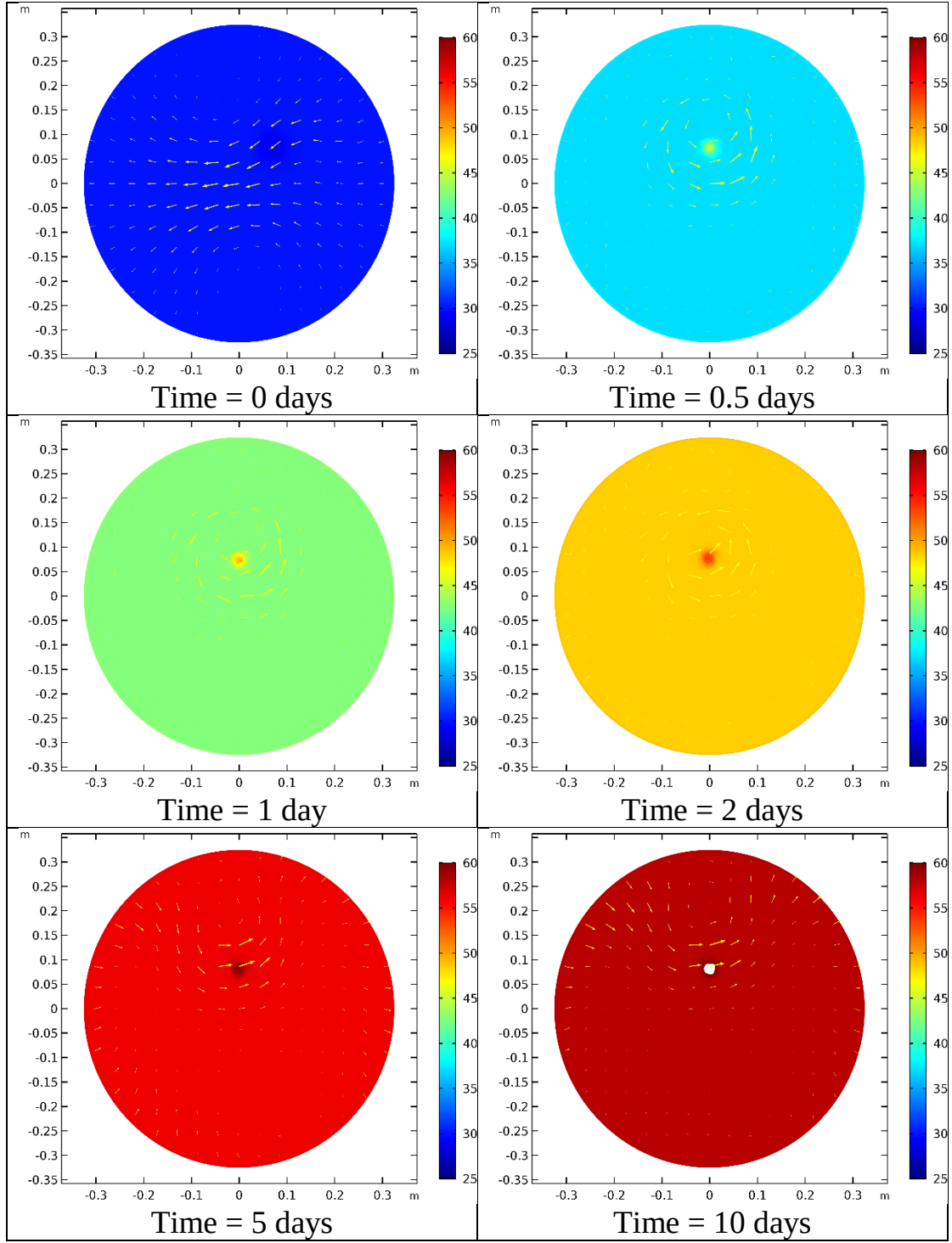


Figure 5.32: Temperature and velocity distribution at the bottom of the evaporation chamber at different times

The evaporation rate in the continuous process is shown in in Fig. 5.31. It seems to be a replica of the temperature plot. This is because the evaporation rate is a strong function of the temperature difference between the evaporation chamber and the condenser side. In this simulation, the

condenser side temperature is maintained constant, hence the evaporation temperature is the driving force for the evaporation. The evaporation rate can be seen to settle around a steady state value of around 3×10^{-4} kg/s (almost 26 kg/day).

The temperature and velocity profiles at the bottom of the evaporation chamber at different time are shown in Fig. 5.32. It can be seen that the temperature is almost uniform throughout. The highest temperature is at the point where the hot feed water enters the evaporation chamber. The effect of this hot temperature is not seen much in the radial direction as the hot water enters vertically. Most of the heat transfer is due to the bulk movement of water inside the tank, and conduction (diffusion) plays a minor role. The velocity vectors can also be seen swirling around the point of the feed water entry. It can also be seen that the velocity here is higher than the other areas in the evaporation chamber. The magnitude of the velocity is shown by the length of the vectors.

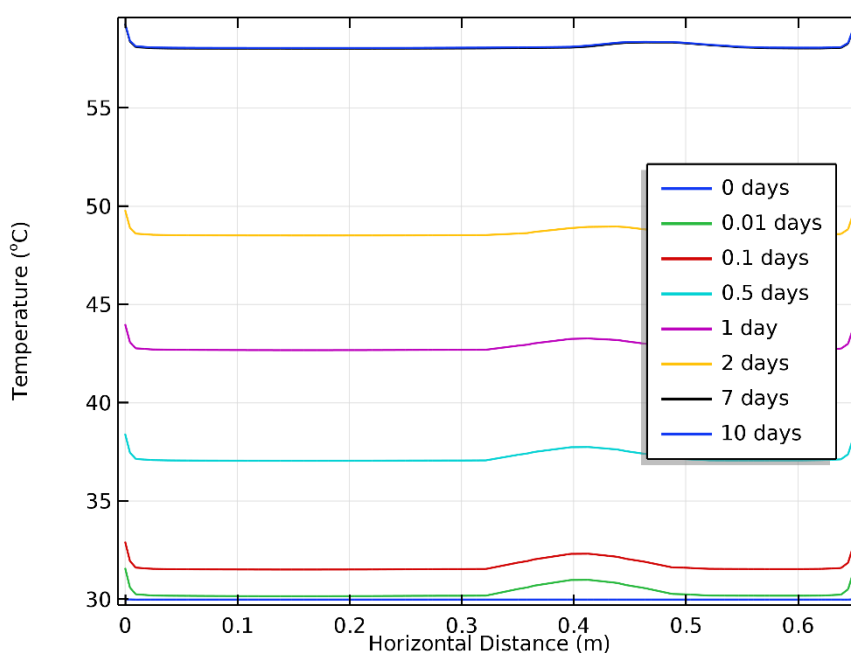


Figure 5.33: Temperature distribution along the centerline at different times on the top surface in a horizontal plane in the evaporator

The temperature at the top surface of the evaporation chamber at different time are shown in Fig. 5.33. Just like the bottom surface of the chamber, the temperature here also appears to be uniform throughout the surface. A slight rise in temperature is seen at the point that is directly above the feed water entry.

The distribution of the temperature and velocity in the vertical direction is observed by selecting a vertical plane at the center of the evaporation chamber. This plane is selected such that it passes through the middle of the feed water inlet pipe. This plane is shown in Fig. 5.34.

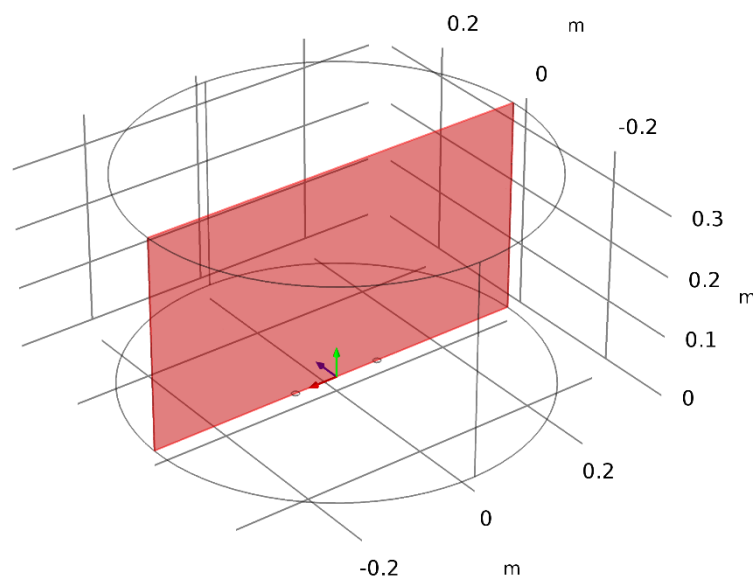


Figure 5.34: The shaded area indicates the vertical plane selected for the temperature and velocity measurement in the evaporation chamber

Temperature and velocity variations are shown for different times (0, 0.1, 0.5, 1, 5 and 10 days, respectively) in Fig. 5.35. Just like the temperature in the horizontal surface, the temperature in the vertical surface also looks uniform. The only exception is the region above the inlet point. As expected, the temperature is higher in this area as compared to the other areas. This vertical plane was selected because it shows the

temperature difference in the areas above the inlet point. In other planes, the temperature distribution is more uniform.

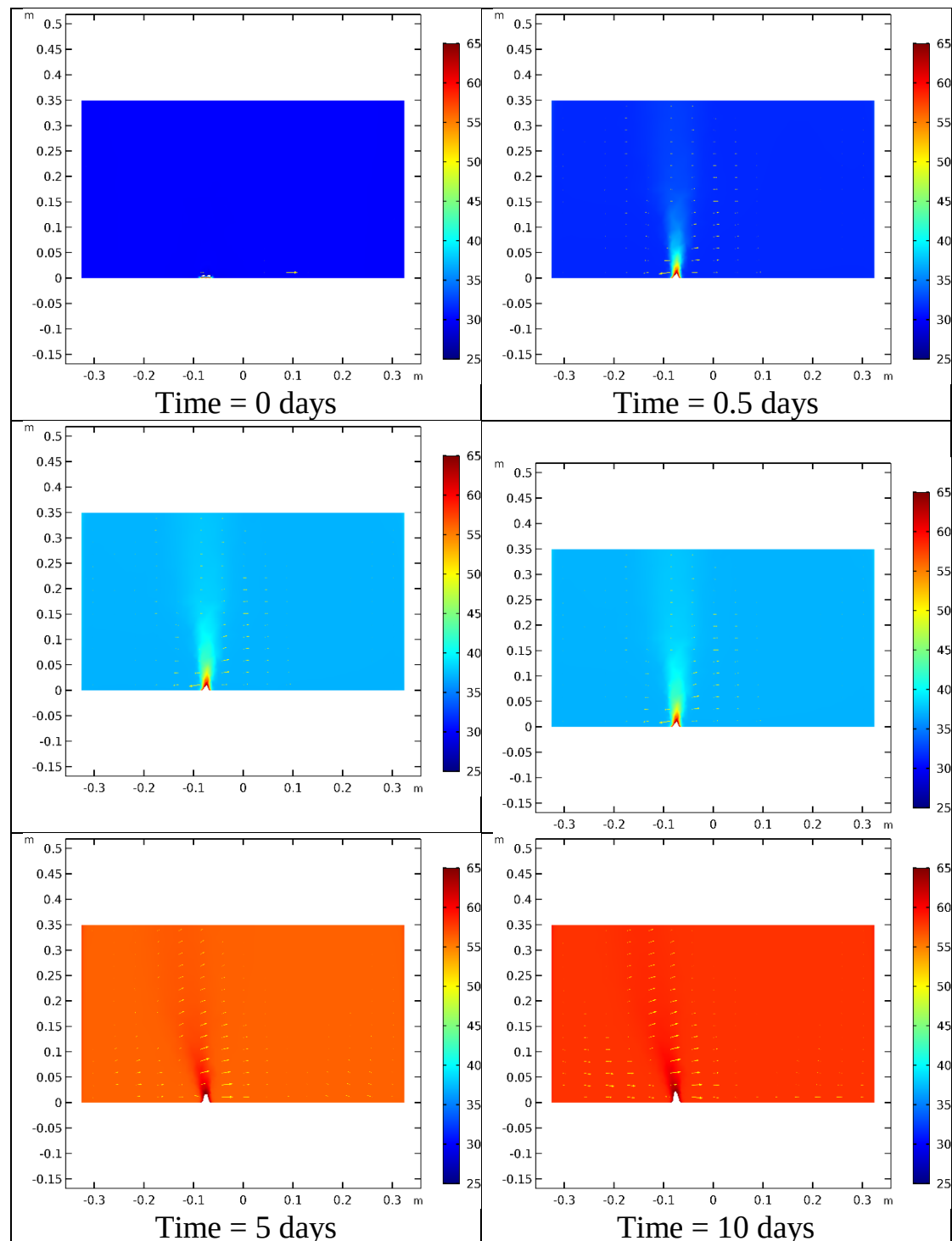


Figure 5.35: Temperature and velocity distribution at the selected vertical plane along the center of the evaporation chamber at different times

The velocity is represented by the arrow vectors in Fig 5.35. The length of the vector indicates the magnitude of the velocity. We can see that the velocity is higher near the water inlet point. Water can be seen to swirl around the inlet point and then from both the left and the right it goes downward into the brine withdrawal pipe.

The temperature distribution in the vertical line passing through the center of the evaporation chamber is shown in Fig. 5.36. It can be seen more clearly here that the temperature is almost uniform in the vertical direction as well.

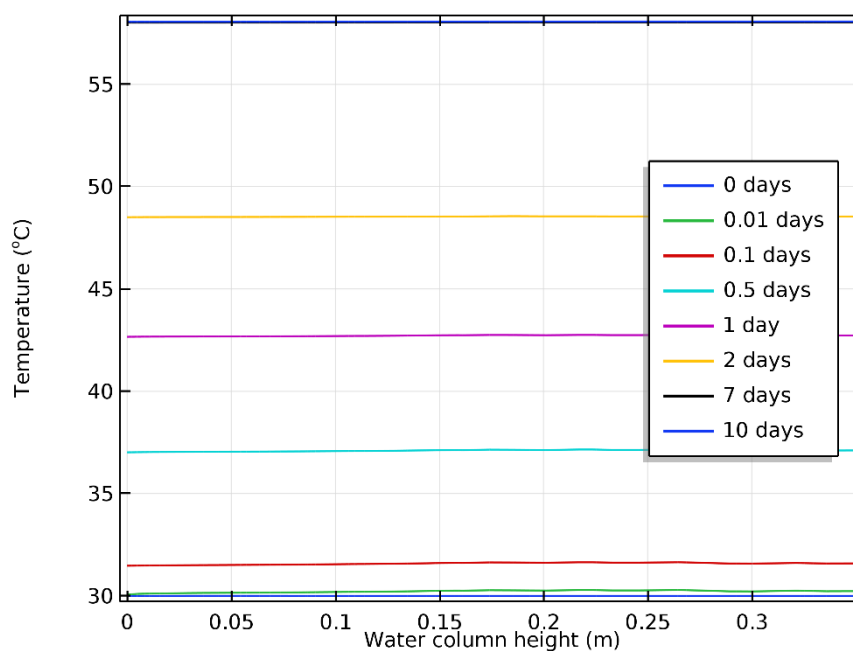


Figure 5.36: Temperature distribution along the vertical line passing through the center of the evaporation chamber at different times

The temperature distribution in the vertical line passing through the center of the feed water inlet pipe is shown in Fig. 5.37. It can be seen that the temperature drops quickly to equalize with the temperature of the water in the evaporation chamber. This is because the inlet is at a small rate whereas the volume of the water in the evaporation chamber is comparatively much larger. The sink in the temperature at time zero below

the initial temperature of 30 °C is just a numerical error that the simulation comes across as it tries to deal with the temperature difference of the inlet and the initial condition. In reality, the temperature would not fall below the initial condition.

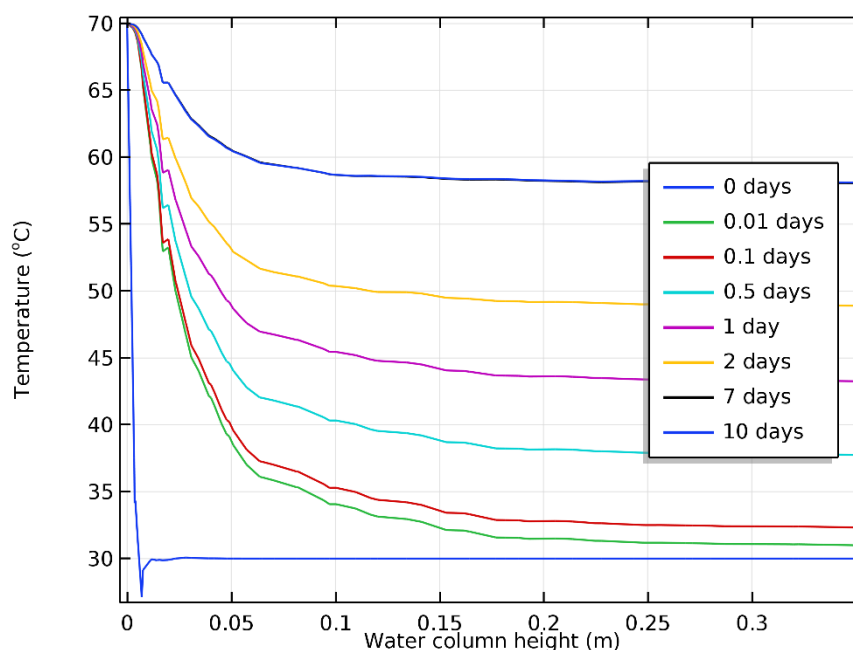


Figure 5.37: Temperature distribution along the vertical line above the seawater feed pipe at different times

5.4. Parametric Studies using COMSOL

The effect of different parameters on the freshwater production rates and other system variables was studied for both the batch and continuous operations. The results of these parametric studies are presented in the coming sub-sections.

5.4.1. Batch Operation

The impact of the evaporation temperature, condenser side temperature and the evaporator size on the production of freshwater was

studied based on the developed COMSOL model. The results shown in Figs. 5.38 – 5.40 are for a study time of 2 days (48 hours).

5.4.1.1. Effect of Evaporation Temperature

The evaporation temperature is perhaps the most important factor that affects the production rate. As can be concluded obviously from Eq. 3.4, higher evaporation temperature will give a higher production rate. The production rate increases in an inverse exponential growth with the increase in temperature. Similar results were reported by previous studies (Al-Kharabsheh & Goswami, 2004; Gude & Nirmalakhandan, 2009). This effect can be seen in Fig. 5.38.

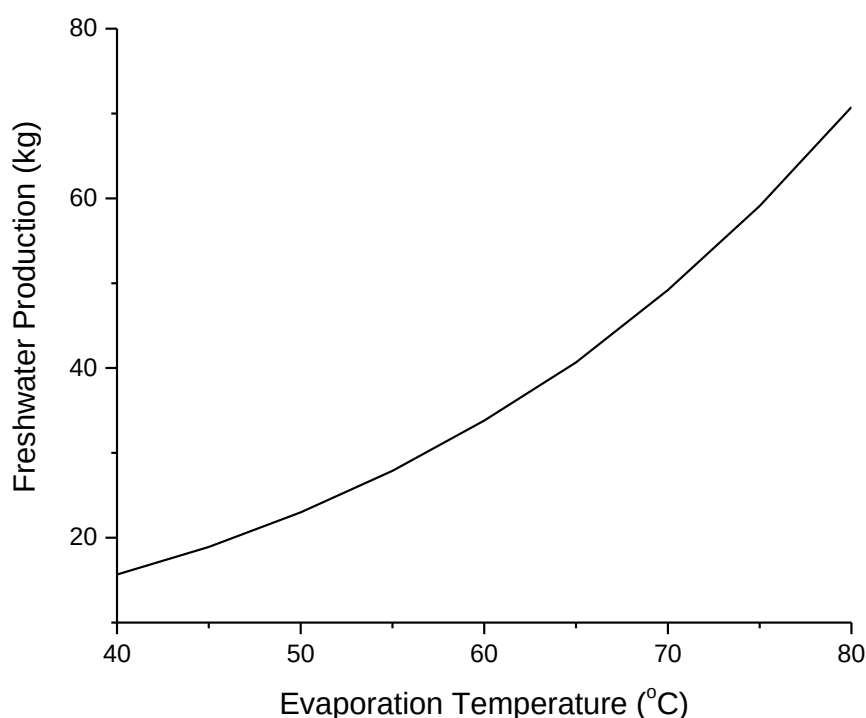


Figure 5.38: Accumulated production amount as a function of the evaporation temperature

A general conclusion would be that higher temperatures are more suitable. However, the thermal efficiency of the process has been reported to fall with an increase in the inlet temperature (Gude et al., 2010). It is

thus important to study the cost-effectiveness of raising the inlet temperature and find the optimum operating temperature.

5.4.1.2. Effect of Freshwater Side Temperature

Another important factor in the production of freshwater in NVD processes is the temperature of the cooled desalinated water chamber, or the condenser. The rate of evaporation is a function of the pressure difference between the vapors above the evaporation chamber and in the condenser. A higher temperature on the condenser side will result in a higher vapor pressure and this will in turn lower the evaporation rate. As expected, Fig. 5.39 shows that the lower the freshwater temperature, the higher the condensation rate. If the ambient temperatures are not used for cooling, then using a cooling mechanism will add to the cost of the process. A cost analysis is required to determine the optimum cooling temperature that can be used in the process.

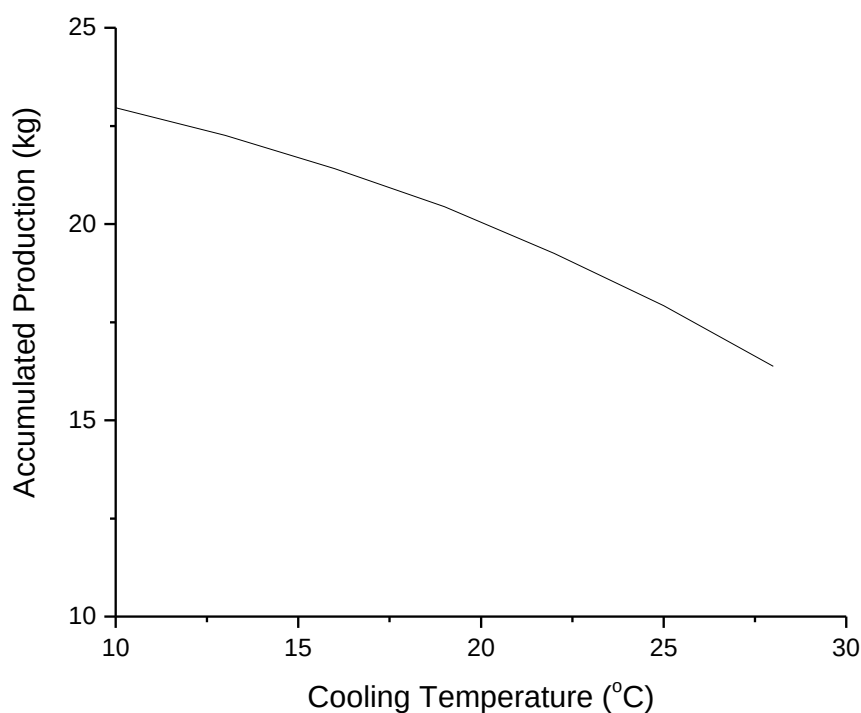


Figure 5.39: Accumulated production amount as a function of condenser side temperature

5.4.1.3. Effect of Evaporator Size

The evaporator size is a key contributor to the overall evaporation rates. Obviously, a larger area will result in higher evaporation rates. To study the effect of the change in surface area, the tank diameter was changed. The results are shown in Fig. 5.40. The relation appears to be somewhat non-linear, as was also reported by Al-Kharabsheh and Goswami (2004).

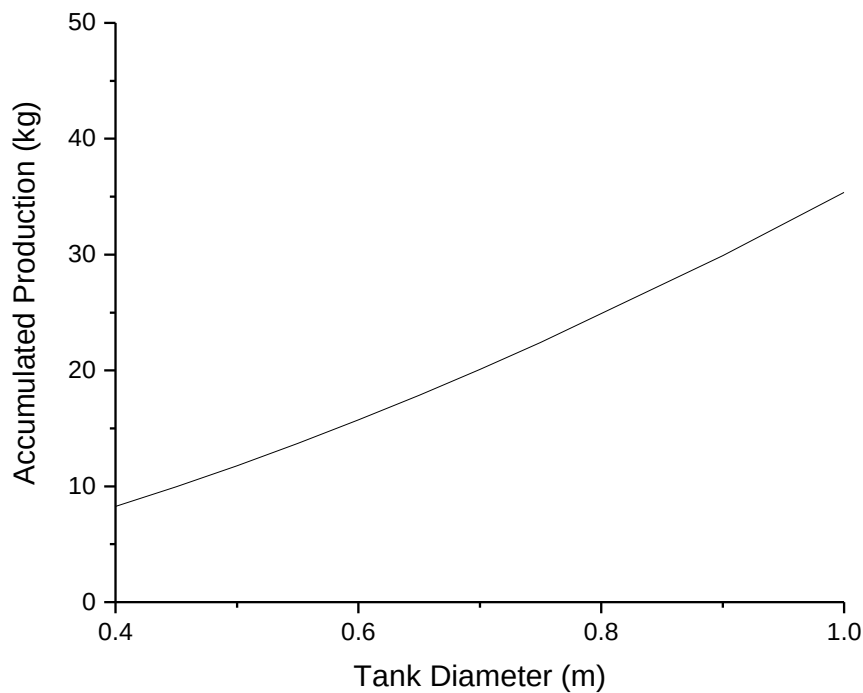


Figure 5.40: Accumulated production amount as a function of evaporator size

5.4.2. Continuous Operation

The impact of the evaporation temperature, condenser side temperature and the evaporator size on the production of freshwater was studied based on the developed COMSOL model. It was seen that the process took approximately 11 days to reach a steady state after any of the

parameters was changed. The results discussed are based on the values at the new steady state condition.

5.4.2.1. Effect of Evaporation Temperature

As was seen from the batch operation, the production rate increases exponentially with the increase in the evaporation temperature. This can be seen Fig. 5.41. The performance of the desalination system is somewhat reflected by the recovery ratio. The recovery ratio (RR) is defined as the ratio between the freshwater produced and the inlet feed seawater. It can be mathematically expressed as:

$$RR = \frac{m_f}{m_s} \quad (5.3)$$

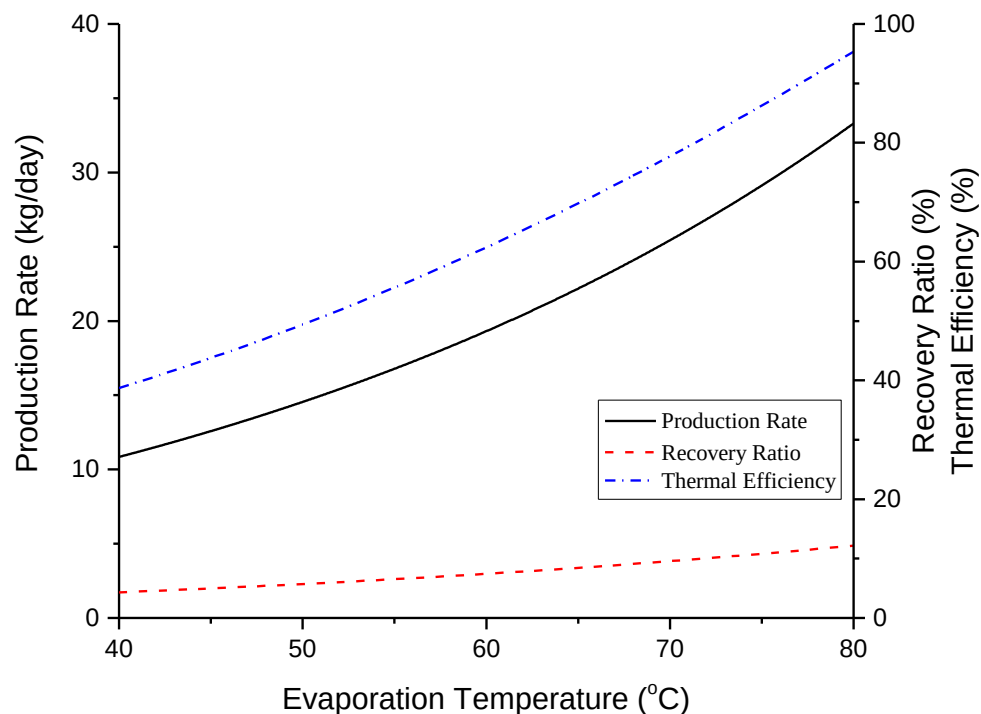


Figure 5.41: Accumulated production amount, recovery ratio and thermal efficiency as a function of the evaporation temperature

The Recovery Ratio (RR) can also be seen to increase non-linearly with the increase in temperature. The values are comparable with thermal desalination processes, which have lower RR values as compared to Reverse Osmosis (Shih, 2005). Another property used to determine the performance of a desalination system is its thermal efficiency. The thermal efficiency can be defined as the ratio between the total energy used for producing the vapor to the total energy added to the system. It can be expressed mathematically as:

$$\eta_{th} = \frac{m_f h_{fg}}{m_s h_s - m_b h_b + Q_{ext}} \quad (5.4)$$

where Q_{ext} represents any other heat flow in/out of the system.

It can be seen from Fig. 5.41 that the thermal efficiency increases with the increase in the evaporation temperature. Al-Kharabsheh and Goswami (Al-Kharabsheh & Goswami, 2004) reported a similar trend. As the temperature in the evaporation chamber increases, the temperature difference between the hot and the cold side increases and hence we get better evaporation rates. If the losses to the environment are minimized, and the entire vapor is condensed on the condenser side, a higher thermal efficiency is expected. If the process is not well insulated, then the rise in temperature difference will mean more heat losses to the ambient and this will result in a drop in the thermal efficiency (Gude et al., 2010).

5.4.2.2. Effect of Freshwater Side Temperature

The freshwater temperature in the cooled freshwater chamber is also a crucial factor in the freshwater production rate. The lower the temperature on this freshwater chamber, the higher the production rate and the recovery ratio (see Fig. 5.42). In cold regions, such low temperatures can be obtained

naturally by the ambient conditions. In other areas, a shade around the freshwater chamber, will be enough to provide low enough temperature for condensation. However, in hot areas, a certain cooling mechanism should be used to keep the temperature low enough to provide the driving force for the desalination process. In this case, the cooling load will add to operation cost and hence has to be seen as what temperature will be economically more viable.

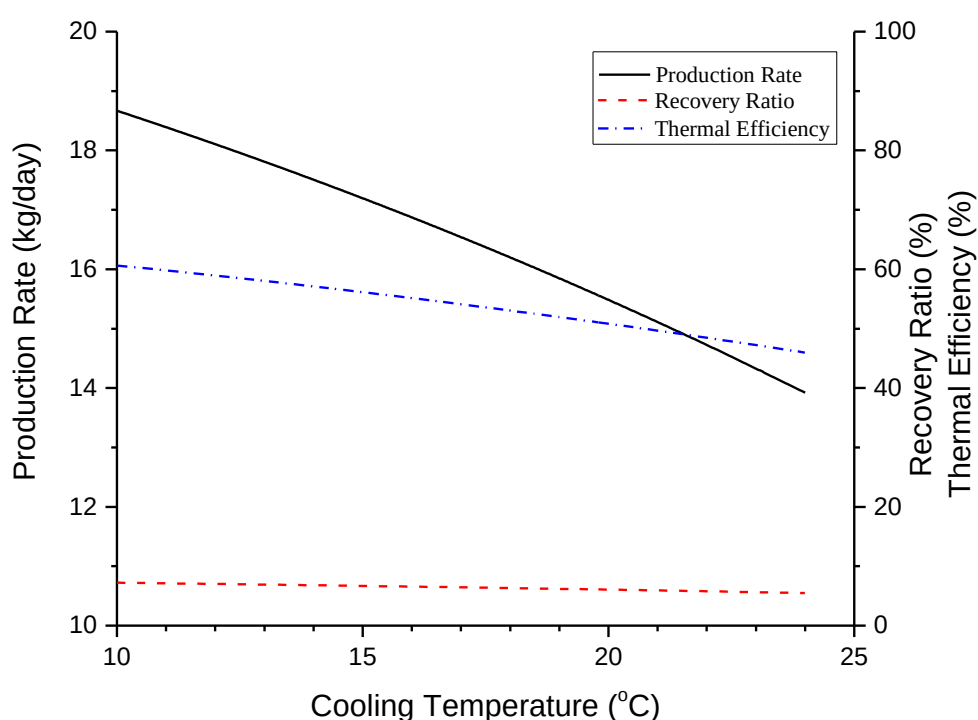


Figure 5.42: Accumulated production amount, recovery ratio and thermal efficiency as a function of condenser side temperature

An increase in the freshwater temperature also results in a lower thermal efficiency. The recovery ratio decreases slightly with the increase in the freshwater temperature. Hence, the cooling temperature should be maintained as low as economically feasible. If the vapor is produced at 60 °C, then to condense 18.6 kg/day of vapor and cool it to 10 °C, will require

removal of approximately 550 W and the value goes down to 390 W if the cooling temperature is maintained at 25 °C with the corresponding production rate. This is not a big number and can easily be obtained from solar photovoltaic energy.

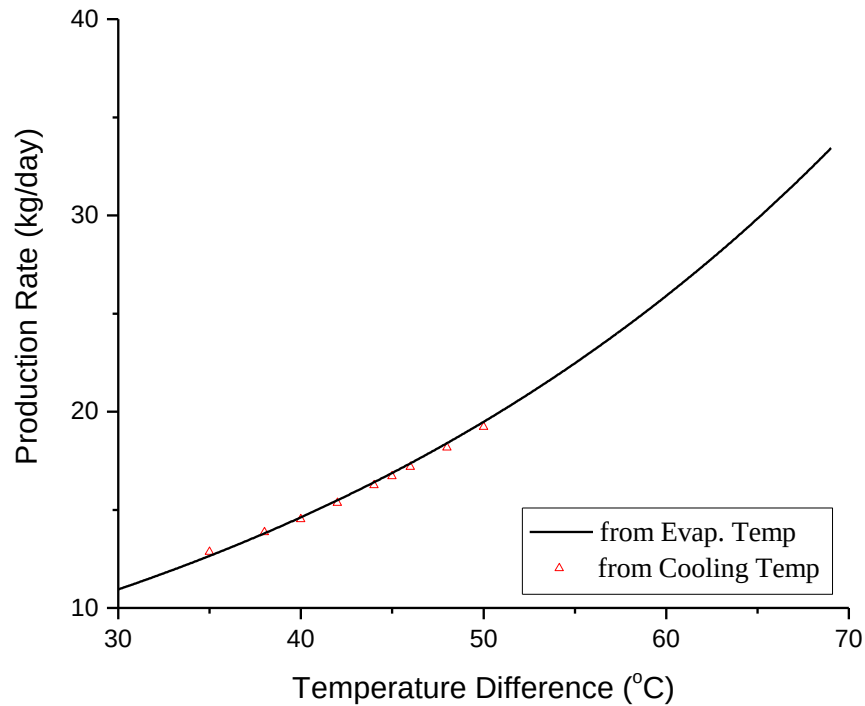


Figure 5.43: Variation of daily production rate with temperature difference

The production amount is presented as a function of the temperature difference between the evaporator chamber and the cooled freshwater chamber in Fig. 5.43. The results of the COMSOL simulation by varying the evaporation chamber temperature at a constant cooling temperature of 10 °C are shown by the solid line, whereas the red marks represent the production rate obtained by varying the cooling temperature at a constant heating temperature of 60 °C. The two values are presented together to check the consistency of the results by varying different control parameters. The production rate varies non-linearly with the change in the

temperature difference in a trend similar to the vapor pressure variation with the temperature. A similar trend has been shown by Martinson et al. (2007).

5.4.2.3. Effect of Brine Withdrawal Rate

We can see from Fig. 5.44 that the increase in the brine withdrawal rate has a significant effect on the evaporation rate. As, the withdrawal rate is increased, the evaporation rate too goes up. This is because the increase in the brine withdrawal rate means a higher brine exit temperature. The reasons for this is that the hot seawater fed to the evaporation chamber, spends less time in the evaporation tank and the effect of cooling due to evaporation is reduced. The exit temperature is higher when the brine withdrawal rate is higher, this automatically causes the evaporation rate to rise because the water in the evaporation tank will be at a higher temperature and hence the driving force for the evaporation (i.e. ΔT) will be higher.

One thing to note with the brine withdrawal rate is that some sort of optimum value needs to be found. Higher brine withdrawal rates would mean loss of energy and lower brine withdrawal rates would mean that the system might suffer from scaling. A low brine rate will not be enough to carry out all the salt deposits with it. Since the system is operated under vacuum, it will not be easy to open and clean the salt deposits. Hence, reducing the brine rate to very low values will not be advisable.

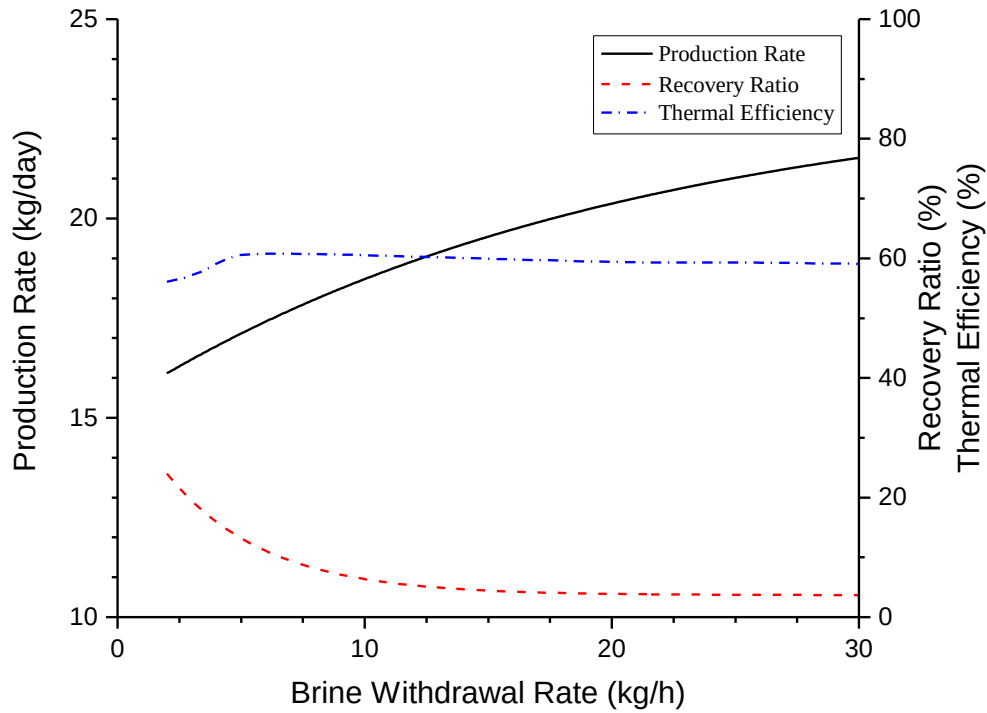


Figure 5.44: Accumulated production amount, recovery ratio and thermal efficiency as a function of brine withdrawal rate

From these results, it can be deduced that a higher withdrawal rate is preferred for the system. However, higher withdrawal rates mean higher heating load required for the inlet seawater. Therefore, an optimal withdrawal rate has to be determined based on the evaporation rate, concentration of the brine and the added cost of heating the inlet temperature. It can also be observed from Fig. 5.44, that higher withdrawal rates mean lower recovery ratios. This means that the exit stream is discharged out of the system carrying with it the energy that was required for heating it. Some recovery system should be designed to retrieve some of the energy lost through this exit stream.

The recovery ratio can be seen to drop with an increase in the brine withdrawal rate. The freshwater production rate does not increase proportionally with the increase in the withdrawal rate, hence the recovery

ratio drops. At low withdrawal rates, the inlet flow rate is low and this small flowrate of the hot water inlet is not enough to sufficiently heat the large mass of water present in the evaporation chamber. The evaporation rate itself causes a temperature drop and the heat supplied to the system through the feed water is not enough. Hence, a low thermal efficiency is noted at low withdrawal rates. As the withdrawal rate increases, the thermal efficiency again falls because a large amount of heated water is being removed without proper utilization of the heat added to the system.

5.4.2.4. Effect of Water Height in Evaporation Chamber

The effect of the height of the water column in the evaporation chamber on the freshwater production rates was almost linear (see Fig. 5.45). As the height in the evaporation chamber increases, it means there is a larger volume of water in the evaporation chamber. The inlet temperature is the same, however due to the evaporation, the temperature at the surface drops. This lower temperature causes the evaporation rate to fall. However, in the presence of a larger volume, the temperature drop was not as high. The large mass of water acts as a buffer to the effect of the evaporation cooling. This explains why the production rate is higher when the water column height is increased in the evaporation chamber. The thermal efficiency and the recovery ratio also increase with the height of water column in the evaporation chamber. A bigger water column height seems preferable but with any variations in the system, the water level will fluctuate and could overflow into the freshwater chamber. Hence, it is better to leave some space above the water surface to accommodate any unexpected fluctuations. For most of the experimental and theoretical investigation, a height of 35 cm was used.

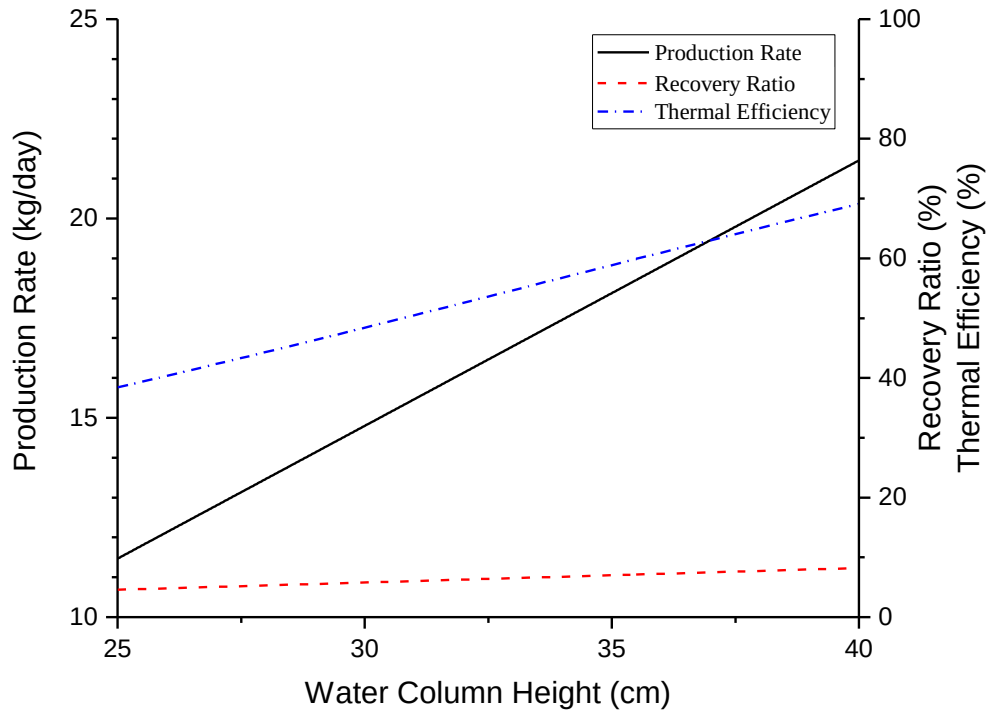


Figure 5.45: Accumulated production amount, recovery ratio and thermal efficiency as a function of the water column height in the evaporation chamber

5.5. NVD using a Solar Parabolic Trough Collector

The Parabolic Trough Collector (PTC) was simulated using the equations mentioned in Section 3.4. The model was tested against results from the literature and it was found to give accurate results. The model was then used with the solar irradiation and temperature of the Kingdom of Bahrain to investigate the performance of the PTC in Bahrain conditions. The parameters used for the PTC are listed in Table 5.4. For this study, data for the months of August and January were used which are generally the hottest and the coldest months, respectively. A one-week data was used by averaging the temperature and solar data obtained from the Bahrain Meteorological Directorate for the first weeks of the mentioned months. The data used was for the years 2010 – 2014.

Table 5.4: Parameters used in the PTC model

Parameter	Value	Unit
PTC length	2	m
PTC width	0.8	m
Focal Length	0.286	m
Glass tube transmissivity	0.82	
Glass tube absorptivity	0.11	
Glass tube diameter	6.35	cm
Glass tube thickness	2.8	mm
Glass tube thermal conductivity	1.05	W/m.K
Absorber tube transmissivity	0.01	
Absorber tube absorptivity	0.96	
Absorber tube diameter	12.7	mm
Absorber tube thickness	1.245	mm
Absorber tube thermal conductivity	377	W/m.K

The results of the simulation of the PTC along with the solar irradiation values and the ambient temperatures are shown in Fig. 5.46 and 5.47 for the summer and winter periods, respectively. The solar irradiation value for summer peaked at around 800-850 W/m² while it was only around 600-700 W/m² during the winter. The PTC output can be observed to have a maximum temperature of around 55 °C during the summer and around 35 °C during the winter. During the non-sunlight hours, the PTC output temperature is seen to be same as the ambient temperature.

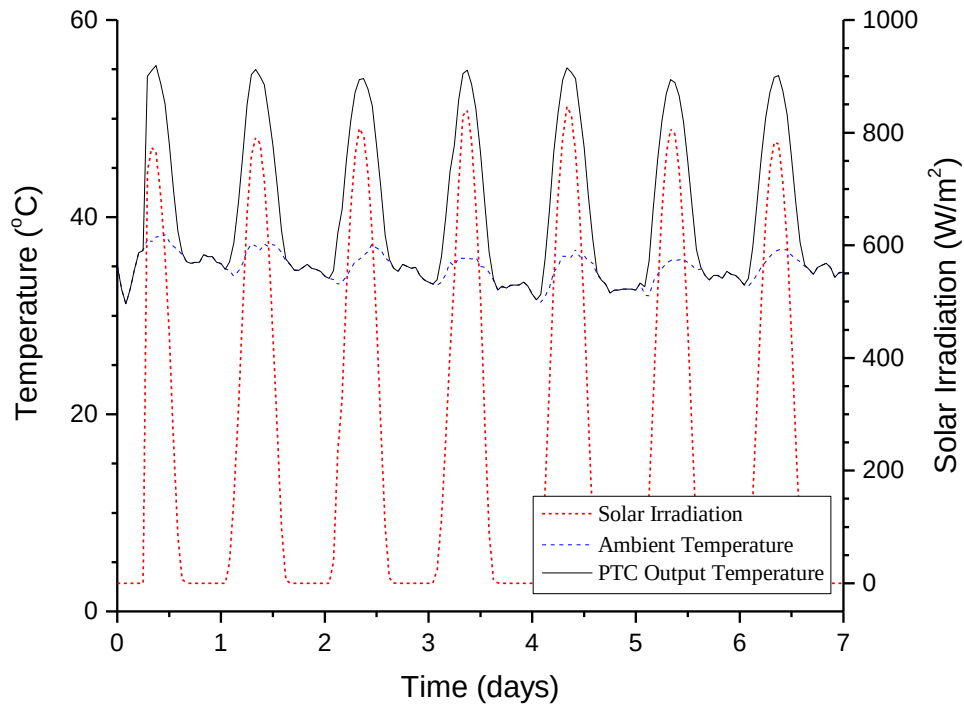


Figure 5.46: Solar irradiation and input and output temperatures of the PTC (during the summer period)

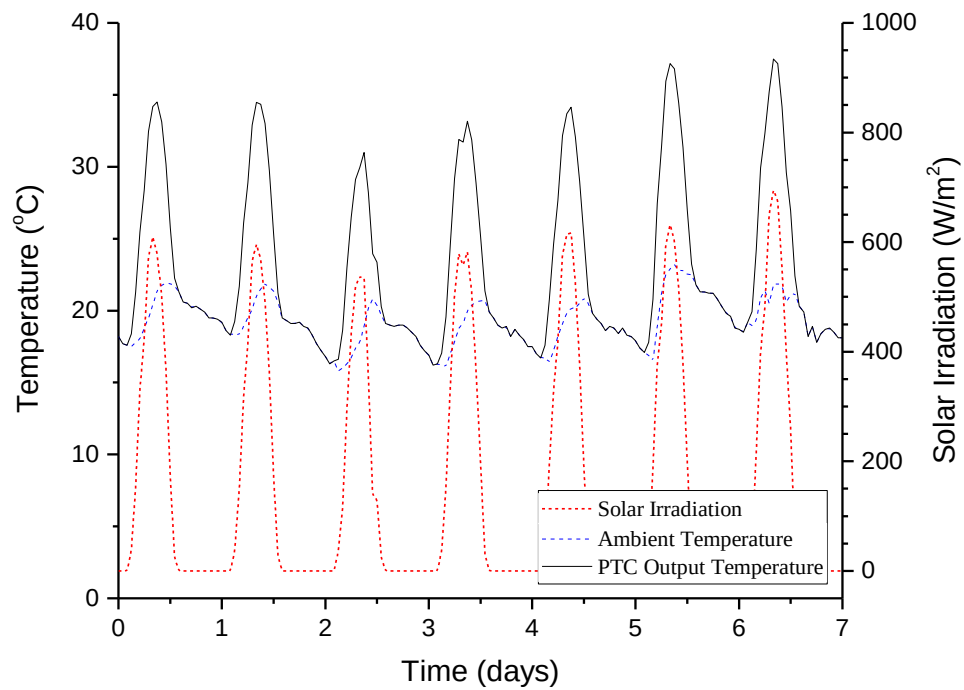


Figure 5.47: Solar irradiation and input and output temperatures of the PTC (during the winter period)

The PTC output temperature were used as input to the COMSOL model as a function of the time. The ambient temperature data was also used to compute the radiation losses and convection losses from the desalination unit. The 7 days simulation shows the accumulated output to be linear, as shown in Fig. 5.48. The evaporation rate hence seems to be somewhat constant. One would expect the evaporation rates to be higher during the day time and much lower or even close to zero during the night time. However, if we look at the surface temperature, where the evaporation actually happens, we see that the surface temperature is almost constant. Since the volume of water in the evaporation chamber is large and the inlet flowrate is very small, hence the impact of this inlet temperature is very little. Additionally, the evaporation itself causes a temperature drop, which also helps to lower the effect of the incoming temperature change. During the nighttime, the water is already warm in the evaporation chamber, and even if the water enters the chamber at ambient temperature, the water does not cool down so fast. During the summer, the ambient temperature is high and might be close to the warm temperature in the evaporation chamber.

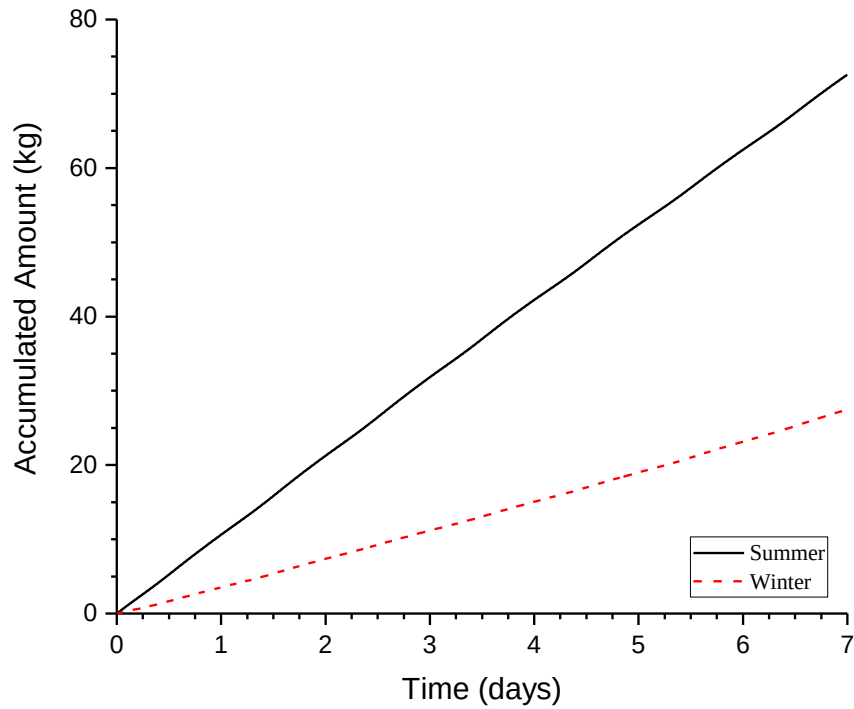


Figure 5.48: Accumulated production amount for the summer and winter operation

The performance of the system was studied by looking at the recovery ratio. The recovery ratio for both the summer and winter operation is shown in Fig. 5.49. It is almost constant and only a very little change is observed over the 7 days period. As explained earlier, the evaporation rates are almost constant so it automatically implies that the RR will also be constant. The average recovery ratio for the summer period was 30.16% and for the winter it was a much lower value, 14.05%. The recovery ratios are based on a brine withdrawal rate of 1 kg/h.

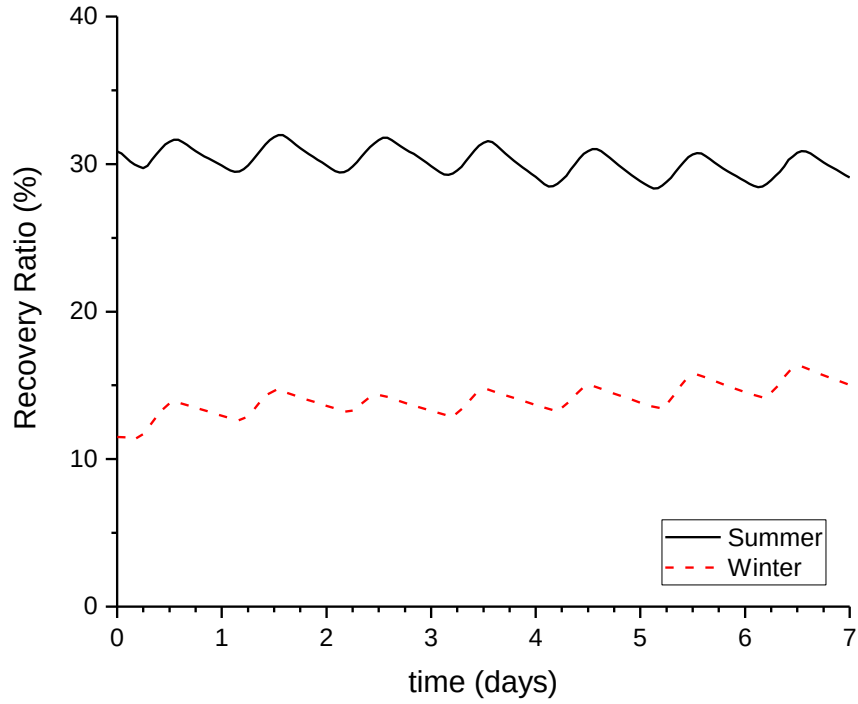


Figure 5.49: Comparison of the recovery ratios during the summer and winter operation

Another value used to demonstrate the performance of this solar energy based desalination system was the thermal efficiency. Thermal efficiency (η_{th}) is defined as the ratio of the energy used for the evaporation of water to the total energy absorbed the solar collector. It can be expressed mathematically as:

$$\eta_{th} = \frac{\int m_f h_{fg} dt}{\int I_{an} A_a dt} \quad (5.5)$$

The thermal efficiency over the entire 7 days period for summer operation was determined to be 73.49%, while it was 43.74% for the winter operation. The PTC output temperatures were not very high during the winter and hence the thermal efficiency is low. The values for η_{th} calculated for each day separately were close to the above-mentioned

values, except for the first day, where, as expected the efficiency was marginally lower.

5.6. Cost Analysis

The total cost of a process comprises the capital cost and the operating cost. In a simple system like the one presented in this study, the capital cost is very small. The total cost of the built unit used in the experimental setup was less than BD 1,000 (approximately USD 2,650). However, this cost includes measurement instruments and other attachments, which are not required for unit that is produced for the purpose of water production only. The required elevation for the NVD process was obtained by placing the tank on the roof of an existing building. In absence of any such facility, some structure has to be built to provide the required height. Hence, the actual capital cost will depend on such factors that vary from place to place. The cost analysis shown here is only for the operating cost.

The energy requirements for the desalination process include the heating load (heating the water from the ambient to the set temperature), the evaporation energy (the latent heat required to evaporate the water at the set temperature), the cooling load (the energy required to condense the water vapor and then to cool it to the condenser temperature) and the heat losses from the evaporation chamber. The total energy required for the four different operating temperatures in the batch operation is shown in Fig. 5.50. The most important component of the operation cost is the heating cost. In the experimental analysis, an electrical heater with a heating power of 1.0 kW was used. Mechanical pumping is required only for the startup phase, hence it will be ignored in this calculation. A vacuum pump may be used if the pressure builds up during the continuous process.

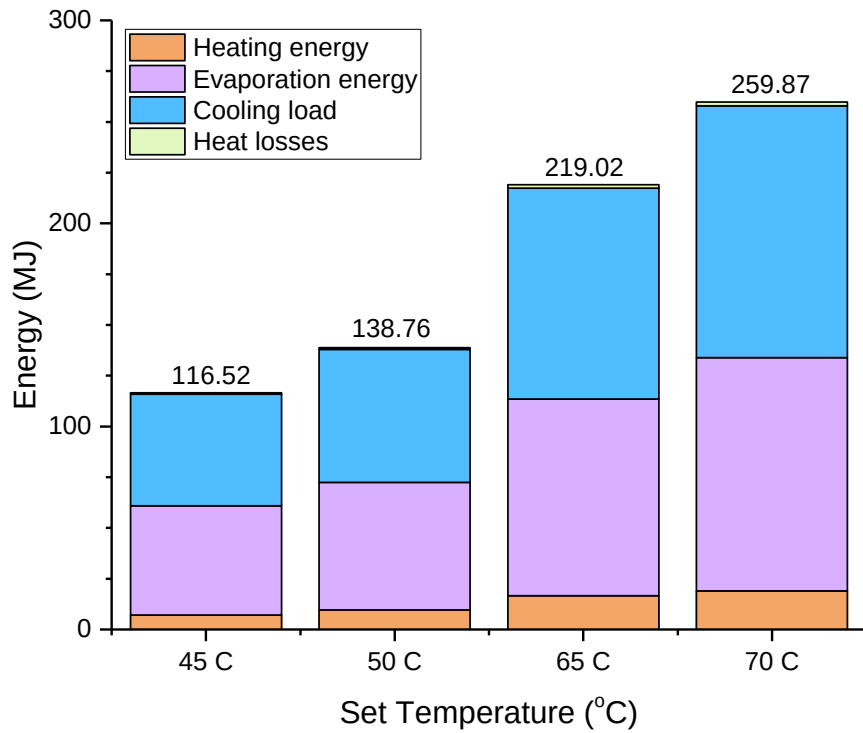


Figure 5.50: Breakdown of the total energy requirements for the batch desalination process at different operating temperatures

The specific energy requirement (i.e. energy required to produce one kg of freshwater) for the desalination system is shown in Fig. 5.51. The specific energy consumption increases slightly with the increasing operating temperature. The rate of increase is almost linear. This behavior agrees with what is predicted by the theoretical thermodynamic analysis (Gude & Nirmalakhandan, 2009). The specific energy consumption corresponding to the operating temperatures of 45 and 70 °C were 1.467 and 1.480 kWh/kg, respectively. Due to the hot climatic conditions in the Kingdom of Bahrain, the cooling load necessary for condensation represented about 43% of the total energy consumption. It is to be noted that the minimum theoretical specific energy required for a phase-change desalination process is the latent heat of vaporization of water, which is around 0.63 kWh/kg. The cost of electricity per unit (1 unit = 1 kWh) in the Kingdom of Bahrain is BHD 0.003 and BHD 0.016 for domestic and

non-domestic purposes, respectively. Hence the cost of water production, based on the highest specific energy value obtained in this study, is BHD 0.0044 (USD 0.01) per kg or BHD 0.0237 (USD 0.06) per kg based on domestic and non-domestic electrical rates, respectively.

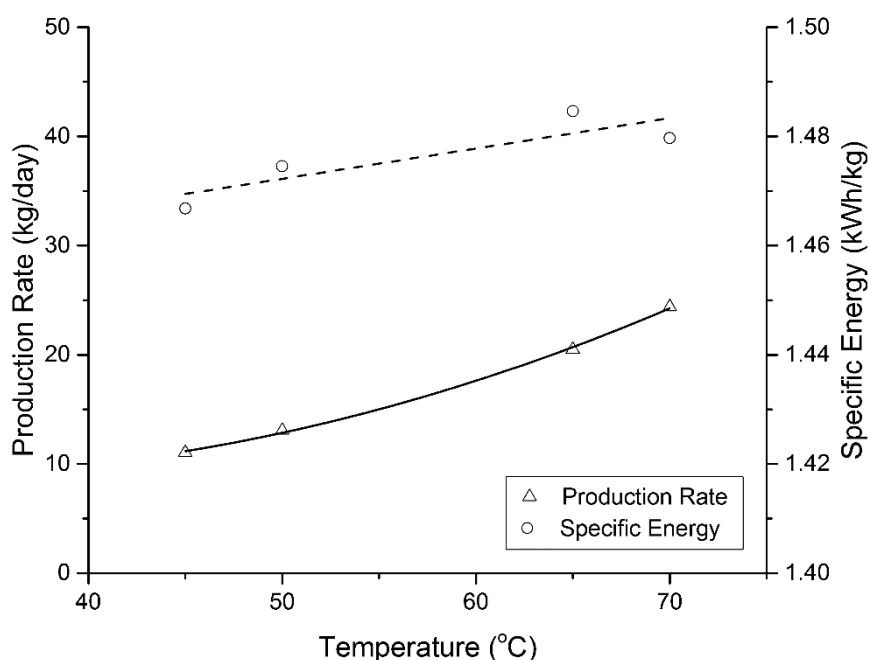


Figure 5.51: Specific energy requirements and water production rate for the batch desalination process at different operating temperatures

For the continuous process, the heat requirement load will be similar to the batch process. From the experimental runs, it can be seen that the pressure builds up in the system and a vacuum pump was employed to regenerate the vacuum. The vacuum pump was needed after approximately 8 hours. Therefore, it will be operated for 3 times during a day, with a maximum operating time of 10 minutes. A 0.3 kW vacuum pump was employed in the experiment, so a total of 0.15 kWh of electricity was consumed for the vacuum pump. No mechanical pumping is required for the operation of the process, hence this small amount of energy required

for the vacuum pump does not change the specific energy requirement by much.

The Kingdom of Bahrain enjoys a daily direct normal irradiance of around 5.5 kWh/m² (Muneer et al., 2007). This means that the heating load required by such an NVD desalination process can easily be catered for using solar energy. Other renewable or waste heat sources can also be used as an alternative means for heating the feed water.

Chapter 6

Conclusion

Table of Contents

Subject	Page No.
Chapter Six Conclusion	156

Chapter 6: Conclusion

An experimental system for water desalination that operates on the principle of Natural Vacuum Distillation (NVD) was studied theoretically and experimentally. The system has a novel conical roof design. This new geometry solves the problem of vapor transport from the evaporation chamber to the condenser side in the previously studied inverted U-shaped design. It also enhances the condensation amount and experimental results have shown that almost all the vapor produced in the evaporation chamber is condensed on to the freshwater side.

Experimental studies were carried out on the system in batch and continuous modes of operation. The following results were obtained:

- As the temperature of the heated water increased, the freshwater production rate increased. However, this also caused an increase in the specific energy requirement for the water production. Hence, operation at lower temperatures is more feasible.
- In the batch mode, water production rate increased from 11.2 L/day to 24.4 L/day as the operating temperature was increased from 45 °C to 70 °C.
- In the continuous mode, the production rate increased from 9.05 L/day to 15.65 L/day as the operating temperature was increased from 50 °C to 70 °C.
- The effect of the feed water salinity was also studied and a drop of almost 5% in the freshwater production was observed as the salinity was increased from 0.1% to 3.5%.
- The bubbles that are formed in the evaporation chamber due to the heating of water directly in this chamber, are carried to the withdrawal pipe and can cause obstruction to the continuous

operation. It is therefore preferred to heat the water before feeding it to the evaporation chamber.

- During continuous operation, the vacuum in the system may diminish as a result of the accumulation of the non-condensable gases. A vacuum pump is almost a necessity to purge such gases to ensure uninterrupted operation.

Comparing the experimental results of the current study with those presented in the literature, we can see that the design used in the current study was able to deliver a higher water production rate. Since the various systems studied by previous researchers have been of different sizes, a comparison was done with the water production rate per unit area of the evaporator. The results achieved from the current experimental setup (in the batch mode) are 33.2 L/day/m². This number is higher than the results reported by Al-Kharabsheh and Goswami (2003), 6.5 L/day/m², Midilli and Ayhan (2004), 25.6 L/day/m², Gude and Nirmalakhandan (2009), 16.5 L/day/m² and Ambarita (2016) who reported a water production rate of 6.6 L/day/m².

A theoretical model was also developed to simulate the NVD process. The model was validated using the experimental results. The model showed a lag as compared to the experimental values. This was due to the usage of an electrical heater in the experiments, while a constant temperature boundary was used in the simulation. However, after the initial lag, the model followed the experimental results very accurately. A commercial finite element analysis software, COMSOL® , was used for the simulation. The simulation was used for parametric studies on the NVD system. The following results were obtained from these studies:

- Freshwater production amount was found to be in direct proportion to the evaporation temperature and inversely proportional to the freshwater side temperature.
- For the batch process simulation, production amount increased from 15.66 L to 70.75 L as the operating temperature was increased from 40 °C to 70 °C for a 48 hours operating period.
- In the continuous mode, the production rate increased from 11 L/day to 33 L/day as the operating temperature was increased from 40 °C to 80 °C.
- For the batch process simulation, production amount decreased from 28.71 L to 20.47 L as the cooling temperature was increased from 10 °C to 27.5 °C for a 48 hours operating period.
- In the continuous mode, the production rate dropped from 18.6 L/day to 13.8 L/day as the cooling temperature was increased from 10 °C to 24 °C.
- The freshwater production amount was also found to increase non-linearly with the evaporation surface area. The production amount increased 3 times as the evaporation chamber diameter was increased from 0.4 m to 1.0 m.
- The recovery ratio and the thermal efficiency both increased with the increase in the heating temperature for the continuous operation. An increase in the freshwater side temperature caused the production rate, recovery ratio and the thermal efficiency to fall.
- The brine withdrawal rate had a big impact on the system performance. Freshwater production rate increased from 15.8 L/day to 21.7 L/day as the brine withdrawal rate was increased from 2 to 30 kg/h. Although the production rate increased as the brine withdrawal rate was increased, the recovery ratio and the thermal

efficiency decreased as the brine withdrawal rate was increased. The thermal efficiency showed a peak at a withdrawal rate of 5 kg/h.

- The depth of water in the evaporation chamber also affected the production rate. The recovery ratio did not show much change but noticeable increments were seen in the production amount and the thermal efficiency of the system as the depth of water was increased. Production rate was seen to increase from 11.8 to 22.0 L/day as the water column height in the evaporation chamber increased from 25 to 40 cm.

The system behavior was also studied theoretically to analyze the feasibility of using solar energy for the desalination process. A 1.6 m² parabolic trough collector (PTC) was designed and modeled theoretically for the climatic conditions of the Kingdom of Bahrain. The PTC results showed maximum temperatures of 55.4 °C and 37.5 °C for the feed water during the months of August and January, respectively. Freshwater production rates of 10.37 kg/day and 3.93 kg/day were obtained for the two months respectively.

For further studies, it is recommended to study the possibility of using a multistage configuration to enhance the freshwater production rate. In this study, the solar energy integration was studied theoretically. It will be more useful to design an experimental setup that operates entirely on renewable energy to establish concrete evidence on the feasibility of such a desalination system. The conical roof of the experimental setup is not used and this space can be utilized for solar photovoltaic (PV) panels. Such a conical configuration for the solar PV panels will not require a special orientation for them and neither a solar tracking system. These solar PV panels can be used to generate power for heating or for driving the vacuum pump.

References

- Abutayeh, M. & Goswami, D. Y. (2009). Solar Flash Desalination under Hydrostatically Sustained Vacuum, *Journal of Solar Energy*, (131), 031016.
- Abutayeh, M. & Goswami, D. Y. (2010a). Experimental Simulation of Solar Flash Desalination, *Journal of Solar Energy*, (132), 041015.
- Abutayeh, M. & Goswami, D. Y. (2010b). Passive Vacuum Solar Flash Desalination, *AIChE Journal*, 56(5), 1196-1203.
- Abutayeh, M., Goswami, D. Y. & Stefanakos, E. K. (2013). Theoretical and Experimental Simulation of Passive Vacuum Solar Flash Desalination, *Journal of Solar Energy*, (135), 021013.
- Al-Kharabsheh, S. & Goswami, D. Y. (2003a). Analysis of an innovative water desalination system using low-grade solar heat, *Desalination*, (156), 323-332.
- Al-Kharabsheh, S. & Goswami, D. Y. (2003b). Experimental study of an innovative solar water desalination system utilizing a passive vacuum technique, *Solar Energy*, (75), 395-401.
- Al-Kharabsheh, S. & Goswami, D. Y. (2004). Theoretical analysis of a water desalination system using low grade solar heat, *Journal of Solar Energy Engineering*, (126), 774-780.

- Al-Shammiri, M. (2002). Evaporation rate as a function of water salinity, *Desalination*, (150), 189-203.
- Al-Shammiri, M. & Safar, M. (1999). Multiple-effect distillation plants: state of the art, *Desalination*, (126), 45-59.
- Alty, T. & Mackay, C. A. (1935). The Accommodation Coefficient and the Evaporation Coefficient of Water, *Proceedings of the Royal Society of London*, (149), 104-116.
- Ambarita, H. (2016). Study on the performance of natural vacuum desalination system using low grade heat source, *Case Studies in Thermal Engineering*, (8), 346-358.
- Ambarita, H. (2017). Numerical Study on Natural Vacuum Solar Desalination System with Varying Heat Source Temperature, *IOP Conference Series: Materials Science and Engineering*, (180), 012024.
- Attia, A. A. (2016). Heat pump seawater distillation system using passive vacuum generation system, *Desalination*, (397), 151-156.
- Aydin, D., Casey, S. P. & Riffat, S. (2015). The latest advancements on thermochemical heat storage systems, *Renewable and Sustainable Energy Reviews*, (41), 356-367.

- Ayhan, T. & Al-Madani, H. (2010). Feasibility study of renewable energy powered seawater desalination technology using natural vacuum technique, *Renewable Energy*, (35), 506-514.
- Ayhan, T. & Al-Madani, H. (2015). Combined effects of evaporation and cavitation on the performance of a renewable energy powered natural vacuum desalination unit in Bahrain, *Desalination and Water Treatment*, 57(28) 12929-12940.
- Bayer, M. E. (1967). Response of Cell Walls of *Escherichia coli* to a Sudden Reduction of the Environmental Osmotic Pressure, *Journal of Bacteriology*, 93(3), 1104-1112.
- Bemporad, G. A. (1995). Basic hydrodynamic aspects of a solar energy based desalination process, *Solar Energy*, 54(2), 125-134.
- Bird, R. B., Stewart, W. E. & Lightfoot, E. N. (2002). Transport Phenomena. 2 ed. New York: John Wiley and Sons.
- Burn, S., Hoang, M., Zarzo, D., Olewniak, F., Campos, E., Bolto, B. & Barron, O. (2015). Desalination techniques - A review of the opportunities for desalination in agriculture, *Desalination*, (364), 2-16.
- Chu, Y. & Meisen, P. (2011). Review and Comparison of Different Solar Energy Technologies. Global Energy Network Institute.

COMSOL Inc.. (2016). Multiphysics Simulation Software. Retrieved from <https://www.comsol.com/comsol-multiphysics>.

Daintith, J. (2009). A dictionary of physics, 7 ed. New York: Oxford University Press

Davidovits, P., Worsnop, D. R., Jayne, J. T., Kolb, C. E., Winkler, P., Vrtala, A., Wagner, P. E., Kulmala, M., Lehtinen, K. E. J., Vesala, T. & Mozurkewich, M. (2004). Mass accommodation coefficient of water vapor on liquid water, *Geophysical Research Letters*, 31(22), L22111.

Davidovits, P., Kolb, C. E., Leah, R. W., Jayne, J. T. & Worsnop, D. R.. (2011). Mass Accommodation and Chemical Reactions at Gas-Liquid Interfaces, *Chemical Reviews*, 111(4), 76-109.

Delyannis, E. & El-Nashar, A. (2010). A Short Historical Review of Renewable Energy. In: D. M. K. Al-Gobaisi (Ed.), *Renewable Energy Systems and Desalination*, (pp. 1-48) Oxford: EOLSS Publishers Co Ltd.

Duffie, J. A. & Beckman, W. A. (2013). Solar Engineering of Thermal Processes. 4 ed., New Jersey: John Wiley & Sons, Inc.

Eames, I. W., Maidment, G. G. & Lalzad, A. K. (2007). A theoretical and experimental investigation of a small-scale solar-powered

barometric desalination system, *Applied Thermal Engineering*, (27), 1951-1959.

Eltawil, M.A., Zhengming, Z. & Yuan, L. (2009). A review of renewable energy technologies integrated with desalination systems, *Renewable and Sustainable Energy Reviews*, (13), 2245-2262.

Fernandes, D. F., Pitié, F., Cáceres, G. & Baeyens, J. (2012). Thermal energy storage: “How previous findings determine current research priorities”, *Energy*, (39), 246-257.

Fernandez-Garcia, A., Zarza, E., Valenzuela, L. & Perez, M. (2010). Parabolic-trough solar collectors and their applications, *Renewable and Sustainable Energy Reviews*, (14), 1695–1721.

Fujikawa, S. & Akamatsu, T. (1980). Effects of the non equilibrium condensation of vapour on the pressure wave produced by the collapse of a bubble in a liquid, *Journal of Fluid Mechanics*, (97), 481-512.

Gadhamshetty, V., Gude, V. G. & Nirmalakhandan, N. (2014). Thermal energy storage system for energy conservation and water desalination in power plants, *Energy*, (66), 938-949.

- Gude, V. G. (2015). Energy storage for desalination processes powered by renewable energy and waste heat sources. *Applied Energy*, (137), 877-898.
- Gude, V. G. & Nirmalakhandan, N. (2008). Desalination using Low-grade heat sources, *Desalination*, (134), 95-101.
- Gude, V. G. & Nirmalakhandan, N. (2009). Desalination at low temperatures and low pressures, *Desalination*, (244), 239-247.
- Gude, V. G. & Nirmalakhandan, N. (2010). Sustainable desalination using solar energy, *Energy Conversion and Management*, (51), 2245-2251.
- Gude, V. G., Nirmalakhandan, N. & Deng, S. (2010). Low temperature process to recover impaired waters, *Desalination and Water Treatment*, (20), 281-290.
- Gude, V. G., Nirmalakhandan, N. & Deng, S. (2011), Sustainable low temperature desalination: A case for renewable energy, *Journal of Renewable and Sustainable Energy*, (3) 043108.
- Gude, V. G., Nirmalakhandan, N. & Deng, S. (2012). Desalination at low temperatures: an exergy analysis, *Desalination and Water Treatment*, (40), 272-281.

- Gude, V. G., Nirmalakhandan, N., Deng, S. & Maganti, A. (2012). Feasibility study of a new two-stage low temperature desalination process, *Energy Conversion and Management*, (56), 192-198.
- Gude, V. G., Nirmalakhandan, N., Deng, S. & Maganti, A. (2012). Low temperature desalination using solar collectors augmented by thermal energy storage, *Applied Energy*, (91), 466-474.
- Hertz, H. (1882). On the Evaporation of Liquids, Especially Mercury, in Vacuo, *Annals of Physics*, (17), 177.
- Hickman, K. C. D. & Torpey, W. A. (1954). Evaporation of Resting Water, *Industrial and Engineering Chemistry*, 46(7), 1446-1450.
- Hill, P. G. (1966). Condensation of water vapor during supersonic expansion in nozzles, *Journal of Fluid Mechanics*, (25), 593-620.
- Hogg, R. V., Tanis, E. A. & Zimmerman, D. L. (2015). Probability and Statistical Inference. 9th ed. New Jersey: Pearson.
- Holman, J. P. (2010). Heat Transfer. 10 ed., New York: McGraw-Hill Publishing.
- Incropera, F. P., DeWitt, D. P., Bergman, T. L. & Lavine, A. S. (2012). Principles of Heat and Mass Transfer. 7 ed., New Jersey: John Wiley & Sons, Inc.

- Jebasingh, V. K. & Joselin Herbert, G. M. (2016). A review of solar parabolic trough collector, *Renewable and Sustainable Energy Reviews*, (54), 1085-1091.
- Jitsuno, T. & Hamabe, K. (2012). Vacuum Distillation System Aiming to Use Solar-Heat for Desalination, *Journal of Arid Land Studies*, (22), 153-155.
- Kalogirou, S. A. (2004). Solar thermal collectors and applications. *Progress in Energy and Combustion Science*, (30), 231-295.
- Kalogirou, S. (2013). Solar Energy Engineering: Processes and Systems. 2 ed., Amsterdam: Academic Press.
- Kandlikar, S. G. & Vij, S. K. (1978). Geometrical aspects of a cylindrical parabolic collector, *Proceedings of the International Solar Energy Congress*, New Delhi: Pergamon Press Inc., 1254-1258.
- Keren, Y., Rubin, H., Atkinson, J., Priven, M. & Bemporad, G. A. (1993). Theoretical and experimental comparison of conventional and advanced solar pond performance, *Solar Energy*, 51(4), 255-270.
- Kerskes, H. (2016). Thermochemical Energy Storage. In: T. Letcher, (Ed.) *Storing Energy*, (pp. 345-372), Elsevier Inc.
- Khawaji, A. D., Kutubkhanah, I. K. & Wie, J. (2008). Advances in seawater desalination technologies, *Desalination*, (221), 47-69.

- Kline, S. J. & McClintock, F. A. (1953). Describing uncertainties in single-sample experiments, *Mechanical Engineering*, (75), 3-8.
- Knudsen, M. (1915). Maximum Rate of Vaporization of Mercury, *Annals of Physics*, (47), 697-708.
- Kreith, F., Manglik, R. M. & Bohn, M. S. (2011). Principles of Heat Transfer. 7 ed. Boston: Cengage Learning, Inc.
- Kronenberg, G. & Lokiec, F. (2001). Low-temperature distillation processes in single- and dual-purpose plants, *Desalination*, (136), 189-197.
- Kucera, J. (2014). Desalination - water from water. New Jersey: John Wiley & Sons, Inc.
- Kucera, J. (2015) Reverse Osmosis, 2 ed. Beverly: Wiley-Scrivener Publishing
- Langmuir, I. (1916). The Evaporation, Condensation and Reflection of Molecules and the Mechanism of Adsorption, *Physical Review*, (8), 149-176.
- Linsley, R. K., Franzini, J. B., Freyberg, D. L. & Tchobanoglous, G. (1992). Water Resources Engineering. 4 ed. New York: McGraw-Hill.

- Liu, X., Zhang, X., Fang, Y. & Zhu, M. (2014). The Natural Vacuum Desalination Technology in Seawater Desalination, *Applied Mechanics and Materials*, (675-677), 851-855.
- Lotfi, A., Vrabec, J. & Fischer, J. (2014). Evaporation from a free liquid surface, *International Journal of Heat and Mass Transfer*, (73), 303-317.
- Macedo-Valencia, J., Ramirez-Avila, J., Acosta, R., Jaramillo, O. A. & Aguilar, J. O. (2014). Design, construction and evaluation of parabolic trough collector as demonstrative prototype, *Energy Procedia*, (57), 989-998.
- Maidment, G. G., Eames, I. W., Psaltas, M. & Lalzad, A. (2006). Potential application of a flash-type barometric desalination plant powered by waste heat from electric power stations in Cyprus, *Applied Energy*, (83), 1089-1100.
- Maidment, G. G., Eames, I. W., Psaltas, M. & Yiakoumetti, K. (2007). Flash-type barometric desalination plant powered by waste heat from electricity power stations in Cyprus, *Applied Energy*, (84), 66-77.

- Maroo, S. C. & Goswami, D. Y. (2009). Theoretical analysis of a single-stage and two-stage solar driven flash desalination system based on passive vacuum generation, *Desalination*, (249), 635-646.
- Martinson, E., Moore, B. & Raviv, D. (2007). Turning waste heat into water: a new desalination process, *WIT Transactions on Ecology and the Environment*, (102), 893-900.
- Mekhilef, S., Saidur, R. & Safari, A. (2011). A review on solar energy use in industries, *Renewable and Sustainable Energy Reviews*, (15), 1777-1790.
- Midilli, A. & Ayhan, T. (2004a). Natural vacuum distillation technique- Part I: Theory and Basics, *International Journal Energy Research*, (28), 355-371.
- Midilli, A. & Ayhan, T. (2004b). Natural vacuum distillation technique- Part II: Experimental investigation, *International Journal Energy Research*, (28), 373-389.
- Moffat, R. J. (1988). Describing the Uncertainties in Experimental Results, *Experimental Thermal and Fluid Science*, (1), 3-17.
- Moore, B. A., Martinson, E. & Raviv, D. (2008). Waste to water: a low energy water distillation method, *Desalination*, (220), 502-505.

- Muneer, T., Alnaser, W. E. & Fairouz, F. (2007). The insolation on vertical surface having different directions in the Kingdom of Bahrain, *Desalination*, (209), 269-274.
- Narusawa, U. & Springer, G. S. (1975). Measurements of evaporation rates of water, *Journal of Colloid and Interface Science*, 50(2), 392-395.
- Nayar, K. G., Sharqawy, M. H., Banchik, L. D. & Lienhard, J. H. (2016). Thermophysical properties of seawater: A review and new correlations that include pressure dependence, *Desalination*, (390), 1-24.
- Nirmalakhandan, N. & Gude, V. G. (2010). Sustainable recovery of potable water from saline waters. New Mexico.
- Nomura, T. & Akiyama, T. (2017). High-temperature latent heat storage technology to utilize exergy of solar heat and industrial exhaust heat, *International Journal of Energy Research*, 41(2), 240-251.
- Ophir, A. & Lokiec, F. (2005). Advanced MED process for most economical sea water desalination, *Desalination*, (182), 187-198.
- Panin, G. N. & Brezgunov, V. S. (2007). Influence of the salinity of water on its evaporation, *Atmospheric and Oceanic Physics*, 43(5), 663-665.

- Parker, S. A. (1982). Barometric distillation system. United States of America, Patent No. 4,319,965.
- Patience, G. S. (2013). Experimental Methods and Instrumentation for Chemical Engineers, Elsevier.
- Prüger, W. (1940). Rate of vaporization of liquids, *Zeitschr. Physik*, (21), 202-244.
- Puretec (2015) What is Reverse Osmosis? [online] Available at: <https://puretecwater.com/reverse-osmosis/what-is-reverse-osmosis> [Accessed 21 Jan. 2017].
- Rashid, A., Ayhan, T. & Abbas, A. (2016). Natural vacuum distillation for seawater desalination – A review, *Desalination and Water Treatment*, (57), 26943-26953.
- Reali, M. (1984). A refrigerator-heat-pump desalination scheme for fresh-water and salt recovery, *Energy*, 9(7), 583-588.
- Reali, M. (2006). Solar barometric distillation for seawater desalting Part II: Analyses of one-stage and two-stage distillation technologies, *Desalination*, (190), 29-42.
- Reali, M. (2007a). Solar barometric distillation for seawater desalting Part III: Analyses of one-stage and two-stage solar vapour thermo-compression distillation technologies, *Desalination*, (207), 304-323.

- Reali, M. (2007b). Solar barometric distillation for seawater desalting Part IV: Analyses of one-effect and two-effect distillation technologies, *Desalination*, (212), 219-237.
- Reali, M., Modica, G., El-Nashar, A. M. & Marri, P. (2004). Solar barometric distillation for seawater desalting Part I: Basic layout and operational/technical features, *Desalination*, (161), 235-250.
- Schmitt, F. G. (2007). About Boussinesq's turbulent viscosity hypothesis: historical remarks and a direct evaluation of its validity, *Comptes Rendus Mecanique*, (335), 617-327.
- Schrage, R. W. (1953). A Theoretical Study of Interphase Mass Transfer. New York: Columbia University Press.
- Scott, W. (2011) Multi-Stage Flash Distillation for Desalination [online] Available at: <http://www.brighthubengineering.com/structural-engineering/109915-multi-stage-flash-distillation-for-desalination/> [Accessed 18 Jan. 2017].
- Sharqawy, M. H., Lienhard, J. H. & Zubair, S. M. (2010). Thermophysical properties of seawater: a review of existing correlations and data, *Desalination and Water Treatment*, (16), 354-380.

- Shenvi, S. S., Isloor, A. M. & Ismail, A. F. (2015). A review on RO membrane technology: Developments and challenges, *Desalination*, (368), 10-26.
- Shih, H. (2005). Evaluating the technologies of thermal desalination using low grade heat, *Desalination*, (182), 461-469.
- Simonovic, T., Stamenic, M., Tanasic N. & Trninic, M. (2016) Effect of small deviation of incident angle on thermal performance of parabolic-trough solar collector, 4th International Symposium on Environmental Friendly Energies and Applications (EFEA), Belgrade, 2016, pp. 1-4.
- Tokaty, G. A. (1994). A History and Philosophy of Fluid Mechanics. New York: Dover Publications Inc.
- Veolia (2016) MED: Multiple Effect Distillation [online] Available at: <http://technomaps.veoliawatertechnologies.com/multipleeffectdistillation/en/> [Accessed 19 Jan. 2017].
- Youssef, P. G., Al-Dadah, R. K. & Mahmoud, S. M. (2014). Comparative Analysis of Desalination Technologies, *Energy Procedia*, (61), 2604-2607.

Zhang, H., Baeyens, J., Cáceres, G., Degrève, J. & Lv, Y. (2016).

Thermal energy storage: Recent developments and practical aspects,

Progress in Energy and Combustion Science, (53), 1-40.

Zimmerman, W. (2006). Multiphysics Modeling with Finite Element

Methods. 1 ed., New Jersey: World Scientific Publishing Co.

Appendices

Table of Contents

Title	Page No.
Appendix A – Evaporation Rate Calculation	A-1
Evaporation Rate Calculation	A-1
The mass accommodation factor	A-6
Comparison of the evaporation equations	A-8
Appendix B – Thermophysical properties of water	B-1
Water density as a function of temperature and salinity	B-1
Water specific heat as a function of temperature and salinity	B-2
Water thermal conductivity as a function of temperature and salinity	B-3
Water specific enthalpy as a function of temperature and salinity	B-4
Water Latent heat of vaporization as a function of temperature and salinity	B-5
Water Vapor Pressure as a function of temperature and salinity	B-6
Water Thermal Diffusivity as a function of temperature and salinity	B-7

Appendix A

Evaporation rate calculation

Evaporation rate calculation

To model a desalination system based on NVD, it is very important to know the evaporation rate. It would not be wrong to say that the evaporation rate calculation is the most important factor in the modeling of an NVD system.

Evaporation is often misleadingly defined as “the transfer of water from the liquid state to the vapor” (Linsley et al., 1992). Although this definition is acceptable in most of the cases, it is misleading because water is not the only substance that evaporates. It is also not a very good definition because it does not give any idea about the process of evaporation itself.

The process of evaporation can be better explained by the concept of the vapor pressure of a liquid. Molecules of a liquid are in constant movement. The average rate of the motion of these molecules depends on the kinetic energy of the system, which is indirectly measured as temperature. Molecules that possess a higher kinetic energy often break the weak bonds that hold the liquid molecules together, and move to the gas phase. These molecules can remain in the vapor phase or return to the liquid phase. Any liquid with a free surface experiences transfer of molecules from the liquid to the vapor phase, causing the liquid to have a vapor pressure over the surface of the liquid. The liquid will continue to lose molecules to the vapor phase until an equilibrium is reached. The vapor pressure at equilibrium is dependent on the kinetic energy (or temperature) and the properties of the liquid. Generally, the vapor pressure of a liquid

will increase with an increase in the temperature. Evaporation can thus be defined as the process of equilibrating the vapor pressure of the liquid to the equilibrium partial vapor pressure of the liquid in the gas phase.

Many equations have been reported in the literature for determining evaporation rates of pure water. They were all based on the equation derived by the Nobel Award winning physicist Irving Langmuir to measure the vapor pressure by knowing the evaporation rate (Langmuir, 1916). His derivation was based on the assumption that the rate at which molecules are lost due to evaporation to a gas with no partial pressure of the evaporating substance, is the same as the rate at which molecules of the substance would hit the surface if it were in equilibrium with the vapor. Langmuir's equation for vapor pressure is:

$$p_v = \sqrt{2\pi m_{ev} kT} \frac{\sigma_1}{\tau} \quad (\text{A1.6})$$

where m_{ev} is the evaporation rate, T is the temperature, k , σ_1 and τ are parameters that depends on the substance and its molecular structure.

This equation can be rearranged to calculate the evaporation rate knowing the partial pressure and vapor pressure of the substance. Langmuir conceived the liquid as a gas in which vapor was dispersed in accordance with the liquid's vapor pressure. Taking the difference of the flux of molecules moving from the liquid to the gas and from the gas to the liquid, he calculated the net evaporation rate:

$$\frac{dm}{dt} = (p_v - p_p) \sqrt{\frac{M}{2\pi RT}} \quad (\text{A1.7})$$

where dm/dt is the mass evaporation rate per unit area ($\text{kg/m}^2.\text{s}$)

p_v is the vapor pressure of the liquid at the given temperature

p_p is the partial pressure of the vapor in the ambient

M is the molar mass

R is the universal gas constant

T is the absolute temperature

This equation gives us the maximum possible evaporation rate. According to this equation, if $p_v > p_p$ then we will have net evaporation and if $p_p > p_v$, then the evaporation rate becomes negative, implying net condensation. Although the equation has been validated by many observations, this equation was found to predict evaporation rates 100 times higher than some experimental observations. For example, Hickman and Torpey found the equation to be valid for all liquids they tested, including water, if the interfacial surface is constantly renewed, thereby removing surface impurities that retard the evaporation process at the surface (Hickman & Torpey, 1954). Knudsen made a small modification to this equation providing more accurate evaporation rates (Knudsen, 1915).

$$\frac{dm}{dt} = \alpha(p_v - p_p) \sqrt{\frac{M}{2\pi RT}} \quad (\text{A1.8})$$

where α is referred to as the accommodation factor. It is defined as the fraction of liquid molecules hitting the liquid surface that will be accommodated by the vapor phase. It may also be defined as the probability that a molecule leaves the liquid phase and enters the vapor phase. Naturally, α has values from 0 to 1.

This equation is more commonly referred to as the Hertz-Knudsen equation. The importance of Knudsen's work was the introduction of the

accommodation factor, α , which he used to explain the difference in the calculated and experimental results.

Schrage assumed that the distribution of incoming molecules in the vapor phase could be represented by a Maxwellian distribution with a certain net bulk velocity (Schrage, 1953). He derived the formula:

$$\frac{dm}{dt} = \alpha p_v \sqrt{\frac{M}{2\pi RT}} \left[1 - \left(\frac{\gamma_0}{\gamma_s} \right) \left(\frac{T_0}{T_s} \right)^{1/2} \Gamma \right] \quad (\text{A1.9})$$

where the subscript s refers to the surface which is in contact with the gas and the subscript 0 refers to the apparent condition of the gas at the surface. The correction factor, Γ , was used to simplify the derivation of the equation and the author gives an equation to determine this factor. γ refers to the mass density. This equation reduces to the Hertz-Knudson expression, if both the correction factor and the temperature ratio T_0/T_s are assumed equal to one. However, he argues that there was no reason to assume that the temperature of the gas phase should be equal to the liquid phase. He also showed that this ratio would always be less than one regardless of evaporation or condensation.

The Hertz-Knudson and the Schrage formulae have been widely used to calculate the rate of liquid evaporation. However, the accuracy of the results of these equations relies on the values used for the mass accommodation factor and the parameters representing the conditions that exist at the interface of the evaporating liquid.

Fujikawa and Akamatsu (Fujikawa & Akamatsu, 1980) modified Schrage's equation to obtain the following expression.

$$\dot{m} = \frac{\alpha_m}{\sqrt{2\pi R_v}} \left(\frac{p_v^*}{\sqrt{T_{li}}} - \Gamma \frac{p_v}{\sqrt{T_{mi}}} \right) \quad (\text{A1.10})$$

where p_v is the actual vapor pressure, p_v^* is the equilibrium vapor pressure, T_{mi} and T_{li} are the temperatures of the vapor and liquid at the phase interface. R_v is the gas constant for the vapor. An equation to determine the correction factor, Γ , was also suggested. This equation is similar to the one obtained by Schrage, except that the correction factor is also a function of the evaporation rate and the vapor pressure. Hence, it becomes an implicit equation for the evaporation rate.

In analogy to the equation developed by Fujikawa and Akamatsu (Eq. A1.5), Bemporad proposed an equation to calculate the evaporation rate of water in an NVD environment (Bemporad, 1995):

$$\dot{m}_{ev} = A \times \alpha_m \times \left(f(C_s) \frac{p(T_s)}{\sqrt{T_s + 273}} - \frac{p(T_f)}{\sqrt{T_f + 273}} \right) \quad (\text{A1.11})$$

where A is the cross-sectional area of the evaporation section,

α_m is the mass accommodation factor, $\text{kg.m}^{-2}\text{Pa}^{-1}\text{s}^{-1}.\text{K}^{0.5}$

$f(C_s)$ is a correction factor for salt concentration,

T_s is the temperature of the salt-side (i.e. evaporation section temperature),

T_f is the temperature of the cooled distilled water side,

$p(T_s)$ and $p(T_f)$ are the saturation pressures of water at the two temperatures

The correction factor $f(C_s)$ is given by the following equation, as given by Bemporad:

$$f(C_s) = 1 - 0.0054C_s \quad (\text{A1.12})$$

Bemporad's equation does not take into account the pressure drop in the pipe connecting the saline water and the distilled water chambers. Al-Kharabsheh and Goswami (Al-Kharabsheh & Goswami, 2004) modified Bemporad's equation to include this pressure drop:

$$\dot{m}_{ev} = A \times \alpha_m \times \left(f(C_s) \frac{p(T_s)}{\sqrt{T_s + 273}} - \frac{p(T_f) + \Delta P}{\sqrt{T_f + 273}} \right) \quad (\text{A1.13})$$

Researchers working on NVD systems have mainly used Bemporad's equation for calculating the evaporation rate. Other equations are applicable in cases where water evaporates into its own vapor. In the cases of NVD, water evaporates from both the saline water and the distilled water columns. The net evaporation rate in this case, is controlled by the difference in the vapor pressures on the two columns, as shown by Eq. A1.7.

The mass accommodation factor

Several values of α_m have been reported in the literature. Earlier researchers have considered it unity or close to unity. Alty and Mackay (Alty & Mackay, 1935) carried out an experimental procedure to measure the coefficient. They determined the value to be 0.036. Schrage (Schrage, 1953) cited different values of α_m for water, but considered the value of 0.04 reported by Prüger (Prüger, 1940) to be more accurate because of the better and more reliable experimental procedure used for the determination of the coefficient. Hill (Hill, 1966) also cited many researchers but stuck to the value of 0.036 reported by Alty and Mackay.

Fujikawa and Akamatsu (Fujikawa & Akamatsu, 1980) have also reported many different values of α_m for water. They studied the effect of values of 0, 0.01, 0.04, 0.1 and 1.0 for the accommodation factor. They concluded that 0.04 was the most suitable value for water.

Davidovits et al. (2004) presented a review of works done on the mass accommodation coefficient of water vapor on liquid water. They concluded from their work that the different works cited in literature range over three orders of magnitude. Works have been cited which report values as low as 0.001 and going up to 1, for earlier measurements of the coefficient. More recent measurements done after 1987, however, still span two orders of magnitude, ranging between 0.01 and 1. This review was updated in 2011 by the same authors (Davidovits et al., 2011). They still reported the range of values (0.001 – 1) as the earlier review. However, they did cite recent studies, which reported values of 0.3, 0.57, 0.58, 0.62, and 1.0 for the mass accommodation coefficient of water.

Lotfi et al. (2014) carried out a recent study on evaporation of water from a free surface. Earlier works have all reported the coefficient independent of temperature. This study presents a relation for the evaporation coefficient changing with the T/T_c value (proximity to the critical condition). Values were determined as ranging from 1 for low temperatures ($T/T_c = 0.53$), 0.865 for medium temperatures ($T/T_c = 0.65$) and 0.555 for high temperatures ($T/T_c = 0.84$).

Bemporad (1995) has cited previous works for his equation but he says that the value of α_m for water should be determined experimentally, and it lies in the range of 10^{-7} to 10^{-6} . This is much lower than the lowest value (0.001) reported by any other works. He also does not have any experimental data to back his selection of the mentioned range of values. However, it is worth noting that he could have combined the actual α_m with other factors (as is evident from his equation and his choice of units for this dimensionless factor). The reason for this low value could also be the fact that the experimental setup that he has proposed in his study has a saline water column and a distilled water column. Evaporation takes place

simultaneously from both sides and he used his equation to predict the net rate of transfer of vapor from the saline water side (higher temperature) to the distilled water side. In the recent work by (Ambarita, 2017), a value of 9×10^{-9} was used for the accommodation factor. This very low value was used in the simulation to match the very low production rates determined by the experimental study.

Comparison of the evaporation equations

The mass rate of water evaporating into its own vapor was measured experimentally by Narusawa and Springer (1975). Selected data from their results was used to compare the evaporation rates calculated by the other equations. Fig. A1.1 shows a graphical comparison of these results.

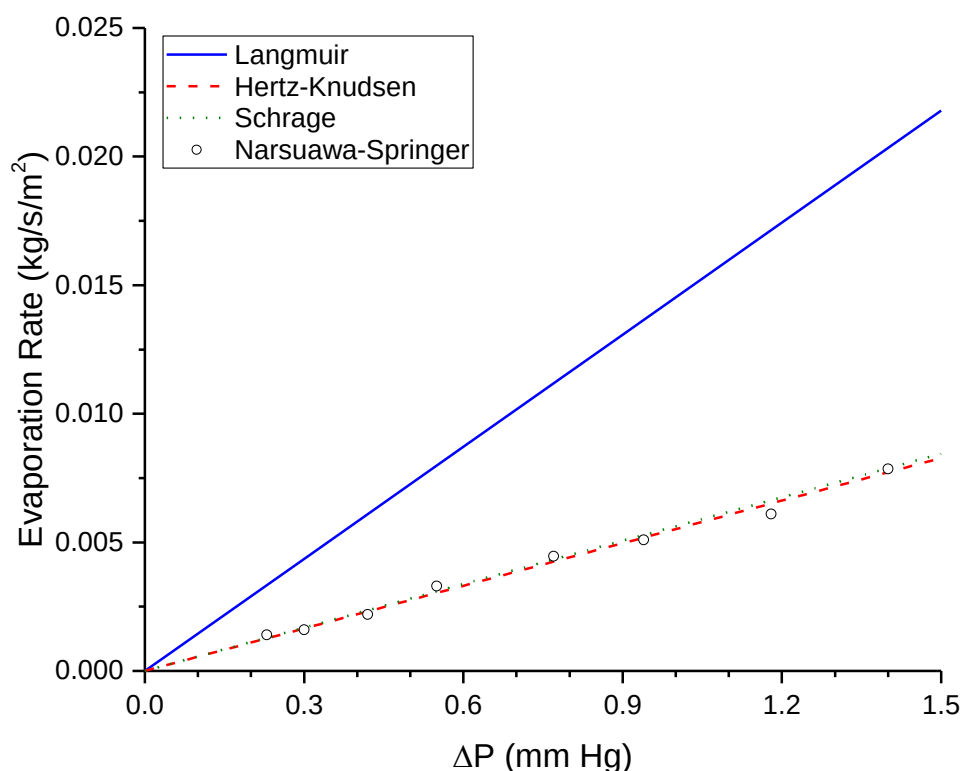


Figure A1.52: Comparison of evaporation rates by different equations

It can be seen from Fig. A1.1 that evaporation rate increases linearly with increase in the difference between the saturation pressure and the

vapor pressure of water in the system. Langmuir's equation does not take into consideration the mass accommodation factor for evaporation and hence we can see that the calculated evaporation rates are much higher than the experimental results. This was noted by Hertz (1882), when he observed that his experimental results were almost 10% of the maximum evaporation rates he calculated theoretically. Knudsen, later, used this idea and introduced the concept of the mass accommodation factor for evaporation (Knudsen, 1915). Results from Hertz-Knudsen equation and Schrage (1953) equation are almost identical with little differences. This is because of the assumption that the liquid and vapor are at the same temperature at the interface. According to Schrage, there does exist a temperature difference in the two phases at the interface, but with the lack of data for the phase interface, his equation was used with the assumption mentioned above. Other authors have noted negligible changes in the temperatures of the two phases including Narusawa and Springer (1975) who observed a maximum temperature difference of 0.5 °C between the two phases.

The experimental data in Fig. A1.1 is for the evaporation of water into its own vapor. The experimental data obtained for evaporation rates in an NVD system are presented in Fig. A1.2. Experimental data obtained by Alkharabsheh and Goswami (2003) and Gude and Nirmalakhandan (2009) are shown along with the theoretical limits given by the range of values for the accommodation factor reported by Bemporad (1995). Although, different values for the evaporation rates have been reported by the two studies, based on their experimental setup, it can be seen that results from both these experiments fall within the two bounds. While Alkharabsheh and Goswami (2003) did not report the value of the accommodation factor

used in their work, Gude and Nirmalakhandan (2009) have reported an experimentally determined value of 5×10^{-7} for their study.

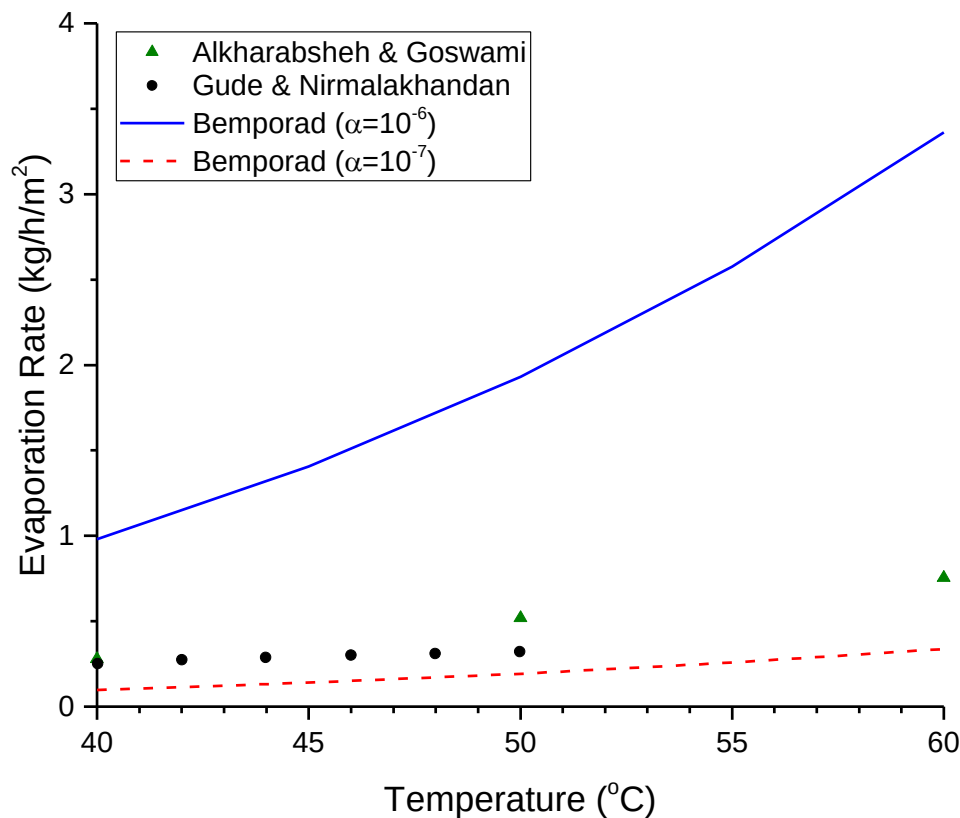


Figure A1.53: Comparison of experimental evaporation rates in NVD systems with the prediction of Bemporad's (1995) equation

The equation by Bemporad was not included in the comparison of Fig. A1.1, because his equation relates saturation pressures of saline water and distilled water in a vacuum chamber with a temperature difference maintained on both sides. His equation was compared to others by calculating evaporation rates, keeping the partial pressure of vapor constant at 7.5 mm Hg and calculating P_{sat} as a function of temperature. For Bemporad's equation, the distilled water was assumed to be held at a temperature of 10 °C and the saline water with a concentration of 3.5%. The results of the comparison are shown in Fig. A1.3.

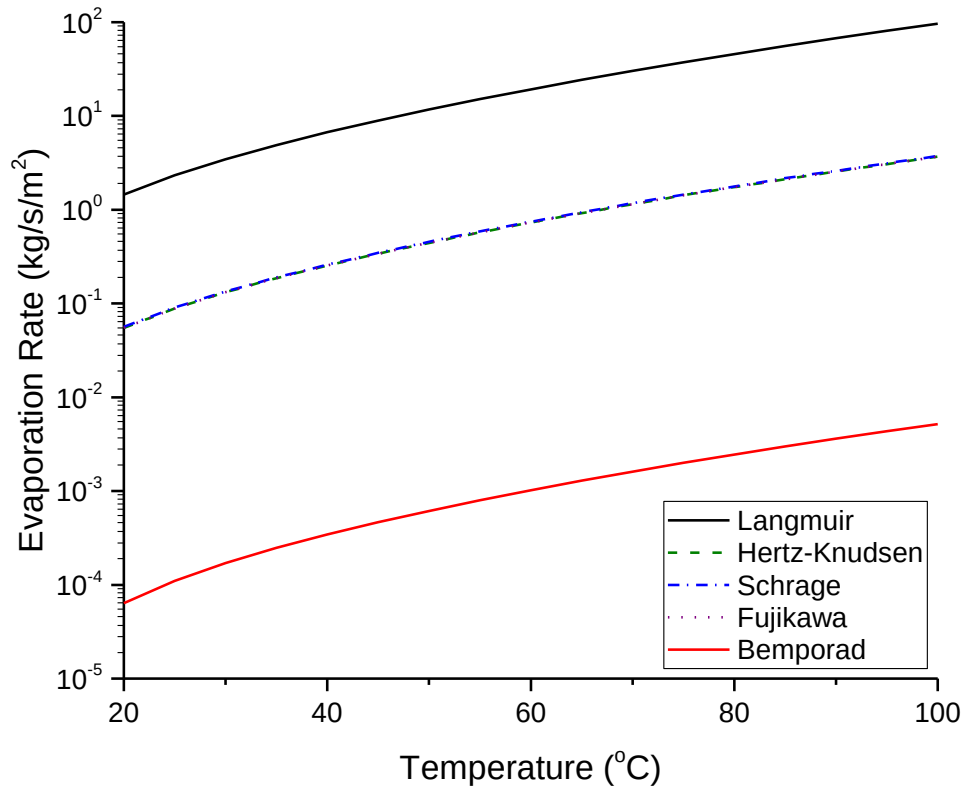


Figure A1.54: Evaporation rate as a function of temperature

Again, it can be seen that Langmuir's equation suggests high evaporation rates as discussed earlier. Results from Hertz-Knudsen, Schrage and Fujikawa are almost identical. The reason again is that for Schrage, temperature difference between the two phases was ignored and for Fujikawa, the correction factor was taken as one. If we look at the evaporation rates calculated by Bemporad's equation, we see that these are much lower than values suggested by others. The reason being that the values he has suggested for the accommodation factor are very low. He suggests a range of $10^{-7} - 10^{-6}$ for the values.

The mass accommodation factor is a dimensionless quantity but Bemporad reports it with units of $\text{kg m}^{-2} \text{Pa}^{-1} \text{s}^{-1} \text{K}^{0.5}$ hence it is obvious that he has combined the accommodation factor with other constants that appear in the equation. These are found in the equations as:

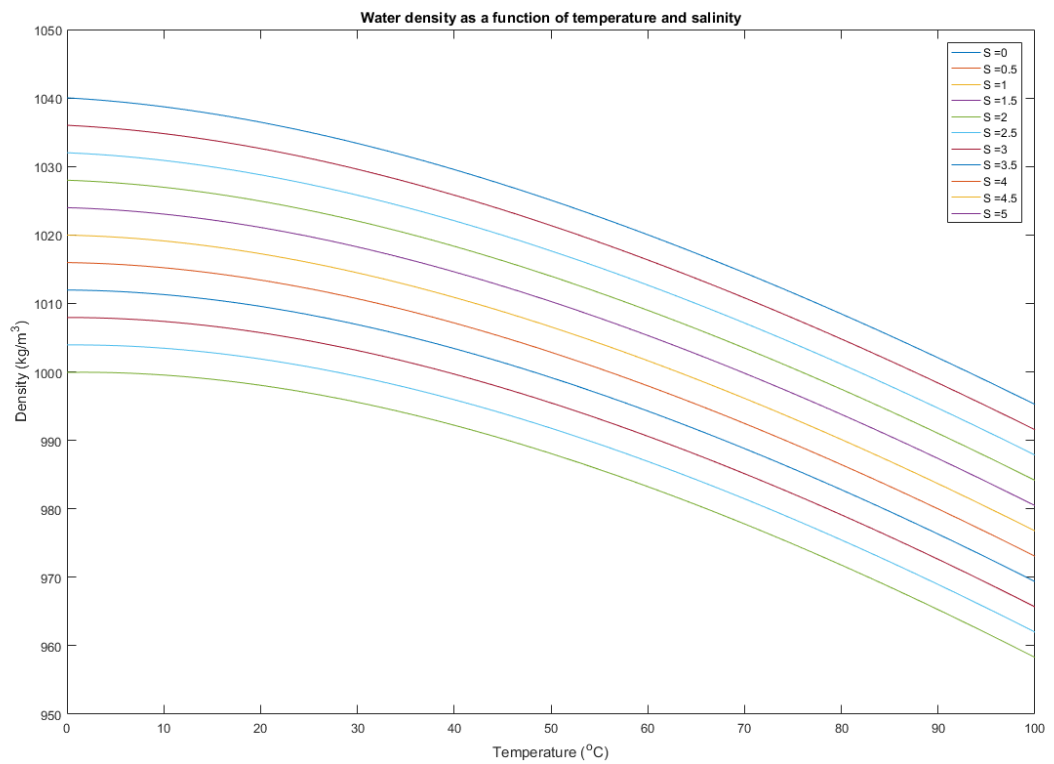
$$\sqrt{\frac{M}{2\pi R}} = \sqrt{\frac{0.018}{2(3.14159)(8.314)}} = 0.0186$$

Even if this value is combined with the value of mass accommodation factor (reported in the literature in the range of 0.01 – 1.0), we end with the value of 1.86×10^{-4} as the lowest value. Hence, the range suggested by Bemporad seems very low and hence shows very little evaporation rates. One important reason could be that the configuration studied by Bemporad is such that two water columns (saline water and distilled water) are held under near vacuum conditions and water is expected to evaporate from both surfaces. Hence, the evaporation rates here are the net evaporation rates that can be expected in such a system. Using the other equations and higher values of the mass accommodation factor with NVD would result in unrealistic evaporation rates.

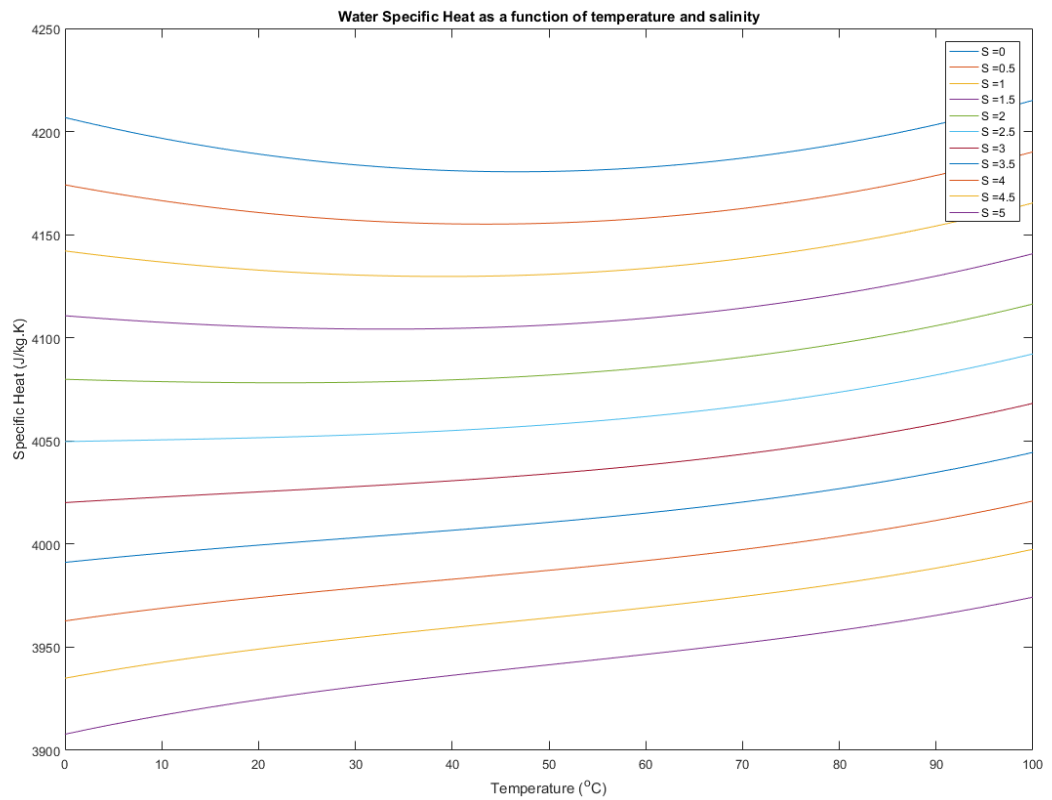
Appendix B

Thermophysical Properties of Seawater

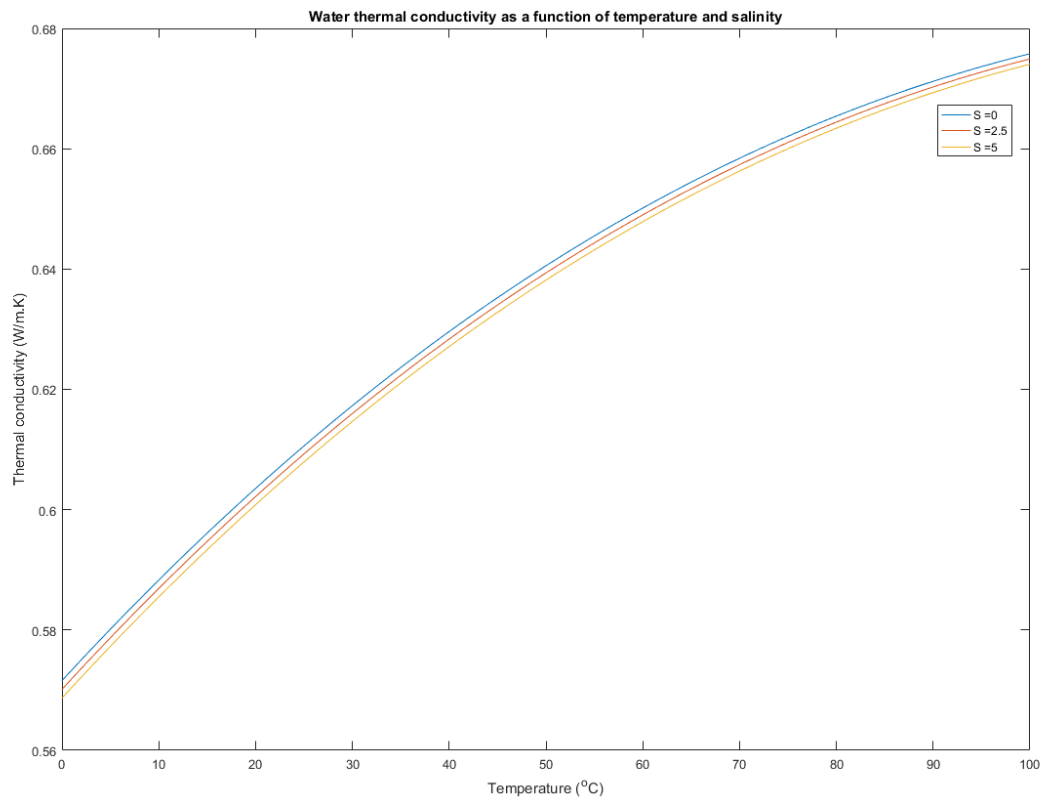
Water Density as a Function of Temperature and Salinity



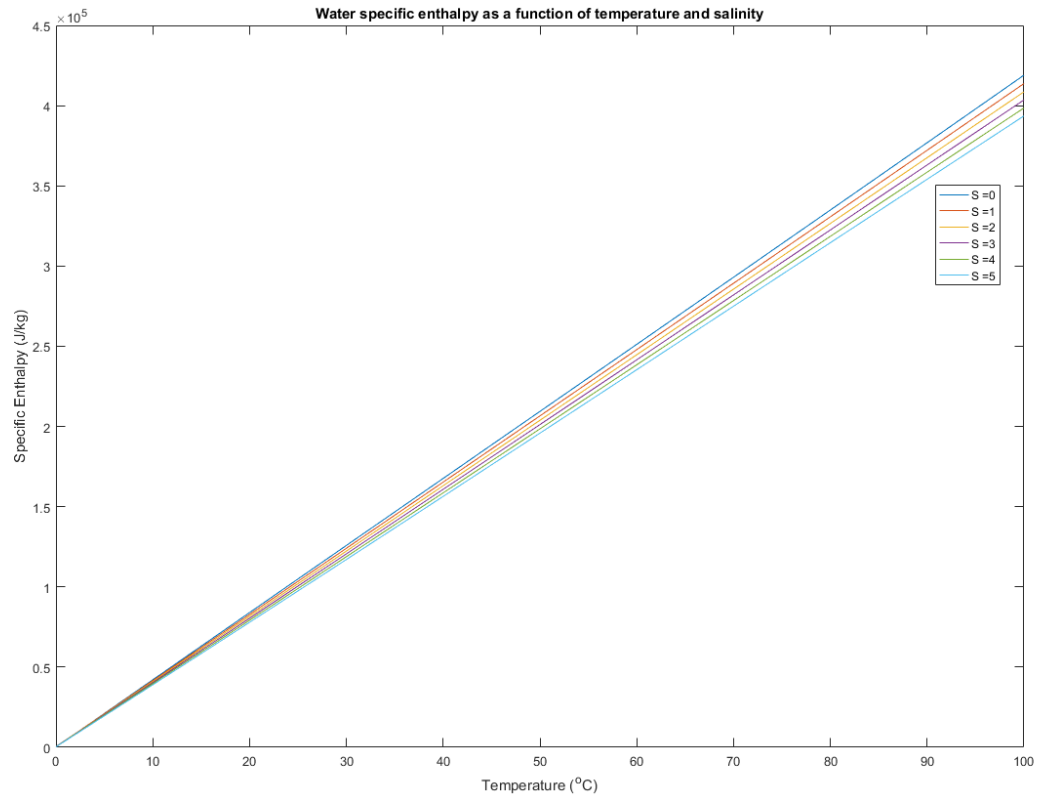
Water Specific Heat Capacity as a Function of Temperature and Salinity



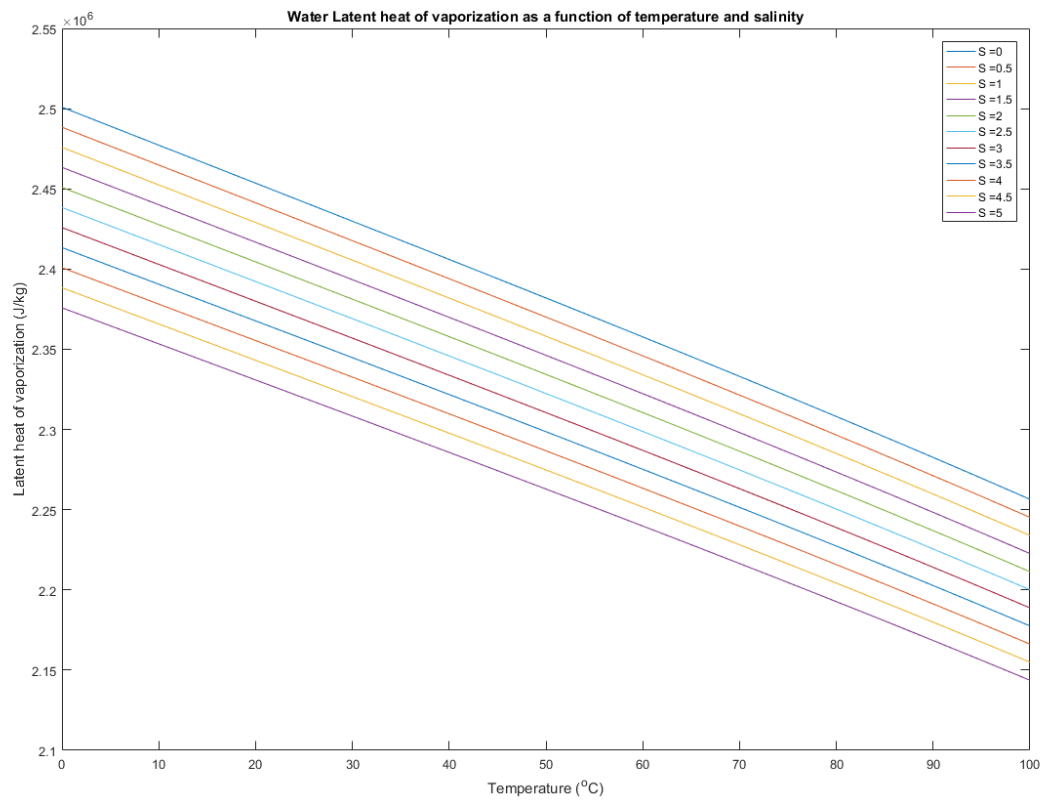
Water Thermal Conductivity as a Function of Temperature and Salinity



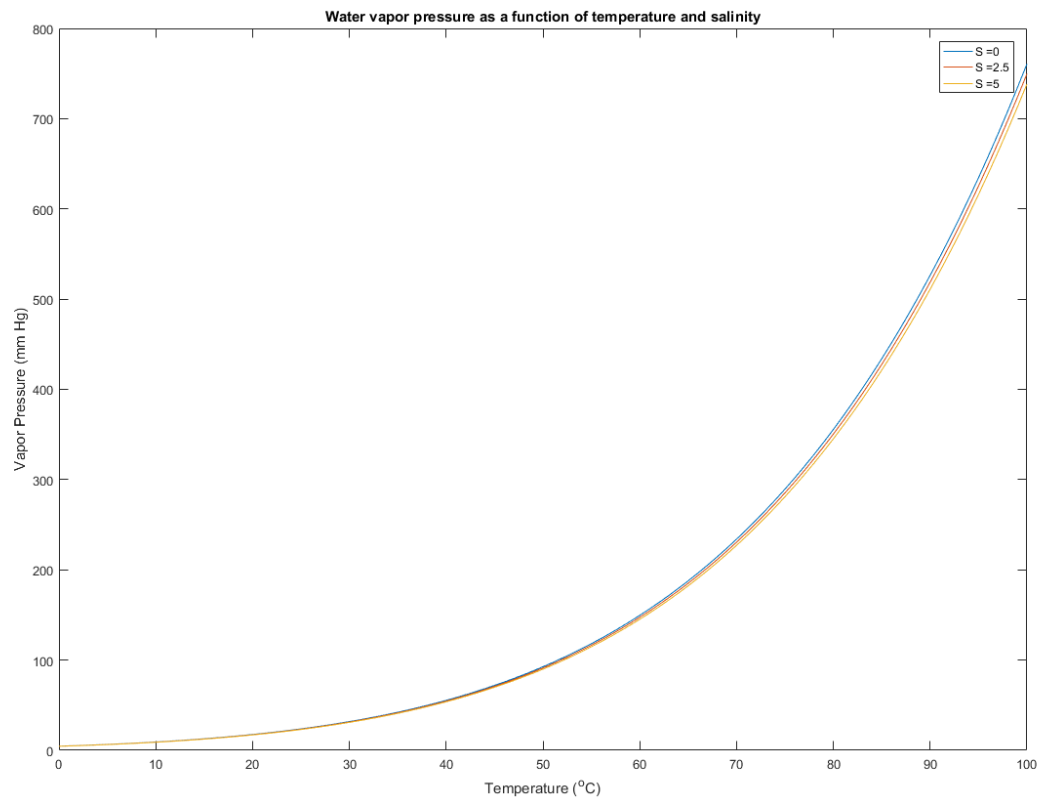
Water Specific Enthalpy as a Function of Temperature and Salinity



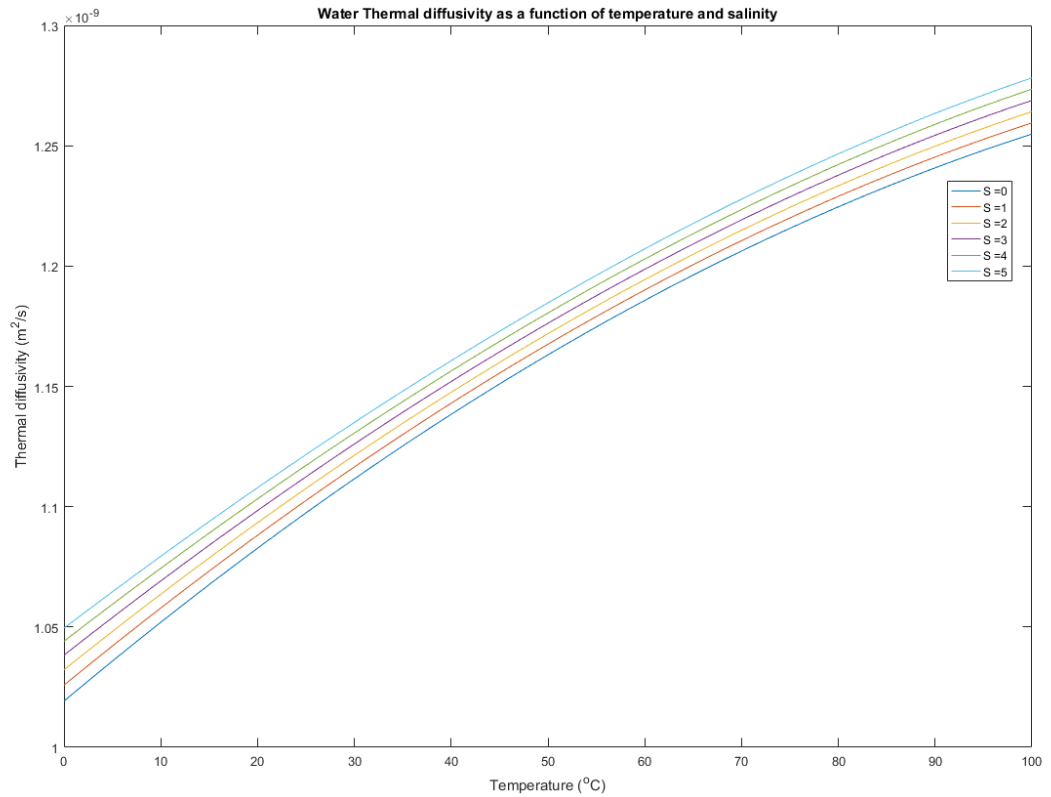
Water Latent Heat of Vaporization as a Function of Temperature and Salinity



Water Vapor Pressure as a Function of Temperature and Salinity



Water Thermal Diffusivity as a Function of Temperature and Salinity



الملخص

تقدم هذه الدراسة تحقيقاً تجريبياً ونظرياً لنظام تحلية جديد حيث يستخدم هذا النظام الجاذبية الأرضية والضغط الجوي لخلق حالة فراغ يمكنه من تبخير المياه في درجات حرارة منخفضة نسبياً حيث تسمح هذه الحرارة المنخفضة باستخدام مصادر حرارية منخفضة الجودة في عملية التحلية وتنفرد هذه الرسالة في اقتراح تصميم جديد ألا وهو الشكل مخروطي المقلوب الذي يسمح لوجود المبخر والمكثف معا تحت سقف واحد.

قد تم تصميم النموذج التجريبي بشكل كلي في هذه الدراسة حيث تم وضع خزان التبخر الرئيسي الذي يجسد غرفة التبخير وكذلك غرفة المياه المحلاة على ارتفاع ١٠,٣ متر فوق مستوى سطح الأرض لتحقيق التوازن بين الارتفاع البارومتري للمياه والضغط الجوي وتم توصيل غرفة التبخير إلى كل من خزان المياه المالحة وخزان المحلول الملحي وخزان المياه العذبة ، وتقع جميع هذه الخزانات على مستوى سطح الأرض.

وفي هذا البحث تم دراسة عملية تحلية المياه نظرياً وتجريبياً للتحقق في آثار ظروف التشغيل المختلفة على معدلات الإنتاج. وأجريت الدراسات التجريبية على نماذج مغلقة و مفتوحة لنظام التحلية و لوحظ أن العامل الرئيسي الذي يؤثر على معدل الإنتاج هو درجة حرارة المياه المالحة. وقد تم إنتاج المياه العذبة بمعدل ٢٤,٤ لتر/ يوم للنموذج المغلق و ١٥,٦٥ لتر/ يوم للنموذج المفتوح باستخدام درجة حرارة تسخين تبلغ ٧٠ درجة مئوية.

بالإضافة إلى ذلك، تم تطوير نموذج نظري باستخدام برنامج المحاكاة (COMSOL)، لمحاكاة أداء النموذج النظري مع التصميم التجريبي حيث إتفقت كلتا النتائج. وقد وجد أن كمية إنتاج المياه العذبة تتناسب مباشرة مع درجة حرارة التبخر وتتناسب عكسياً مع درجة حرارة المياه العذبة. كما وجد أن كمية إنتاج المياه العذبة تزداد بشكل غير خطي مع مساحة سطح التبخر وقد ارتفع الإنتاج والكفاءة الحرارية على حد سواء مع زيادة درجة حرارة المياه المالحة في النموذج المفتوح في حين تسببت زيادة درجة حرارة المياه المحلاة في انخفاض معدل الإنتاج والكفاءة الحرارية على حد سواء. وقد كان لمعدل سحب المحلول الملحي تأثير كبير على أداء النظام حيث بلغ النظام ذروة الكفاءة الحرارية عندما كان معدل سحب المحلول الملحي ٥ كغ / ساعة كما أثر عمق المياه في غرفة التبخر على معدل الإنتاج بطريقة مطردة.

جامعة البحرين

قسم الهندسة الكيميائية

كلية الهندسة



تحتية المياه باستخدام الطاقة الشمسية وتقنية التقطير الفراغي دراسة الأداء وتحديد الظروف المثلى

أطروحة مقدمة كجزء من متطلبات الحصول على درجة الدكتوراه
في الهندسة الكيميائية

إعداد

إعجاز رشيد غلام

الرقم الجامعي: ١٩٩٥٣٨٨٠

إشراف

د. عبد الرحيم عباس

(الأستاذ الدكتور)

د. تيومان ايهان

(الأستاذ الدكتور)

جامعة البحرين

مملكة البحرين

فبراير ٢٠١٨م