TRANSIENT ANALYSIS OF SOLAR DIFFUSION DRIVEN DESALINATION

By

FADI ALNAIMAT

A DISSERTATION PRESENTED TO THE GRADUATE SCHOOL OF THE UNIVERSITY OF FLORIDA IN PARTIAL FULFILLMENT OF THE REQUIREMENTS FOR THE DEGREE OF DOCTOR OF PHILOSOPHY

UNIVERSITY OF FLORIDA

2011
To my family
ACKNOWLEDGMENTS

First and foremost, I must thank Dr. James Klausner for providing me this opportunity to work with him. Thanks for the technical assistance and for giving me the opportunity to do research in the field of multiphase flow and heat transfer. His inspiration and guidance have enabled me to achieve my goals. Thank you. I would like to express my appreciation to Professor Renwei Mei for his technical assistance in this research. I would like to thank Dr. David Hahn and Dr. Ben Koopman, for serving on my supervisory committee. I would like to thank the Middle East Desalination Research Center (MEDRC) for funding this project.
# TABLE OF CONTENTS

<table>
<thead>
<tr>
<th>Table of Contents</th>
<th>Page</th>
</tr>
</thead>
<tbody>
<tr>
<td>ACKNOWLEDGMENTS</td>
<td>4</td>
</tr>
<tr>
<td>LIST OF TABLES</td>
<td>7</td>
</tr>
<tr>
<td>LIST OF FIGURES</td>
<td>8</td>
</tr>
<tr>
<td>LIST OF ABBREVIATIONS</td>
<td>12</td>
</tr>
<tr>
<td>ABSTRACT</td>
<td>15</td>
</tr>
<tr>
<td>CHAPTER</td>
<td></td>
</tr>
<tr>
<td>1 INTRODUCTION TO FRESHWATER SHORTAGES AND CURRENT DESALINATION TECHNOLOGIES</td>
<td>17</td>
</tr>
<tr>
<td>Freshwater Demand and Scarcity</td>
<td>18</td>
</tr>
<tr>
<td>Desalination Technologies</td>
<td>20</td>
</tr>
<tr>
<td>Energy and Desalination</td>
<td>26</td>
</tr>
<tr>
<td>Seawater Properties and Desalination Challenges</td>
<td>30</td>
</tr>
<tr>
<td>Environmental Impact of Desalination</td>
<td>36</td>
</tr>
<tr>
<td>Renewable Energy Utilization for Seawater Desalination</td>
<td>38</td>
</tr>
<tr>
<td>Solar Diffusion Driven Desalination Process</td>
<td>44</td>
</tr>
<tr>
<td>Research Objectives and Scope of Wok</td>
<td>46</td>
</tr>
<tr>
<td>2 HEAT AND MASS TRANSFER WITHIN THE DIRECT CONTACT EVAPORATOR AND CONDENSER</td>
<td>49</td>
</tr>
<tr>
<td>Theoretical Heat and Mass Transfer Formulation</td>
<td>49</td>
</tr>
<tr>
<td>Evaporator</td>
<td>50</td>
</tr>
<tr>
<td>Condenser</td>
<td>58</td>
</tr>
<tr>
<td>Heat and Mass transfer Coefficients</td>
<td>62</td>
</tr>
<tr>
<td>Storage Tank and Solar Collector</td>
<td>63</td>
</tr>
<tr>
<td>Solution of System of Equations</td>
<td>66</td>
</tr>
<tr>
<td>Computational Method</td>
<td>66</td>
</tr>
<tr>
<td>Typical Computational Results</td>
<td>67</td>
</tr>
<tr>
<td>3 EXPERIMENTAL STUDY</td>
<td>74</td>
</tr>
<tr>
<td>Experimental Apparatus</td>
<td>74</td>
</tr>
<tr>
<td>Experimental and Computational Results</td>
<td>79</td>
</tr>
<tr>
<td>Evaporator Temperature Profiles for Different Air Mass Fluxes</td>
<td>79</td>
</tr>
<tr>
<td>Evaporator Temperature Profiles for Different Water Mass Fluxes</td>
<td>80</td>
</tr>
<tr>
<td>Condenser Temperature Profiles for Different Air Mass Fluxes</td>
<td>80</td>
</tr>
<tr>
<td>Condenser Temperature Profiles for Different Water Mass Fluxes</td>
<td>81</td>
</tr>
<tr>
<td>Table</td>
<td>Description</td>
</tr>
<tr>
<td>---------</td>
<td>----------------------------------------------------------------------------------------------</td>
</tr>
<tr>
<td>1-1</td>
<td>Common desalination technologies</td>
</tr>
<tr>
<td>1-2</td>
<td>Key operational data of different desalination technologies [13]</td>
</tr>
<tr>
<td>1-3</td>
<td>Feed water salinities [14]</td>
</tr>
<tr>
<td>1-4</td>
<td>Chemical constituents in seawater salt [15]</td>
</tr>
<tr>
<td>5-1</td>
<td>Standard operating conditions and configuration for the solar DDD</td>
</tr>
<tr>
<td>6-1</td>
<td>Components cost for the solar DDD in the U.S. and China</td>
</tr>
<tr>
<td>6-2</td>
<td>Water cost for solar DDD and RO processes</td>
</tr>
<tr>
<td>6-3</td>
<td>Water cost in some countries around the world</td>
</tr>
</tbody>
</table>
# LIST OF FIGURES

<table>
<thead>
<tr>
<th>Figure</th>
<th>Description</th>
<th>Page</th>
</tr>
</thead>
<tbody>
<tr>
<td>1-1</td>
<td>Map of the water scarcity index [2]</td>
<td>19</td>
</tr>
<tr>
<td>1-2</td>
<td>MSF process [5]</td>
<td>21</td>
</tr>
<tr>
<td>1-3</td>
<td>MED process (a) Horizontal falling film MED, (b) Flow diagram of MED [6]</td>
<td>22</td>
</tr>
<tr>
<td>1-4</td>
<td>Mechanical vapor compression process [7]</td>
<td>23</td>
</tr>
<tr>
<td>1-5</td>
<td>Thermal vapor compression combined with multi effect distillation process [8]</td>
<td>24</td>
</tr>
<tr>
<td>1-6</td>
<td>Schematic diagram of Electrodialysis cell [9]</td>
<td>25</td>
</tr>
<tr>
<td>1-8</td>
<td>Market shares of different water desalination technologies worldwide [12]</td>
<td>28</td>
</tr>
<tr>
<td>1-9</td>
<td>Solar intensity distribution around in the U.S. [26]</td>
<td>40</td>
</tr>
<tr>
<td>2-1</td>
<td>Countercurrent flow through a packed bed evaporator</td>
<td>51</td>
</tr>
<tr>
<td>2-2</td>
<td>Evaporator differential control volume depicting liquid/vapor/solid</td>
<td>52</td>
</tr>
<tr>
<td></td>
<td>interactions.</td>
<td></td>
</tr>
<tr>
<td>2-3</td>
<td>Condenser differential control volume depicting liquid/vapor/solid</td>
<td>58</td>
</tr>
<tr>
<td></td>
<td>interactions.</td>
<td></td>
</tr>
<tr>
<td>2-4</td>
<td>Specified evaporator inlet water temperature.</td>
<td>69</td>
</tr>
<tr>
<td>2-5</td>
<td>Temporal and spatial water temperature distribution along the height of the</td>
<td>70</td>
</tr>
<tr>
<td></td>
<td>evaporator.</td>
<td></td>
</tr>
<tr>
<td>2-6</td>
<td>Temporal and spatial air temperature distribution along the height of the</td>
<td>70</td>
</tr>
<tr>
<td></td>
<td>evaporator.</td>
<td></td>
</tr>
<tr>
<td>2-7</td>
<td>Temporal and spatial packing temperature distribution along the height of</td>
<td>71</td>
</tr>
<tr>
<td></td>
<td>the evaporator.</td>
<td></td>
</tr>
<tr>
<td>2-8</td>
<td>Temporal and spatial humidity ratio distribution along the height of the</td>
<td>71</td>
</tr>
<tr>
<td></td>
<td>evaporator.</td>
<td></td>
</tr>
<tr>
<td>2-9</td>
<td>Temporal and spatial water temperature distribution along the height of the</td>
<td>72</td>
</tr>
<tr>
<td></td>
<td>direct contact condenser.</td>
<td></td>
</tr>
<tr>
<td>2-10</td>
<td>Temporal and spatial air temperature distribution along the height of the</td>
<td>72</td>
</tr>
<tr>
<td></td>
<td>direct contact condenser.</td>
<td></td>
</tr>
</tbody>
</table>
2-11 Temporal and spatial packing temperature distribution along the height of the condenser ................................................................. 73

2-12 Temporal and spatial humidity ratio distribution along the height of the condenser ................................................................. 73

3-1 Schematic diagram of the experimental direct contact evaporator and condenser desalination facility ................................................................. 75

3-2 Solar diffusion driven desalination facility ........................................................................................................................................ 76

3-3 Simulated Solar Thermal Power for 10 m² of Collector Area .................................................................................................................. 76

3-4 Schematic diagram of the solid state relay connected to a data acquisition system .................................................................................................................................................................................................................. 77

3-5 Polypropylene packing material ......................................................................................................................................................... 78

3-6 Evaporator transient temperature and humidity ratio variation (G=0.5 kg/m²-s). ............................................................................................................................................................................................................... 82

3-7 Evaporator transient temperature and humidity ratio variation (G=1.02 kg/m²-s). ............................................................................................................................................................................................................... 82

3-8 Evaporator transient temperature and humidity ratio variation (L=1.0 kg/m²-s). ............................................................................................................................................................................................................... 83

3-9 Evaporator transient temperature and humidity ratio variation (L=2.15 kg/m²-s). ............................................................................................................................................................................................................... 83

3-10 Condenser transient temperature and humidity ratio variation (G=1.02 kg/m²-s). ............................................................................................................................................................................................................... 84

3-11 Condenser transient temperature and humidity ratio variation (G=0.5 kg/m²-s). ............................................................................................................................................................................................................... 84

3-12 Condenser transient temperature and humidity ratio variation (L=2.0 kg/m²-s). ............................................................................................................................................................................................................... 85

3-13 Condenser transient temperature and humidity ratio variation (G=1.0 kg/m²-s) ............................................................................................................................................................................................................... 85

4-1 Dynamic response of evaporator exit water and air temperatures for different liquid hold-up .................................................................................................................. 91

4-2 Dynamic response of evaporator exit water and air temperatures for a_w/a=0.5 and a_w/a=1.0 ............................................................................................................................................................................................................... 92
4-3 Dynamic response Evaporator exit water and air temperature for different packed bed material. ............................................................... 92
4-4 Evaporator exit water and air temperature responses to a step change in the inlet water temperature, \( L/G=2.0 \). ................................................................. 93
4-5 Evaporator exit water and air temperature responses due to a step change in the inlet air temperature, \( L/G=2.0 \). ................................................................. 93
4-6 Evaporator exit water and air temperature responses due to a step change in the inlet water temperature, \( L/G=0.5 \). ................................................................. 94
4-7 Evaporator exit water and air temperature responses due to a step change in the inlet air temperature, \( L/G=0.5 \). ................................................................. 94
5-1 Schematic diagram of the solar DDD ......................................................... 95
5-2 Air pressure drop in the packed bed for different water and air fluxes ............... 97
5-3 Solar heat input and fresh water production rate ........................................ 99
5-4 Total fresh water produced and saline water volume in the saline water storage tank ...................................................................................... 100
5-5 Variation in fresh water production efficiency a) with respect to temperature, b) with respect to time ................................................................. 100
5-6 Energy consumption and the specific energy consumption for the solar DDD. 101
5-7 Total fresh water produced for various saline water storage tank sizes. ....... 102
5-8 Total fresh water produced for various solar collector efficiencies .................. 103
5-9 Fresh water production rate for different condenser inlet water temperatures.. 104
5-10 Total fresh water production rate for different condenser inlet water temperatures. ............................................................................................... 105
5-11 Saline water tank temperature for different condenser inlet water temperatures. ............................................................................................... 105
5-12 Fresh water production efficiency and condenser exit air temperature for various condenser inlet water temperatures. ........................................ 106
5-13 Condenser water temperature and heat removed ........................................ 107
5-14 Fresh water production rate and total fresh water produced. ......................... 107
5-15 Fresh water temperature and solar heat input ............................................. 109
5-16 Fresh water production rates and total fresh water produced ......................... 110
5-17 Saline water tank temperature history .............................................................. 110
5-18 Fresh water production rates for various saline water storage tank sizes .......... 111
5-19 Total fresh water produced for various saline water storage tank sizes .......... 111
5-20 Temperature in the system for various fresh water storage tank sizes ............ 112
5-21 Total fresh water produced for various fresh water storage tank sizes ............ 113
5-22 Specific electrical energy consumption .......................................................... 113
5-23 Evaporator water and air temperatures .......................................................... 114
5-24 Condenser water and air temperatures ............................................................ 115
5-25 Water temperatures and water production rates with delayed operating time .. 117
5-26 Fresh water production and fresh water production efficiency with delayed operating time ................................................................................................... 117
5-27 Energy consumption with delayed operating time ............................................ 118
5-28 Fresh Water Production with delayed operating time for various tank sizes..... 118
5-29 Energy consumption with delayed operating time for various tank sizes ......... 119
5-30 Seawater salinity in the solar DDD ................................................................... 121
# LIST OF ABBREVIATIONS

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>A</td>
<td>Surface area ((m^2))</td>
</tr>
<tr>
<td>(A_c)</td>
<td>Cross sectional area of the packing ((m^2))</td>
</tr>
<tr>
<td>a</td>
<td>Specific area of packing material ((m^2/m^3))</td>
</tr>
<tr>
<td>C</td>
<td>Salinity (ppm)</td>
</tr>
<tr>
<td>CF</td>
<td>Concentration factor</td>
</tr>
<tr>
<td>(C_p)</td>
<td>Specific heat (kJ/kg or J/kg)</td>
</tr>
<tr>
<td>D</td>
<td>Molecular diffusion coefficient ((m^2/s))</td>
</tr>
<tr>
<td>d</td>
<td>Diameter (m)</td>
</tr>
<tr>
<td>(d_p)</td>
<td>Diameter of the packing material (m)</td>
</tr>
<tr>
<td>dz</td>
<td>Height of the differential volume (m)</td>
</tr>
<tr>
<td>G</td>
<td>Air mass flux (kg/m(^2)-s)</td>
</tr>
<tr>
<td>g</td>
<td>Gravitational acceleration (m/s(^2))</td>
</tr>
<tr>
<td>H</td>
<td>Evaporator or condenser height (m)</td>
</tr>
<tr>
<td>h</td>
<td>Enthalpy (kJ/kg)</td>
</tr>
<tr>
<td>(h_{fg})</td>
<td>Latent heat of vaporization (kJ/kg)</td>
</tr>
<tr>
<td>k</td>
<td>Mass transfer coefficient (m/s)</td>
</tr>
<tr>
<td>L</td>
<td>Water mass flux (kg/m(^2)-s)</td>
</tr>
<tr>
<td>(M_v)</td>
<td>Vapor molecular weight (kg/kmol)</td>
</tr>
<tr>
<td>(\dot{m})</td>
<td>Mass flow rate (kg/s)</td>
</tr>
<tr>
<td>P</td>
<td>Total pressure (Pa or kPa)</td>
</tr>
<tr>
<td>R</td>
<td>Universal gas constant (kJ/kmol-K)</td>
</tr>
<tr>
<td>T</td>
<td>Temperature ((^\circ)C or (^\circ)K)</td>
</tr>
<tr>
<td>U</td>
<td>Heat transfer coefficient (W/m(^2)-K)</td>
</tr>
<tr>
<td>v</td>
<td>Air/vapor velocity (m/s)</td>
</tr>
<tr>
<td>V</td>
<td>Control volume (m(^3))</td>
</tr>
<tr>
<td>(V_G)</td>
<td>Air/vapor volumetric flow rate (m(^3)/s)</td>
</tr>
<tr>
<td>(\mu)</td>
<td>Dynamic viscosity (kg/m-s)</td>
</tr>
<tr>
<td>(\rho)</td>
<td>Density (kg/m(^3))</td>
</tr>
<tr>
<td>(\sigma_L)</td>
<td>Surface tension of liquid (N/m)</td>
</tr>
<tr>
<td>(\sigma_C)</td>
<td>Critical surface tension of the packing material (N/m)</td>
</tr>
</tbody>
</table>
ω  Humidity ratio
Φ  Relative humidity
NuL  Nusselt number
PrL  Prandtl number
ShL  Sherwood number
ScL  Schmidt number
Fr  Froude number
We  Webber number
Re  Reynold number
q_{max}  Maximum specific heat input (kJ/kg)
t’  Dimensionless time
T_L*  Dimensionless liquid temperature
T_a*  Dimensionless air temperature
T_{pack}  Dimensionless packed bed temperature
z  Distance
z’  Dimensionless distance
α_L  Liquid volume fraction (m³/m³),
α_{mix}  Air/vapor mixture volume fraction (m³/m³).
α_{pack}  Packed bed volume fraction (m³/m³).
η_L  Dimensionless liquid enthalpy
η_{fg}  Dimensionless latent heat
η_V  Dimensionless vapor enthalpy
η_{fw}  Fresh water production efficiency

SUBSCRIPTS
a  Air
cond  Condenser
evap  Evaporator
fw  Fresh water
sw  Saline water
salt  Salt in seawater
G    Air/vapor mixture
h    High
i    Interface
in   Inlet parameter
L    Liquid phase
low  Low
out  Exit parameter
sat  Saturate state
V    Vapor phase
mix  Mixture (gas and vapor)
solar Solar collector
This research concerns a transient analysis of the solar diffusion driven desalination (DDD) process, in which solar energy is utilized as the driving source to distill seawater. The process is based on evaporating seawater into a dry air stream using a packed bed, and then condensing out the vapor to produce freshwater. A rigorous theoretical framework for the transient heat and mass transfer during the evaporation and condensation within backed beds is developed in this work. The analysis relies on one-dimensional conservation equations, and as such closure relationships are required. The conservation equations are solved numerically using a finite difference scheme to predict water, air/vapor mixture and packed bed temperatures and humidity ratio within the evaporator and the condenser. The heat and mass transport models account for the transient variations within the packed-bed due to time varying inlet air and water temperatures and humidity. A laboratory scale solar simulated DDD facility has been constructed to operate in a re-circulating mode. Detailed experiments have been run to examine the efficacy of the analysis. Transient heating to the evaporator water varies from 1 to 8 kW over a 2.5 hour time span. Water mass fluxes ranging from 1.0 to 2.15 kg/m²-s and air mass fluxes ranging from 0.5 to
1.02 kg/m²-s are considered. The measured exit water and air/vapor mixture
temperatures and humidity ratio agree well with those predicted using the transient
model. The model appears to provide a very good predictive capability for the thermal
and mass transport within the evaporator and condenser. The model is utilized as a
design tool to identify the optimal operating conditions and analyze the performance of
the desalination processes. The fundamental transient analysis should find utility in a
broad variety of applications. The performance of the solar DDD has been investigated
under the different operating conditions. Several methods have been proposed to
improve the water production and reduce the specific energy consumption. It has been
demonstrated that the system energy consumption can be reduced significantly as low
as 1.65 kWh/m³. The system water production can be improved up to 7.5 Liter per m²
of solar collector area. Economic analysis on a small scale solar DDD unit has been
carried out. It is believed that solar DDD can be promising for small scale applications.
CHAPTER 1
INTRODUCTION TO FRESHWATER SHORTAGES AND CURRENT DESALINATION TECHNOLOGIES

Water scarcity, combined with a shortage of energy, should be regarded as a future threat to human activity and development in any country or civilization around the world. The world fresh water supply is diminishing, and in the next few years many countries in the world are expected to face water shortages that will significantly influence their economic and population growth. Water problems are attracting the attention of the public, researchers, and decision makers worldwide. Supplying sufficient quality water is essential to the development of any country. One of the main drivers of water shortages is an ever-increasing population. By 2025, more than 3 billion people could be living in water-stressed countries, and 14 countries will slip from water stress to water scarcity [1].

Water covers about two-thirds of the earth surface. Approximately 97% of the water on the planet is saline. The remaining 3% is fresh water, where the majority of it, 69% of the fresh water, is locked up in glaciers and icecaps, and 30% is ground water. About 1% of the fresh water is surface water in the form of lakes, rivers, and swamps. Water scarcity occurs in both arid and non-arid regions, and is created when there is disparity between water supply and demand. Intensive withdrawal of groundwater aquifer or surface water results in a significant decrease in the water reserve and its quality. Around the world, irrigation is the largest freshwater consumer, where 60% of freshwater consumed is used for irrigation. The remaining portion is distributed among domestic use, industrial use, electricity production, livestock use, and mining use. Furthermore, water pollution is considered a major problem for many countries
worldwide. Since the inception of the industrial revolution, industrial plants have created more waste that directly or indirectly contaminate the water supply.

Numerous countries in the world are constantly searching for new sources of water. In the Arabian Gulf region, overdrawing of groundwater and continuing population growth have created a need for new water sources. The long-term survivability of such countries relies upon novel ways to deal with the growing water deficiency. The best way to ensure a sustainable future is to utilize alternative energy technologies to provide an inexpensive power supply that can be used to produce freshwater. Water desalination has emerged as an alternative source of freshwater and holds a promising solution for water scarcity.

**Freshwater Demand and Scarcity**

The heading above shows that if you have a subheading of a certain level, you must have more than one. The rationale is that you cannot have a list of only one item. Water scarcity is commonly represented by the water stress index (WSI), which is defined as the average amount of fresh water withdrawal to the amount of long-term available water resources. Fig.1-1 shows a map for the water scarcity index around the world. As shown, many areas around the world such as the Middle East, Northern China, areas of North Africa, and the middle and western parts of the United States are considered be a water impoverished area. The more the country is scarce of water, the greater the WSI is. Regions that severely lack freshwater or overly exploit their water resources have a WSI greater than 1.

Desalination is defined as the removal of salt from seawater or brackish water to produce freshwater. In the last few decades, the tendency toward utilizing desalination technologies to alleviate water scarcity has intensified in order to create new sources of
freshwater to meet the essential water needs. Desalination in general requires large amounts of energy. Energy is essential for producing freshwater using any desalination technologies, and when there is a shortage of either one, crises arise.

![Map of the water scarcity index](image)

**Figure 1-1. Map of the water scarcity index [2]**

Desalination is defined as the removal of salt from seawater or brackish water to produce freshwater. In the last few decades, the tendency toward utilizing desalination technologies to alleviate water scarcity has intensified in order to create new sources of freshwater to meet the essential water needs. Desalination in general requires large amounts of energy. Energy is essential for producing freshwater using any desalination technologies, and when there is a shortage of either one, crises arise.

To produce potable freshwater, countries are depending on desalination to meet their needs. Water desalination has been of interest to many countries where there exists a scarcity of potable water. In particular, desalination is the main source of water for arid countries in the Middle East, arid remote areas, and some islands. For many
decades, these countries have been exploring the use of desalination to produce fresh water. Half of all the world’s desalination capacity is in the Middle East, Persian Gulf and North Africa regions [3]. Saudi Arabia leads the world by having the largest desalination capacity followed by the USA and UAE, respectively.

**Desalination Technologies**

Desalination technologies can be classified based on their separation mechanism into thermal desalination and membrane desalination. The most frequently applied desalination techniques are Reverse Osmosis (RO), Multi-Stage Flash (MSF), Multiple-Effect Distillation (MED), Electrodialysis (ED), and Vapor Compression (VP). Table 1 shows an overview categorization of the most commonly used desalination technologies.

Table 1-1. Common desalination technologies

<table>
<thead>
<tr>
<th>Thermal Desalination</th>
<th>Membrane Desalination</th>
</tr>
</thead>
<tbody>
<tr>
<td>Multi-Stage Flash (MSF)</td>
<td>Reverse Osmosis (RO)</td>
</tr>
<tr>
<td>Multi-Effect Distillation (MED)</td>
<td>Electrodialysis (ED)</td>
</tr>
<tr>
<td>Vapor Compression (VC)</td>
<td></td>
</tr>
<tr>
<td>Solar Still</td>
<td></td>
</tr>
</tbody>
</table>

Thermal desalination involves a phase change process between liquid and vapor by which freshwater is produced. Membrane desalination utilizes a membrane to separate the salt from the water. In general, thermal desalination is more energy intensive than membrane desalination. On the other hand, thermal desalination can handle more salt concentration of seawater than membrane desalination and at the same time produce better water quality. Amongst all of the desalination techniques, RO and MSF are the most commonly used desalination technologies.

Multi-Stage Flash (MSF), among all thermal desalination techniques, is the most favorable desalination technique in the Middle East and Gulf region. MSF produces
nearly 40% of the total desalinated water worldwide [4]. The principle of the MSF process is based on generating water vapor by heating seawater and introducing it to an evacuated chamber, where a sudden pressure drop takes place and water splashes to vapor. This process is repeated successively in a series of chambers in which the pressure is reduced at different stages. Water vapor is condensed on a heat exchanger bundle and collected to produce freshwater. Usually, the seawater is heated to 100°C prior entering the MSF process. Scaling occurs on the surfaces of the desalination system components when salts like sulfates and carbonates exist at high concentration and temperature, which causes mechanical and thermal problems. In order to avoid scaling, an upper limit of the salt concentration must be imposed, which limits the performance of the process. MSF is a widely established and mature technology. Fig.1-2 shows a schematic diagram of a typical MSF process.
The Multiple-Effect Distillation (MED) process operates in a similar fashion to MSF, where the process takes place in a series of vessels collectively referred to as an effect. The brine is introduced to the first effect, which is usually sprayed on bundle of tubes or plates, to create a falling film. Fig.1-3 shows a schematic diagram of MED [6]. Water vapor is generated at the falling film and thereafter directed to the next effect where it releases its latent heat to condensation to the incoming brine in the second effect. This result in condensing the water vapor generated from the previous effect and at the same time recuperating heat for the following stage [6]. This process is repeated in a series of effects where the temperature of the brine decreases successively.

Figure 1-3. MED process (a) Horizontal falling film MED, (b) Flow diagram of MED [6]
Vapor compression (VC) is another thermal distillation process that utilizes the latent heat of condensation of water vapor as its primary heat source. VC systems are categorized into mechanical vapor compression (MVC), which utilizes a mechanical compressor, and thermal vapor compression (TVC), which utilize a steam jet compressor. Compressed water vapor flows inside a bundle of tubes, which results in high temperature condensation. The latent heat of condensation transfers through the tube bundle to a brine that is usually sprayed on the outside surface of the tubes. The brine is evaporated and then enters the condenser. Fig.1-4 shows a schematic diagram of mechanical vapor compression process [7].

![Figure 1-4. Mechanical vapor compression process [7]](image)

Thermal vapor compression is usually coupled with multiple-effect distillation, which utilizes water vapor produced in each effect. A steam jet is used to extract the low-pressure steam from the vessels by creating a vacuum and then after mixing it with
high-pressure steam that is supplied to the system. The pressure of the resulting
steams mixture is then raised in the diffuser to the pressure required for the heating
steam in the first effect. Fig.1-5 shows a schematic diagram of thermal vapor
compression combined with a multi effect distillation process [8].

Reverse Osmosis (RO) is a membrane based desalination method that has been
rapidly growing in popularity. RO accounts for 44% of all desalination production
worldwide [4]. As opposed to thermal desalination technologies that utilize thermal
energy to distill water, RO is a pressure driven separation technique. It is based on
forcing saline water through a semi-permeable membrane to separate salts from the
water to produce potable water. The applied pressure that is required to drive the
process greatly depends on the salinity of the water. RO desalination is currently
favored over thermal desalination, as RO is currently the most cost-effective commercial
desalination technology. One drawback with RO is that the product water has higher levels of salinity than that produced with thermal desalination techniques—in some instances, salt concentrations exceed the standards for certain industrial applications and therefore further treatment is required.

Electrodialysis (ED) is another membrane-based separation mechanism that is usually used for brackish water desalination. The electrodialysis stake consists of several cells of anion and cation exchange membranes arranged in an alternating sequence. Fig.1-6 shows a schematic diagram of an ED cell [9]. The ED process consists of an anode and cathode that are separated by different membranes through which salt ions migrate to the electrodes.

![Figure 1-6. Schematic diagram of Electrodialysis cell [9]](image)

The movement of ions is controlled by ion-selective membranes between the anode and cathode. When DC voltage is applied between the two electrodes, positively...
charged cations (e.g. K$^+$ or Na$^+$) move toward the cathode, pass through the negatively charged membrane, and retained by the positively charged membrane. Similarly, the negatively charged anions (e.g. Cl$^-$) move toward the anode, pass through the positively charged membrane, and are retained by the negatively charged membrane. As a result of that, the concentration is reduced in the dilute compartment and increased in the concentrate compartment. ED is rarely used for seawater desalination due to the high salt concentration.

**Energy and Desalination**

The heading above shows that if you have a subheading of a certain level, you must have more than one. The rationale is that you cannot have a list of only one item.

Energy and water are essential for the growth and development of any country; and without them no life or civilization will exist. Energy production requires water resources, and water production requires energy resources. In the U.S., water is used to cool thermoelectric power plants; it accounts for 39% of total fresh water withdrawal [10]. Desalination is an energy intensive process that requires a large amount of either mechanical or electrical energy to separate water from salt. Most of energy currently used for desalination is fossil fuel based.

Seawater desalination has been increasing significantly in the last few decades, and due to the fact that energy and water are interdependent, the total energy consumption for seawater desalination has increased tremendously. Fig.1-7 shows the water desalination capacity in the U.S. and worldwide. With continued increases in freshwater demand and energy prices, desalinated water cost will increase as a result. Therefore, energy is considered to be the most important factor when a desalination technology is to be selected.
RO is currently the most widely used desalination technique due to its low energy consumption. The energy consumption for the RO process depends greatly on the salinity of water. RO is best suited for brackish and river water desalination than other thermal desalination technologies due to its low energy consumption. The drawback of all thermal desalination techniques is that they have high specific energy consumption. They can, however, readily handle higher salinity water than membrane desalination technology, and still deliver higher water quality than membrane desalting techniques. Fig.1-8 shows the market share of water desalination capacity worldwide using different desalination technologies.

The selection criterion of a desalination technology depends upon several factors, namely: operational cost, feed water salinity, fresh water quality required, energy cost, and local demand for electricity. Operational cost, which is dictated by the
energy consumption, is the most important factor when a desalting technique is to be considered. High-energy consumption translates to high water cost. Due to its low energy consumption, RO desalination technology has increased significantly in the last few decades. Table 1-2 illustrates key operational data for different desalination technologies. In terms of total energy consumption, it is evident that thermal desalination technologies consume more energy than membrane desalination technologies. On the other hand, energy consumption from thermal distillation is relatively independent of the salt concentration as opposed to membrane desalting techniques, where the energy consumption increases with increasing the salt concentration in the feed water. Therefore, RO and ED are more suitable for brackish water than seawater.

Figure 1-8. Market shares of different water desalination technologies worldwide [12]

In terms of ease of operation and reliability, thermal distillation technologies offer great advantage over membrane desalination processes. Thermal distillation is more
reliable and requires little maintenance to operate as opposed to membrane desalination. For example, RO is more susceptible to fouling than MSF or MED, and requires more maintenance, part replacement and treatment, such as continuous filter replacement and pH adjustment. Therefore, thermal distillation is the most widely used desalination technology in the Middle East region.

Table 1-2. Key operational data of different desalination technologies [13]

<table>
<thead>
<tr>
<th>Desalination Technology</th>
<th>RO</th>
<th>MSF</th>
<th>MED</th>
</tr>
</thead>
<tbody>
<tr>
<td>Electrical Energy Consumption (kWh/m³)</td>
<td>5.0-7.0</td>
<td>3.5-5</td>
<td>2-2.5</td>
</tr>
<tr>
<td>Thermal Energy Consumption (kWh/m³)</td>
<td>-</td>
<td>55-120</td>
<td>30-120</td>
</tr>
<tr>
<td>Feed water salinity, TDS (1000 ppm)</td>
<td>1-45</td>
<td>30-100</td>
<td>30-100</td>
</tr>
<tr>
<td>Fresh water salinity, TDS (ppm)</td>
<td>250-500</td>
<td>5.0-50</td>
<td>1.0-50</td>
</tr>
<tr>
<td>Replacement Part Requirement</td>
<td>Highest</td>
<td>Medium</td>
<td>Medium</td>
</tr>
<tr>
<td>Chemical Consumption</td>
<td>High</td>
<td>Low</td>
<td>Low</td>
</tr>
<tr>
<td>Ratio between product and seawater flow</td>
<td>0.3-0.5</td>
<td>0.09-0.2</td>
<td>0.3-0.6</td>
</tr>
</tbody>
</table>

The selection criteria also depend on the feed water quality, which is usually characterized by water salinity. The salinity of the feed water is represented by the total dissolved solids content (TDS) in the water. Table 1-3 shows feed water salinity for different water resources.

Table 1-3. Feed water salinities [14]

<table>
<thead>
<tr>
<th>Feed Water</th>
<th>TDS (ppm)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Seawater</td>
<td>15.000-50,000</td>
</tr>
<tr>
<td>Brackish water</td>
<td>1.500-15,000</td>
</tr>
<tr>
<td>Drinking water</td>
<td>0-500</td>
</tr>
</tbody>
</table>

Fresh water quality is an important factor for the selection of any desalination technology. The maximum salinity limit specified by the World Health Organization (WHO) for drinkable water is 500 parts per million (ppm). Drinking small amounts of seawater is not harmful; however drinking large amounts of seawater increases the salt
concentration in the human body and thus causes dehydration. Further drinking of seawater results in increasing the salt concentration to a toxic level, where it interferes with the nervous system and may cause seizures or other neurological issues. Depending on the water quality requirement, the proper desalination technology can be selected. Some chemical processes, such as ion exchange, require water with very low TDS. Thermal distillation can produce high quality water with a TDS as low as 5 ppm.

**Seawater Properties and Desalination Challenges**

Even though seawater is nearly inexhaustible and considered to be an excellent choice to be used for desalination, it has a high salinity concentration. Salt in seawater constitutes to approximately 3.3% of seawater mass. Seawater properties vary depending on many parameters such as water depth, water temperature, ocean currents, and neighboring industrial discharge. Some factors such as dissolved atmospheric gases, concentration of heavy metals, chemical discharge from local industries, and biological growth are deterministic factors that not only influence the operation of desalination plants, but also deteriorate their performance. These factors are a major challenge for desalination processes, especially for membrane desalination.

It is of interest to understand the affects seawater salt and chemical constituents on desalination equipments. Salt in seawater is composed of different constituents at different proportions as summarized in Table 1-4. It is clear that chlorine (Cl\(^-\)), sodium (Na\(^+\)), sulfate (SO\(_4^{2-}\)), magnesium (Mg\(^{2+}\)), calcium (Ca\(^{2+}\)), and potassium (K\(^+\)) are contributing to more than 99% of the salt content of seawater. The percentage of each constituent is typically uniform regardless of the seawater salinity. Seawater also contains heavy metal ions such as Cu, Ni, Zn, and Pb that can cause corrosion in metallic components of a desalination system.
Table 1-4. Chemical constituents in seawater salt [15]

<table>
<thead>
<tr>
<th>Chemical</th>
<th>Proportion</th>
</tr>
</thead>
<tbody>
<tr>
<td>chlorine (Cl⁻)</td>
<td>55.1%</td>
</tr>
<tr>
<td>sodium (Na⁺)</td>
<td>30.4%</td>
</tr>
<tr>
<td>sulfate (SO₄²⁻)</td>
<td>7.7%</td>
</tr>
<tr>
<td>magnesium (Mg²⁺)</td>
<td>3.8%</td>
</tr>
<tr>
<td>calcium (Ca²⁺)</td>
<td>1.2%</td>
</tr>
<tr>
<td>potassium (K⁺)</td>
<td>1.1%</td>
</tr>
<tr>
<td>other chemicals</td>
<td>0.42%</td>
</tr>
</tbody>
</table>

Due to air-water exchange, seawater has a large amount of dissolved atmospheric gases (mainly Nitrogen and Oxygen). The solubility of the oxygen depends on the seawater temperature. Generally, oxygen is important for marine life due to its role in biological processes. However, dissolved oxygen is one of the factors that lead to metallic corrosion when it is in contact with seawater.

One of the most important challenges of seawater and brackish water desalination are scaling and fouling. Scaling is referred to salt precipitation on desalination equipment surfaces due to the solubility of salt being exceeded. Fouling is referred to deposition of colloidal and particulate matter, and the biological growth of microorganisms on desalination system surfaces. Both, scaling and fouling, are strongly dependant on the feed water composition [16].

Scaling occurs in thermal and membrane desalination plants. Seawater has high concentrations of scaling formers; and when the concentration exceeds the saturation limit, precipitation occurs. Scaling formers in seawater can be divided into alkaline scale formers, such as CaCO₃, and Mg(OH)₂, and non alkaline formers, such as CaSO₄. In thermal distillation, scaling occurs due to the breakdown of the bicarbonate species as given in the following equations:

\[2HCO₃⁻ \rightarrow CO₃^{2⁻} + 2H₂O + CO₂,\]  \hspace{1cm} (1-1)
\[ Ca^{2+} + CO_3^{2-} \rightarrow CaCO_3, \quad (1-2) \]
\[ CO_3^{2-} + 2H_2O \rightarrow CO_2 + 2OH^-, \quad (1-3) \]
\[ Mg^{2+} + 2OH^- \rightarrow Mg(OH)_2. \quad (1-4) \]

The formation of magnesium hydroxide occurs at high temperature, above 95°C; however, the formation of calcium carbonate can occur even at low temperature [17]. The non-alkaline scaling is less problematic than that of alkaline because calcium sulfate is more soluble than calcium carbonate. The rate of scaling depends upon the seawater temperature, and the concentration of the bicarbonates and calcium carbonate. The formation of CaSO$_4$ can be controlled in MSF by maintaining the highest brine temperature below 122°C.

Fouling is more problematic in RO systems than in MSF systems. In RO systems, fouling occurs on the membrane surface due to the deposition of particulate matter, biological foulants, or organic foulants. Particulate fouling, such as aluminum oxide, iron oxide, and silica, can cause membrane fouling. Biological growth can form a biofilm layer on the membrane surface which inhibits permeate flux across the membrane, resulting in increasing the pressure loss along the feed channel and deteriorating the membrane performance. Humic acid is an organic foulant that is produced by the degradation of the organic matter. Organic foulants form a gel-like layer on the membrane surface that results in reducing in the membrane efficiency.

Seawater properties are not uniform throughout the world. They can vary due to industrial chemical discharge, fresh water runoff from rivers or melting glaciers. Therefore, feed water characterization is important to predict the tendency of scaling or
fouling. The parameters that are commonly used to characterize seawater are TDS, water conductivity, silt density index, and the solubility. They are defined as:

1. The TDS is used to characterize the content of the total dissolved solid in the feed water.
2. Water conductivity is directly proportional to the TDS, and depends strongly on the temperature of the feed water.
3. The silt density index (SDI) is used to predict fouling potential of the feed water.
4. Feed water hardness is used to describe magnesium and calcium content in the feed water.
5. The solubility is used to predict scaling potential of the feed water. The solubility depends on the mole fraction of the salt in water at saturation conditions.

There are certain methods that are currently used to prevent or reduce scaling and fouling in desalination plants. Pre-treatment is commonly used in all desalination technologies to prevent fouling and scaling as well as to provide a high quality feed water without fluctuations. Pre-treatment can be categorized into mechanical or chemical pre-treatment. Mechanical pre-treatment includes screening, sand filtration, cartridge filtration, and membrane filtration. The conventional approach of mechanical pre-treatment includes flocculation, and settling. With the advancement of membrane performance and reduction in their costs, membrane pre-treatment is more widely used than physical pre-treatment [18-20]. Microfiltration and ultrafiltration (UF) are considered as an alternative pre-treatment method and expected to continue to grow in the future. The chemical pre-treatment depends on the type of the physical pre-treatment used. It involves chemical inhibitors addition such as coagulants, disinfectants, and polyelectrolyte. Anti-scaling agents are used to prevent scaling in desalination plants such as the use of sulphuric acid to prevent calcium carbonate scaling [18].
Seawater contains microorganisms such as bacteria, viruses, fungus, algae and higher organisms such as protozoa or biotic debris. To remove bacterial contamination in seawater, many methods are currently used, such as chlorination, microfiltration, ozone, iodine, and ultraviolet germicidal irradiation. Chlorination is the most effective method used to disinfect feed water and to prevent any biological growth. Chlorine is added to the feed water as an ionic solid, such as sodium hypochlorite (NaClO) and Calcium hypochlorite Ca(OCl)₂, or in gaseous form of Cl₂. As chlorine is added to the feed water, it reacts with the organic matter and pathogens and destroys them. The disinfection efficiency depends upon the contact time and dosage of the chlorine. Chlorination is widely used in seawater desalination, sewage treatment, and industrial water treatment. In RO systems chlorination is essential in the pre-treatment stage to prevent bio-fouling of the membranes. Chlorination is the cheapest method used for water disinfections. Ultraviolet (UV) is another method used to disinfect the feed water. However, chlorination is more widely used than UV.

Ozonation is another method that has been developed in recent years to disinfect feed water. Ozone, like any oxidant, has lethal affects on organic matter; it distroys algae and microorganisms. An advantage of using ozonation as a water disinfection method is that it is a very powerful oxidant that requires a short period of time to kill germs and pathogens. Ozonation is a very effective method in removing taste, color and odor problems of treated water. On the other hand, the use of ozonation has some disadvantages, such as high cost compared to chlorination, and difficulties in system installation and operation.
The pH value of seawater is usually reduced to inhibit CaCO₃ scaling. This pH adjustment is necessary in RO systems in order to improve the membrane performance. This can be achieved via addition of acids, such as sulfuric acid (H₂SO₄). Furthermore, in MSF desalination plants the re-circulating seawater has a high concentration of salt and heavy metals, such as Cu and Ni, that can deposit on the condenser tubes and the flashing chamber, which increases corrosion. Flocculation is used to remove any turbidity or color from the treated water. It is commonly used in seawater desalination, sewage treatment, and industrial wastewater treatment. This process can be done via addition of flocculation agents, such as aluminium chlorohydrate, aluminium sulfate, calcium oxide and polyacrylamide. Coagulation is also used to clarify feed water turbidity. Coagulation agents such as ferric chloride (FeCl₃), ferrous sulfate Fe₂(SO₄), or aluminum sulphate (Al₂(SO₄)₃) is added to the water. The addition of coagulation and flocculation agents to the feed water causes dissolved matter to colloid and agglomerate.

Feed water quality varies from one location to another and influences the desalination process. Therefore, pre-treatment is essential for any desalting techniques to prevent scaling and fouling. It should be mentioned that pre-treatment of the feed water is a costly process, and it increases the operational cost of the desalination system. Micro filtration and ultrafiltration has been widely used in pre-treatment of industrial and municipal wastewater. The use of micro filtration and ultrafiltration requires less chemical addition than conventional pre-treatment [21]. Furthermore, conventional pre-treatment requires a larger footprint than micro filtration and
ultrafiltration. However, conventional pre-treatment is still the dominant pre-treatment method used in water desalination and water treatment.

Desalinated water is characterized by low hardness and alkalinity which can be corrosive for pipeline materials and may result in reducing the lifetime of the pipeline infrastructure. The low TDS of desalinated water makes the water unpalatable. Therefore, post-treatment is typically required for the desalinated water so that it conforms to appropriate standards. Post-treatment includes re-hardening the desalinated water, adjusting the pH value and adjusting the CO₂ content. Post-treatment is important to prevent scaling and corrosion. Water corrosiveness is characterized by the Langelier Saturation Index (LSI). Negative LSI indicates the water is corrosive. Many methods are used to make desalinated water less corrosive by adding some chemicals. The addition of chemicals includes dissolution of lime by carbon dioxide, dissolution of limestone by carbon dioxide, and the addition of bicarbonate or calcium chloride. The most commonly used method for re-hardening is the dissolution of limestone by carbon dioxide. This adds alkalinity to the desalinated water to make it less corrosive. Adding these chemical solutions is usually used for small-scale applications due to the high-cost.

**Environmental Impact of Desalination**

The increase in fresh water demand necessitates utilizing alternative water resources such as seawater desalination. However, desalination has unfavorable impacts on the environment. The discharge of concentrated brine, discharge of waste membranes, interference with marine systems and noise and gas emissions are among the top disadvantages of water desalination. The waste of water desalination is an
important challenge that is becoming increasingly important, and attention should be paid to the current practice of desalination waste management.

High energy consumption is considered the most important factor that inhibits growth of seawater desalination. Currently, most desalination plants use power obtained from fossil fuel sources. Thus, the increasing use of seawater desalination will increase gas emissions. To alleviate the environmental impact of water desalination and at the same time provide a sustainable source of drinkable water, renewable energy utilization should be considered. This also mitigates the dependence on fossil fuel.

Discharge of the concentrated brine is the main unpleasant waste from saline water desalination plants. Marine life is strongly affected by the discharge of the concentrated brine. The brine is not only salt concentrated, but also contains chemicals from pre- and post-treatment such as anti-scaling agents. This results in high concentration of salt in the vicinity of the brine discharge point. Brine disposal is a problem that challenges all water desalination technologies. In RO plants, the brine is more concentrated than the brine discharged from thermal distillation plants. However, in thermal distillation the brine is discharged at a relatively high temperature compared with membrane distillation, which is more problematic for the marine life. Furthermore, marine life is also affected by the intake of the seawater for the desalination plant. When a large amount of water is extracted from the sea, marine organisms and algae are sucked into the intake.

The discharge of concentrated brine could possibly damage the marine life near the discharge point. Many methods are currently used for brine disposal. Brine can be
discharged to the sea, land based solar ponds, or injected to deep saline aquifers. The discharge of brine to the sea is the least expensive method. When brine is discharged to the sea, it tends to sink at the bottom of the sea because it has higher density than the seawater. Water can be diluted with seawater to reduce its salinity and then discharged to the sea. Furthermore, operating at lower recovery rate reduces the salinity of the brine. Brine is discharged at high depth of seawater and strong current reduce the detrimental affects of brine on the marine life.

Brine discharge to a solar pond or injection to a deep saline aquifer is used for desalination plants that are located away from the sea. Brine can be used for irrigation salt-tolerant plants [22-24]. However, it has disadvantages because it increases the salt in the soil and also increases the salinity of the ground water. The use of a solar pond for brine disposal is space consuming and carries the risk of contaminating the ground water.

**Renewable Energy Utilization for Seawater Desalination**

Meeting water and energy demands is essential for the national security of any country. As the world population grows, energy and water demand will continue to increase, and this might lead to conflict among countries on these resources. The increase of non-renewable energy utilization for seawater desalination will have an unfavorable impact on climate change and pollution; it leads to an increase in greenhouse gas emissions. Nonrenewable energy such as fossil fuels are becoming increasingly depleted, and extensive technological development for harnessing renewable energy is the only solution for energy and water shortage problems for the future. Renewable energy driven desalination plants account for less than 1% of the total desalination plants [25].
There is an increasing interest in utilizing renewable energy resources to drive seawater desalination. Renewable energy sources, such as solar, wind, biomass, geothermal and tidal, offer the promise as reliable supplies of energy. Utilization of renewable energy is becoming increasingly imperative, especially to reduce fossil fuel dependence, and to achieve energy security.

The best combination of renewable energy and desalination technology must be achieved to ensure the maximum fresh water production as well as the most energy efficient utilization. An excellent example is the integration of thermal desalination technologies with renewable thermal energy, such as solar and geothermal energy. Solar energy is the most promising energy for seawater desalination. Solar energy utilization systems such as flat plat solar collectors, evacuated tubes, and solar ponds absorb the solar energy and convert it to thermal energy that can drive thermal desalination processes. The use of solar energy for desalination can be categorized into direct, where the solar energy is absorbed directly by the saline water (solar pond and solar still), or indirect where the solar energy is absorbed by a solar collector and then transferred to the saline water. Solar energy is well suited for arid regions and rural areas where the solar intensity is high. Fig.1-9 shows the solar intensity distribution in the U.S. An advantage of solar energy utilization is that, during the summer season, especially when the water availability is low compared to other seasons, the solar irradiation intensity is very high which results in desalinating more water.

Alternative energy is abundant in nature, but some difficulties such as harnessing, converting, and storing energy put constraints on the utilization of
alternative energy which makes it difficult to compete with non-renewable energy. Many factors influence the utilization of renewable energy, namely geographical location, season, time, intensity, and unpredictability. Another drawback of renewable energy utilization is its expensive compared to non-renewable energy. Further, discrepancies exists between the supply and demand for renewable energy, as opposed to nonrenewable energy where fuel can be easily stored or burned to produce electrical power or heat and at any time.

![Figure 1-9. Solar intensity distribution around in the U.S. [26]](image)

Alternative energy intensity is highly variable; due to the inherent intermittency, energy should be stored to be used whenever it is needed. The thermal energy harnessed from the sun or geothermal supplies can be stored in phase change materials, storage tanks, or even in a solar pond. The mechanical energy harnessed
from wind turbines is converted to electrical energy that can be stored in batteries, hydrogen fuel cells, capacitors or hydroelectric systems.

For all of the above reasons, renewable energy is more expensive than nonrenewable energy. Thus renewable energy utilization requires extensive research and development to improve the harnessing and storage capabilities of energy in order to compete with non-renewable energy. Innovations in renewable energy utilization are necessary to reduce the dependence on fossil fuel as well as to provide cost-effective energy sources for seawater desalination. Continuous development and investment in the infrastructure of renewable energy will ultimately bring the cost down.

**Humidification/ Dehumidification Process**

The humidification/dehumidification (HDH) process is a thermal desalination process that mimics the natural water purification cycle. It is based on the principles of humidification and dehumidification of an air stream. This process is best suited to utilize solar energy to heat saline water to a sufficiently high temperature such that water is evaporated. The temperature of the saline water is usually raised to 60°C; this is relatively a low water temperature, which gives the HDH process an advantage over other traditional thermal desalination processes such as MSF or VC. The downside of the HDH process is that the conversion rate is low; thus it is best suited for small-scale applications.

The use of the humidification and dehumidification method for saline water desalination has been introduced and analyzed by many researchers in recent years. Farid et al. [27] presented a design for a HDH desalination unit utilizing solar energy. The water temperature was measured and used to calculate the performance factor for
different air and water flow rates. The variation in the daily performance factor ranged from 0.95-1.35. They concluded that decreasing the water mass flow rate to 70 kg/h increases the performance factor due to the increase in the water temperature.

Nawayseh et al. [28] developed a simulation program to find the optimum operating condition for the closed cycle HDH process. They concluded that the performance of the HDH process is very sensitive to the water flow rate and to the humidifier and condenser surface areas. It was recommended to utilize natural air draft circulation. Furthermore, it was found that the air flow rate has an insignificant effect on the fresh water production of the unit, and it was concluded that natural air draft circulation is more favorable than forced circulation. For a water and air mass flow rate of 0.48 kg/min and 0.24 kg/min respectively, the maximum fresh water production for the system is not more than 0.7 kg/(m².hr) and the average daily production is 3.5 kg/(m².day) for 10 hour operation.

Al-Hallaj et al. [29] presented similar work to that of Nawayseh et al. [28] and reported that the air flow rate has little effect on the fresh water production performance when operating at high water temperature (70°C). On the other hand, when operating at low temperature (50°C), increasing the air flow rate improves the production performance. In their experiment the maximum fresh water production was 0.65 kg/hr, and the daily production was 5 L/(m².day). In the study, it was evident that the fresh water production significantly increases with increasing water mass flow rate to an optimum point. Beyond the optimum, the water production decreases with increasing the mass flow rate. They also suggested it is best to operate the unit with natural air
draft circulation at high temperatures; while at low temperature forced air circulation will improve the fresh water production.

Muller [30] developed a simulation program to optimize a small-scale thermal seawater desalination system. They used a thermal storage tank to enable the system to run 24 h/d of operation. They also used highly efficient solar collectors to heat the water to 85°C. Dai et al. [31] used forced convection to circulate the air within an open loop humidification and dehumidification system. The solar collector was replaced by a boiler to heat the water. The thermal efficiency for the desalination process was defined as the ratio of the minimum energy to obtain a fixed amount of water to the total heat input ($\eta_{\text{thermal}} = \frac{m_{\text{in}}h_{\text{fg}}}{Q_{\text{in}}}$). For their system, a thermal efficiency of 80% was achieved.

Furthermore, in the study it was reported that the performance is strongly dependent on the inlet saline water temperature, mass flow rate of the saline water, and the mass flow rate of air.

Orfi et al. [32] added a solar air heater to improve the performance of the system. They also replaced the solar water heater with an electric heater. For a typical day in the summer, the measured water and air temperature at different locations in the system were presented. The experimental results show that increasing the ambient air temperature improves the performance of the system. In the study, a fixed solar radiation intensity of 800 W/m² was used in the simulation, and a maximum fresh water production of 5.2 kg/m²d was observed. It was also reported that increasing the cooling water flow rate increases the production to an optimum value, and beyond that optimum the production decreases.
A comprehensive technical review given by Parekh et al. [33] presents a comparison between different thermal desalination techniques. In the study, the performance of the HDH reported by many researchers was laid out. Further simulations with a thermal storage module and 24 h operation were recommended to improve the process of the HDH. Numerous simulations are required to understand the air and mass flow rate influence on the performance of the HDH desalination process. The main disadvantage of the HDH process is that it relies on natural draft to drive the airflow in the system; therefore, heat and mass transfer coefficients are low. Furthermore, film condensation over tubes is usually used to condense the water vapor in the air stream. Due to the low efficiency of the condenser, large surface area is required which increases the cost of the system. Klausner et al. [34] have developed a thermal distillation process called diffusion driven desalination. The diffusion driven desalination process is considered promising for water desalination when driven by waste heat. Thermodynamic analysis was performed to study the performance of the DDD process. A theoretical model has been developed to describe the heat and mass transfer in the process and the performance has been studied for different operating conditions.

Solar Diffusion Driven Desalination Process

Solar DDD is a thermal distillation process similar to the HDH where a solar collector is used to deliver the thermal energy. A simplified schematic diagram of the Solar DDD process is shown in Figure 1-10. The Solar DDD facility is comprised of two main components: (1) evaporator and (2) condenser. The evaporator and the condenser are mainly composed of a packing material that allows direct contact between air and water. The packing material is characterized by a high surface area to
volume ratio to improve the water and air contact. The evaporator and the condenser entail a nozzle mounted at the top of the packing and used to spray water over the packing material. A basin at the bottom of the diffusion tower is installed and used to collect the saline water and another one is installed at the bottom of the condenser to collect the freshwater.

Figure 1-10. Schematic diagram for the solar diffusion driven desalination process

The DDD process operates on the principle of humidification and dehumidification of an air stream. The first process is called air humidification and occurs in the evaporator. The evaporator admits warm saline water, as low as 45°C, and sprays it at the top of the packing by means of a nozzle. Cold and low humidity air is forced via a fan at the bottom of the diffusion tower to circulate the air through the
facility. The drawn air is flowing counter-currently to the falling water and therefore both will be in direct communication. Heat and mass transfer will be taking place between the water and air leading to humidification of the air. The humidified air discharges the evaporator at the top as fully saturated and then enters the direct contact condenser. In the direct contact condenser, cold fresh water is sprayed and the humidified air is brought into direct communication with cold fresh water. This process is referred to as dehumidification, and therein heat will be transferred from vapor to the cold freshwater resulting in vapor condensation and purified water production. The fresh water is collected at the basin of the condenser and sent to a fresh water production tank or to a heat exchanger to be cooled and recycled again.

For this study, the DDD facility will be ideally equipped with a flat plat solar collector to harness the solar thermal energy and transfer it to the saline water. The saline water discharges the solar collector after being heated, and it is then re-circulated through the evaporator. The main advantage of utilizing solar collectors is to operate the Solar DDD autonomously without the need for other energy sources. Due to its low electrical energy consumption, and suitability for low water temperatures, Solar DDD may be a competitive method for small scale seawater desalination.

Research Objectives and Scope of Work

The main objective of this investigation is to develop a theoretical model for the Solar DDD process that can capture the transient variation of the saline water temperature. While a theoretical model for the DDD process has been extensively reported [35-37]. It is only valid for a steady state operation, as shown in Appendix B. In order to rigorously model the solar DDD process, which operates in a re-circulating mode, a transient model is required. This model will be derived from the basic
principles of conservation of mass and energy. The transient variation of the water temperature, air temperature and packed bed temperature will be investigated. The transient model can be used to predict the fresh water production, and examine the parameters that influence the performance of the solar DDD.

A solar collector is used to harness solar energy, which is installed after the saline water tank as shown in Fig.1-10. Due to the day-night and sunny-cloudy cycle of the solar radiation, an effective method should be implemented to account for this cyclic variation and its influence on the water temperature within the solar collector. This is very crucial to accurately understand its affect on the water temperature, which drives the performance of the solar DDD process. It is expected that any variation in the solar intensity will have an influence on the energy collected by the solar collector; thus the fresh water production will change accordingly as the evaporation process is greatly dependent on the saline water temperature entering the diffusion tower.

The theoretical modeling enables understanding of the underlying physics relating to heat and mass transfer between the working fluids and the packed bed as well as the evaporation and condensation process. Experimental work is required to validate the theoretical model and to investigate the validity of the numerical results. A key outcome of this research is to fully understand the thermal response of the Solar DDD due to the diurnal variation of the solar intensity and observe its affect on the fresh water production. Thus, the objectives of this research are summarized as:

1. Design a cost effective solar diffusion driven desalination facility that is well suited for arid and remote regions.
2. Develop a theoretical model to describe heat and mass transfer in the direct contact evaporator and condenser.
3. Use theoretical modeling to investigate the performance of the solar DDD process.
4. Optimize the system performance to improve fresh water production and reduce the energy consumption.
5. Perform economic analysis on the solar DDD.

Chapter 2 presents a theoretical model for the heat and mass transfer in the direct contact evaporator and condenser. It also presents the computational method used for solving the developed models. Chapter 3 presents an experimental setup and procedure that is used to validate the theoretical model of the evaporator and condenser. Comparison between experimental data and numerical simulation are made in Chapter 3. Chapter 4 discusses the dynamic response of the developed model for a given operating conditions. In Chapter 5, the performance of the solar DDD is analyzed and the parameters that influence the water production and energy consumption are investigated. Economic analysis is carried out in Chapter 6. Chapter 7 concludes the current work.
CHAPTER 2
HEAT AND MASS TRANSFER WITHIN THE DIRECT CONTACT EVAPORATOR AND CONDENSER

Theoretical Heat and Mass Transfer Formulation

Direct contact evaporation and condensation for countercurrent flow within packed beds is associated with high rates of heat and mass transfer and have direct application to various important industrial processes including air striping, evaporative cooling, and distillation among others. A typical configuration for direct contact evaporation and condensation involves liquid distributed over the top of a packed bed that gravitates downward through the bed. As it passes through the bed it wets the surface of a packing that provides a very high surface area per unit volume. A non-condensable gas is blown countercurrently upward through the bed. Extensive heat and mass transfer occurs through the gas/liquid interface. Klausner et al. [35] and Li et al. [36] have provided a rigorous mathematical framework for analyzing heat and mass transfer for the steady state operation of direct contact packed bed evaporators and condensers. There also exist applications where such systems operate cyclically and are characterized by transient heat and mass transfer.

The Diffusion Driven Desalination (DDD) process described by Klausner et al. [34] utilizes both direct contact evaporation and condensation for the distillation of the sea and brackish water. When the DDD process is driven by solar heating, the operation is inherently transient, and a transient heat and mass transfer analysis is required to evaluate its operation over a range of operating conditions. The evaporation and condensation process within the evaporator and condenser primarily depends on the inlet water and air temperature, inlet humidity ratio, and water to air flow ratio.
The thermal performance of cooling towers has been the subject of investigation for more than a century. Most prior work has focused on steady-state operating conditions. In order to study the thermal response of a cooling tower to a varying heat input and operating conditions, a transient model is required. Transient thermal analyses of cooling towers are important because the ambient air temperature can vary significantly between day and night. Recently the dynamic behavior of cooling towers was investigated by Al-Nimr [38, 39]. The thermal capacity of the packed bed is included in the analysis. The transient energy equations account for variations in water, air, and packed bed temperature. The overall heat transfer coefficient between water and air is assumed constant. A closed form solution is obtained for the cooling tower steady state performance. A transient solution is obtained using a perturbation method. The transient solution has not been experimentally verified. Marques et al. [40] studied the transient thermal behavior of counter flow cooling towers. Transient equations of air and water in the cooling tower are derived from energy and mass balances. Also, the transient variation of water temperature in the basin of the cooling tower is accounted for. The heat transfer between the packed bed and the air side is neglected. A 10% step change in the initial conditions of the air flow, water flow, and heat load is used to study the thermal response of the cooling tower. A comparison with experiments is lacking. While these analyses demonstrate some interesting qualitative features on the dynamic response of cooling towers, there is some uncertainty associated with the heat transfer coefficients used to close the one-dimensional analysis.

**Evaporator**

The evaporation process depends primarily on the inlet water and air temperature, inlet air humidity, and water to air mass flow ratio. The conservation of
energy is applied on a differential volume of the packed bed where heat will transfer between the water, air and packed bed. A schematic diagram of the evaporator is shown in Fig. 2-1. The packed bed is not completely wetted, and wetted and unwetted surfaces are distinguished. The ratio of the wetted area to total area depends on the wettability of the fluid/surface combination and the dynamics of the flow through the bed.

Figure 2-1. Countercurrent flow through a packed bed evaporator

In order to derive a system of equations for the transient operation of the evaporator, the following assumptions are made. There are no thermal losses from the
evaporator to the environment, and both air and water vapor may be treated as perfect gases. Fig. 2-2 shows a differential volume of the packed bed where water trickles downward and spreads over the bed to form a film, and air flows upward. The positive z-direction is taken as upward. Heat is flowing from the liquid to the air at the liquid/vapor interface, and from the liquid to the packed-bed at the liquid/solid interface. Heat and mass are flowing at the liquid/vapor interface due to temperature and concentration gradients within the liquid and the air/vapor mixture; thus latent and sensible heat is convected from the liquid film. At the liquid/solid interface, the heat flow is due to convection.

![Evaporator differential control volume depicting liquid/vapor/solid interactions.](image)

Figure 2-2. Evaporator differential control volume depicting liquid/vapor/solid interactions.

The conservation of mass applied to the liquid phase of the control volume gives,

$$\left(\dot{m}_L \right)_{z+dz} - \left(\dot{m}_L \right)_z - d\dot{m}_{v,\text{evap}} = \frac{\partial}{\partial t} \left( \rho_L \alpha_L A_c dz \right), \quad (2-1)$$
where \( \dot{m}_L \) is the mass flow rate of the liquid, and \( d\dot{m}_{V,\text{evap}} \) is the portion of liquid evaporated; \( m_{L,\text{stored}} = \rho_L \alpha_L A_c dz \), where \( \rho_L \) is the liquid density, \( \alpha_L \) is the liquid volume fraction, and \( A_c \) is the cross sectional area. Noting that \( \dot{m}_{L, z+dz} - \dot{m}_{L, z} = \frac{\partial}{\partial z} \left( \dot{m}_L \right) dz \), Eq. (2-1) is simplified to,

\[
\frac{\partial}{\partial z} (\dot{m}_L) dz - d\dot{m}_{V,\text{evap}} = \rho_L A_c \frac{\partial}{\partial t} (\alpha_L) . \tag{2-2}
\]

The liquid volume fraction (holdup) depends on the packing configuration, the gas and liquid mass flow rate and liquid physical properties. For a constant gas and liquid flow rate, the liquid volume fraction is nearly constant, and Eq. (2-2) is further simplified to,

\[
\frac{d}{dz} (\dot{m}_{L, z}) = \frac{d}{dz} (\dot{m}_{V,\text{evap}}) . \tag{2-3}
\]

Likewise, the conservation of mass applied to the gas (air/vapor mixture) side is expressed as,

\[
(\dot{m}_a + \dot{m}_v)_{z} - (\dot{m}_a + \dot{m}_v)_{z+dz} + d\dot{m}_{V,\text{evap}} = \frac{\partial}{\partial t} (m_{\text{mix}}) , \tag{2-4}
\]

where \( \dot{m}_a \) and \( \dot{m}_v \) are the mass flow rates of the air and vapor, respectively, and \( m_{\text{mix}} \) is the mass of the air/vapor mixture within the differential volume which can be written as \( m_{\text{mix}} = \rho_{\text{mix}} \alpha_{\text{mix}} A_c dz \). The volume fraction of the mixture (\( \alpha_{\text{mix}} \)) can be related to the volume fraction of the packed bed (\( \alpha_{\text{pack}} \)) and the liquid volume fraction (\( \alpha_L \)) as

\[
\alpha_{\text{mix}} = 1 - \alpha_L - \alpha_{\text{pack}} . \]

The volume fraction of the liquid is nearly constant for constant water and air flow rate, and thus the rate of change in the volume fraction of the
air/vapor mixture is small and neglected. Noting that

\[
(m_a + m_v)_{z+dz} = (m_a + m_v) + \frac{\partial}{\partial z}(m_a + m_v)_{z} \, dz
\]

and \( \dot{m}_a \) is constant, Eq. (2-4) is simplified to,

\[
-\frac{\partial}{\partial z}(m_v) \, dz + d\dot{m}_{v,\text{evap}} = A_c \, dz (1 - \alpha_L - \alpha_{\text{pack}}) \frac{\partial}{\partial t} (\rho_{\text{mix}})
\]

(2-5)

In the above the mixture density is

\[
\rho_{\text{mix}} = \frac{m_a}{m_a + m_v} \rho_a + \frac{m_v}{m_a + m_v} \rho_v,
\]

(2-6)

where \( \rho_a \) and \( \rho_v \) denote the air and vapor density, respectively. With the assumption of constant air and vapor densities, the time derivative of the mixture density vanishes so that Eq. (2-5) is simplified to

\[
\frac{\partial}{\partial z} (m_v) = \frac{\partial}{\partial z} (\dot{m}_{v,\text{evap}}).
\]

(2-7)

For an air/water vapor mixture, the humidity ratio is related to the relative humidity, \( \Phi \), through,

\[
\omega = \frac{\dot{m}_v}{\dot{m}_a} = \frac{0.622 \Phi P_{\text{sat}}(T_a)}{P - \Phi P_{\text{sat}}(T_a)},
\]

(2-8)

where \( P_{\text{sat}}(T_a) \) is the water saturation pressure corresponding to the air temperature \( T_a \).

The total system pressure is assumed constant when evaluating the humidity ratio.

Using the definition of the mass transfer coefficient and applying it to the differential control volume, the right hand side (RHS) of Eq. (2-7) is

\[
\frac{\partial}{\partial z} (\dot{m}_{v,\text{evap}}) = k_{G} \alpha_w \left[ \rho_{v,\text{sat}}(T_e) - \rho_{v,\text{sat}}(T_a) \right] A_c,
\]

(2-9)

where \( k_{G} \) is the mass transfer coefficient on gas side, \( \alpha_w \) is the specific wetted area of the packed bed defined as the wetted surface area of the packed bed per unit volume of
space occupied. Applying the perfect gas law to the vapor, the gradient of the evaporation rate is,

\[ \frac{\partial}{\partial z} (\dot{m}_{v,\text{evap}}) = k_G a_w \frac{M_x}{R} \left( \frac{P_{\text{sat}}(T_i)}{T_i} - \frac{\Phi P_{\text{sat}}(T_a)}{T_a} \right) A_i, \] (2-10)

where \( T_i \) is the liquid/vapor interfacial temperature. Combining Eqs. (2-7), (2-8) & (2-10) the gradient of the humidity ratio in the evaporator is expressed as,

\[ \frac{\partial \omega}{\partial z} = \frac{k_G a_w}{G} \frac{M_x}{R} \left( \frac{\omega}{P_{\text{sat}}(T_i)} - \frac{\omega}{0.622 + \omega} \frac{P}{T_a} \right), \] (2-11)

where \( G \) is the air mass flux. This is the same relation used for steady state operation [35]. There is no storage at the liquid/water vapor interface, and thus

\[ U_L (T_L - T_i) = U_G (T_i - T_a), \] (2-12)

where \( U_L \) and \( U_G \) are the respective liquid and gas heat transfer coefficients. Eq. (2-12) is used to evaluate the interfacial temperature,

\[ T_i = \frac{T_L + \left( \frac{U_G}{U_L} \right) T_a}{1 + \left( \frac{U_G}{U_L} \right)}. \] (2-13)

Application of the conservation of energy to the water side of the control volume depicted in Fig. 2-2 yields,

\[ (\dot{m}_L h_L)_{z+dz} - (\dot{m}_L h_L)_z - d(\dot{m}_{v,\text{evap}})h_{fg} - U_L a_w (T_L - T_{\text{pack}}) A_i dz \]

\[ -Ua_w (T_L - T_a) A_i dz = \frac{\partial}{\partial t} \left( \rho_L \alpha_L C_p L T_L A_i dz \right), \] (2-14)

where \( h_L \) is the enthalpy of water, \( h_{fg} \) is the latent heat, and \( T_{\text{pack}} \) is the temperature of the packed bed. Eq. (2-14) can be further simplified to,
\[
\frac{\partial}{\partial z}(m_\ell h_{\ell})_z - \frac{\partial}{\partial z}(m_{\text{r, evap}})h_{fg} - U_L a_w (T_L - T_{\text{pack}})A_c - Ua_w (T_L - T_a)A_c = \rho_L A_c \alpha_L C_p L \frac{\partial}{\partial t} (T_L). \tag{2-15}
\]

The first term reflects the change in the liquid enthalpy along the z-direction, the second term accounts for the heat transport through evaporation, the third term accounts for the convective heat transport to the packing, the fourth term accounts for the convective heat transport to the gas/vapor mixture, and the last term accounts for the transient variation of the liquid heat capacity.

Noting that \(\frac{\partial}{\partial z}(m_\ell h_{\ell})_z = m_\ell \frac{\partial}{\partial z}(h_{\ell})_z + h_{\ell} \frac{\partial}{\partial z}(m_\ell)_z\), and substituting Eq. (2-3) for the second term, Eq. (2-15) is written as,

\[
\frac{\partial}{\partial t} \left( \frac{m_\ell (h_L) - \frac{\partial}{\partial z}(m_{\text{r, evap}})(h_{fg} - h_{\ell}) - U_L a_w (T_L - T_{\text{pack}})A_c - Ua_w (T_L - T_a)A_c}{\rho_L A_c \alpha_L C_p L} \right) = \rho_L A_c \alpha_L C_p L \frac{\partial}{\partial t} (T_L). \tag{2-16}
\]

Solving Eq. (2-16) for \(\frac{\partial T_L}{\partial t}\), one obtains

\[
\frac{\partial T_L}{\partial t} = \frac{L}{\rho_L A_L} \frac{\partial T}{\partial z} + \frac{\partial \omega}{\partial z} \frac{G(h_{fg} - h_{\ell})}{\rho_L A_L C_p L} - \frac{U_L a_w (T_L - T_{\text{pack}})}{\rho_L A_c \alpha_L C_p L} - \frac{Ua_w (T_L - T_a)}{\rho_L A_c \alpha_L C_p L}, \tag{2-17}
\]

where \(L\) is the water mass flux, and \(C_p L\) is the water specific heat. Equation (2-17) is a partial differential equation (PDE), first order in time and space with \(T_L\) being the dependent variable, and when solved yields the evolving liquid temperature distribution along the height of the evaporator as time proceeds.

Similarly, the application of the conservation of energy on the air/vapor mixture side yields,

\[
(m_a h_a + \tilde{m}_v h_v)_z - (m_a h_a + \tilde{m}_v h_v)_{z+dz} + d(m_{\text{r, evap}})h_{fg} + U_C (a - a_w) (T_{\text{pack}} - T_a) A_c dz + Ua_w (T_L - T_a) A_c dz = \frac{\partial}{\partial t} (m_a h_a + \tilde{m}_v h_v). \tag{2-18}
\]

56
Expanding the second term in Eq. (2-18) and substituting specific heat in the last term gives,

\[
\begin{align*}
-\frac{\partial}{\partial z} (m_a h_a + m_v h_v) dz + \frac{\partial}{\partial z} (m_{v,evap} h_{fg}) dz h_{fg} + U_G (a - a_w) (T_{pack} - T_a) A_c dz \\
+ U a_w (T_L - T_a) A_c dz = \frac{\partial}{\partial t} (m_a C_p a T_a + m_v C_p T_v).
\end{align*}
\]  

(2-19)

The first term of Eq. (2-19) is written as,

\[
-\frac{\partial}{\partial z} (m_a h_a + m_v h_v) dz = \left( -m_a C_p a \frac{\partial T_a}{\partial z} - m_v C_p v \frac{\partial T_v}{\partial z} - h_v \frac{\partial m_v}{\partial z} \right) dz,
\]

the latent heat of vaporization is expressed as \( h_{fg} (T_a) = h_v (T_a) - h_L (T_a) \), and the last term is written as

\[
\frac{\partial}{\partial t} (m_a C_p a T_a + m_v C_p T_v) = m_a (1 + \omega) C_{p_{mix}} \frac{\partial T_a}{\partial t}.
\]

The air/vapor mixture specific heat is evaluated as

\[
C_{p_{mix}} = \frac{m_a}{m_a + m_v} C_p a + \frac{m_v}{m_a + m_v} C_p v.
\]

(2-20)

With these substitutions, Eq. (2-19) may be rearranged as

\[
\frac{\partial T_a}{\partial t} = \frac{-G}{\rho a \alpha a} \frac{\partial T_a}{\partial z} \left( h_{fg} (T_L) - h_v (T_a) \right) G \frac{\partial \omega}{\partial z} + \frac{U_G (a - a_w)}{\rho a \alpha a (1 + \omega) C_{p_{mix}}} (T_{pack} - T_a)
\]

\[
+ \frac{U a_w}{\rho a \alpha a (1 + \omega) C_{p_{mix}}} (T_L - T_a).
\]

(2-21)

Equation (2-21) is a first order PDE in time and space for \( T_a \). The solution yields the evolution of the air/vapor mixture temperature distribution along the height of the evaporator.

Neglecting thermal diffusion through the packing, conservation of energy on the packed bed side gives

\[
U_L a_w (T_L - T_{pack}) A_c dz - U_G (a - a_w) (T_{pack} - T_a) A_c dz = \frac{\partial}{\partial t} (\rho_{pack} a_{pack} T_{pack} dz C_{p_{pack}} T_{pack}),
\]  

(2-22)
where \( C_{p_{\text{pack}}} \) is the packing material specific heat, and dividing by \( A_c \, dz \) gives,

\[
\frac{\partial T_{\text{pack}}}{\partial t} = \frac{1}{\rho_{\text{pack}} \alpha_{\text{pack}} C_{p_{\text{pack}}}} \left( U_L a_w (T_L - T_{\text{pack}}) - U_G (a - a_w)(T_{\text{pack}} - T_a) \right). \tag{2-23}
\]

Equation (2-23) is a first order PDE in time and space for \( T_{\text{pack}} \). The solution yields the evolution of the packed bed temperature distribution along the height of the evaporator.

**Condenser**

The process for vapor condensation flowing counter-currently through a packed bed in the presence of non-condensable gases is quite similar to that for evaporation, with the main difference being that heat and mass flows from the gas/vapor mixture to the liquid film. A differential volume depicting the direct contact condensation process is shown in Fig. 2-3.

![Condenser differential control volume depicting liquid/vapor/solid interactions.](image)

Figure 2-3. Condenser differential control volume depicting liquid/vapor/solid interactions.
The conservation of mass applied to the liquid phase of the control volume gives,

\[
(m_{L})_{z+dz} - (m_{L})_{z} + d\dot{m}_{v,\text{cond}} = \frac{\partial}{\partial t} (\rho_{L} \alpha_{L} A_{c} dz), \tag{2-24}
\]

Noting that \((m_{L})_{z+dz} - (m_{L})_{z} = \frac{\partial}{\partial z} (m_{L}) dz\), Eq. (2-24) is simplified to,

\[
\frac{\partial}{\partial z} (m_{L})_{z} = -\frac{\partial}{\partial z} (\dot{m}_{v,\text{cond}}). \tag{2-25}
\]

Likewise, the conservation of mass applied to the gas (air/vapor mixture) side is expressed as,

\[
(m_{a} + \dot{m}_{v})_{z} - (m_{a} + \dot{m}_{v})_{z+dz} - d\dot{m}_{v,\text{cond}} = \frac{\partial}{\partial t} (m_{\text{mix}}). \tag{2-26}
\]

The same simplifications used when deriving Eq. (2-5) can be used here to get,

\[
-\frac{\partial}{\partial z} (\dot{m}_{v})_{z} dz - d\dot{m}_{v,\text{cond}} = A_{c} dz (1 - \alpha_{L} - \alpha_{\text{pack}}) \frac{\partial}{\partial t} (\rho_{\text{mix}}). \tag{2-27}
\]

With the assumption of constant air and vapor densities, the time derivative of the mixture density is zero, and Eq. (2-27) is simplified to

\[
\frac{\partial}{\partial z} (\dot{m}_{v})_{z} = -\frac{\partial}{\partial z} (\dot{m}_{v,\text{cond}}). \tag{2-28}
\]

The application of the conservation of energy to the water side of the control volume depicted in Fig. 2-3 yields,

\[
(m_{L} h_{L})_{z+dz} - (m_{L} h_{L})_{z} + d(m_{v,\text{cond}}) h_{fg} + Ua_{w} (T_{a} - T_{L}) A_{c} dz
+ U_{L} a_{w} (T_{\text{pack}} - T_{L}) A_{c} dz
= \frac{\partial}{\partial t} (\rho_{L} A_{c} dz \alpha_{L} C_{pL} T_{L}) , \tag{2-29}
\]

where \( h_{fg} \) is the latent heat of condensation. Eq. (2-28) is simplified to,

\[
\frac{\partial}{\partial z} (m_{L} h_{L})_{z} + d(m_{v,\text{cond}}) h_{fg} + Ua_{w} (T_{a} - T_{L}) A_{c} dz + U_{L} a_{w} (T_{\text{pack}} - T_{L}) A_{c} dz
= \frac{\partial}{\partial t} (\rho_{L} A_{c} dz \alpha_{L} C_{pL} T_{L}) . \tag{2-29}
\]
The first term reflects the change in the liquid enthalpy along the z-direction, the second term accounts for the heat transport through condensation, the third term accounts for the convective heat transport from the gas/vapor mixture to the liquid, the fourth term accounts for the convective heat transport from the packing to the liquid, and the last term accounts for the transient variation of the liquid heat capacity. Noting that

\[
\frac{\partial}{\partial z} (m_L h_L)_z = m_L \frac{\partial}{\partial z} (h_L)_z + h_L \frac{\partial}{\partial z} (m_L)_z
\]

and substituting Eq. (2-25) for the second term, Eq. (2-29) is written as,

\[
(m_L h_L)_z = \frac{\partial}{\partial z} (h_L)_z + U_{wa} (T_a - T_L)_L + U_{wa} (T_{Pack} - T_L)_L = \rho_L \alpha_L C_p L \frac{\partial T_L}{\partial t}. \quad (2-30)
\]

Solving Eq. (2-30) for \( \frac{\partial T_L}{\partial t} \), one obtains

\[
\frac{\partial T_L}{\partial t} = \frac{L}{\rho_L \alpha_L} \frac{\partial T_L}{\partial z} - \frac{\partial}{\partial z} \frac{G (h_{fg} - h_L)}{C_p_L} + \frac{U_{wa} (T_a - T_L)}{\rho_t \alpha_L C_p L} + \frac{U_{wa} (T_{Pack} - T_L)}{\rho_L \alpha_L C_p L}.
\]

Equation (2-31) is a first order PDE in time and space for \( T_L \). The solution yields the evolution of the liquid temperature distribution along the height of the condenser.

Similarly, the application of the conservation of energy on the air/vapor mixture side yields,

\[
(m_a h_a + m_v h_v)_{z+dz} - (m_a h_a + m_v h_v)_{z} = -d(m_{v,cond}) h_{fg} - U_G (a - a_w)(T_a - T_{Pack}) A_z dz - U_{wa} (T_a - T_L)_L A_z dz = \frac{\partial}{\partial t} (m_a h_a + m_v h_v). \quad (2-32)
\]

Expanding the second term in Eq. (2-32) and substituting specific heat in the last term result in,
\[-\frac{\partial}{\partial z}(\dot{m}_a h_a + \dot{m}_v h_v)dz - d(\dot{m}_v, \text{cond}) h_{fg} - U_G (a - a_w) (T_a - T_{\text{pack}}) A, dz\]

\[-U a_w (T_a - T_L) A, dz = \frac{\partial}{\partial t} (m_a C_p T_a + m_v C_p T_v)\] (2-33)

The first term reflects the change in the gas/vapor mixture enthalpy along the z-direction, the second term accounts for the heat transport through condensation, the third term accounts for the convective heat transport from the gas/vapor mixture to the packing, the fourth term accounts for the convective heat transport from the gas/vapor mixture to the liquid, and the last term accounts for the transient variation of the gas/vapor mixture heat capacity. The first term in Eq.(2-33) is written as

\[-\frac{\partial}{\partial z}(\dot{m}_a h_a + \dot{m}_v h_v)dz = \left(-\dot{m}_a C_p \frac{\partial T_a}{\partial z} - \dot{m}_v C_p \frac{\partial T_v}{\partial z} - h_v \frac{\partial \dot{m}_v}{\partial z}\right)dz\]. With similar simplifications used in the derivation of Eq.(2-21), Eq.(2-33) can be simplified to

\[\frac{\partial T_a}{\partial t} = \frac{-G}{\rho_a \alpha_a} \frac{\partial T_a}{\partial z} - \frac{(h_{fg} (T_L) - h_v (T_v)) G}{\rho_a \alpha_a (1 + \omega) C_p_G} \frac{\partial \omega}{\partial z} - \frac{U_G (a - a_w)}{\rho_a \alpha_a (1 + \omega) C_p_G} (T_a - T_{\text{pack}}) - \frac{U a_w}{\rho_a \alpha_a (1 + \omega) C_p_G} (T_a - T_L)\]. (2-34)

Equation (2-34) is a first order PDE in time and space for $T_a$. The solution yields the evolution of the air/vapor mixture temperature distribution along the height of the condenser. Similar to the derivation of Eq.(2-23), the energy equation for the packed bed gives

\[\frac{\partial T_{\text{pack}}}{\partial t} = \frac{1}{\rho_{\text{pack}} \alpha_{\text{pack}} C_{p_{\text{pack}}}} \left(U_G (a - a_w)(T_a - T_{\text{pack}}) - U L a_w (T_{\text{pack}} - T_L)\right)\]. (2-35)

Equation (2-35) is a first order PDE in time and space for $T_{\text{pack}}$. The solution yields the evolution of the packed bed temperature distribution along the height of the condenser.
Following Li et al. [36], the humidity ratio is evaluated based on the local air temperature. It is noted that the relative humidity is approximately unity during the condensation process, and with constant total pressure, the humidity ratio depends only on the local air temperature,

\[
\frac{\partial \omega}{\partial z} = \frac{\partial T_a}{\partial z} \frac{P}{P - P_{sat}(T_a)} \omega_m (b - 2cT_a + 3dT_a^2) .
\]  

(2-36)

An empirical representation of the saturation curve for water is,

\[
P_s(T) = a e^{\left(c - \frac{d}{T} + \frac{e}{T^2}\right)}
\]  

(2-37)

where empirical constants are \(a=0.611379\), \(b=0.0723669\), \(c=2.78793 \times 10^{-4}\), \(d=6.76138 \times 10^{-7}\), \(P_{sat}(\text{kPa})\) is the saturation pressure, and \(T (\degree C)\) is the temperature.

Eqs. (2-31) and (2-34) are coupled first order pde’s, and their solutions yield the temperature evolution of the liquid, and gas/vapor, respectively. The conservation equations for the evaporator and condenser require closure, and the closure relations are described next.

**Heat and Mass transfer Coefficients**

In order to evaluate the heat transfer coefficients on the liquid and gas side, the analogy between the heat and mass transfer is used by Klausner et al. [35]. The heat transfer coefficients are computed as follows,

Heat transfer coefficient on the liquid side

\[
\frac{N}{\rho L \frac{h}{L}} = \frac{S h}{S \frac{h}{L}} ,
\]  

(2-38)

\[
U_L = k_L (\rho_L C_p K_L \frac{K_L}{D_L})^{1/2} ,
\]  

(2-39)

Heat transfer coefficient on the gas side
\[ \frac{N}{P} = \frac{S}{S} \left( \frac{k}{C} \right)^{1/3}, \quad (2-41) \]

\[ U_G = k_G \left( \rho_G C_P \right)^{1/3} \left( \frac{K_G}{D_G} \right)^{2/3}, \quad (2-42) \]

**Overall heat transfer coefficient**

\[ U = \left( U_L^{-1} + U_G^{-1} \right)^{-1}, \quad (2-43) \]

where \( k_L \) and \( k_G \) denote the thermal conductivity of the liquid and gas, respectively, \( D_L \) and \( D_G \) denote the molecular diffusion coefficient of the liquid and gas, respectively, and \( k_L \) and \( k_G \) denote the mass transfer coefficient of the liquid and gas side, respectively. Klausner et al. [35] and Li et al. [36, 37] used Onda’s correlation [41] to evaluate mass transfer coefficients (Appendix B). It is noted here that Onda’s correlation was developed based on steady heat and mass transfer data. Here it is postulated that the thermal and mass species boundary layers will exhibit quasi-steady behavior, and that they will quickly adjust to changes in local velocity and temperature. The coefficient \( C \) in the gas side mass transfer correlation used is 2.0. Under such conditions, it is expected that Onda’s correlation will be useful to close the transient equations (2-11), (2-17), (2-21), (2-23), (2-31), (2-34), (2-35), and (2-36). Experimental verification is required.

**Storage Tank and Solar Collector**

Thermal storage tanks are widely used in renewable thermal energy systems and waste heat recovery systems especially when the harvested energy input and demand are not matched. The saline water storage tank is used to enable operating the solar DDD system in a re-circulating mode. The volume, temperature and salinity of the saline water in the storage tank are time varying parameters that characterize the
dynamic behavior of the desalination process. The effects of the various operating conditions such as the tank size, inlet and exit saline water flow rates, and inlet water temperatures on the thermal behavior of the storage tank must be investigated. Since water evaporation in the evaporator changes with time due to the change in the solar heat input, the mass flow rate of the water exiting the evaporator and then entering the saline water storage tank is not constant. The application of the mass conservation on the water in the saline water tank is expressed as,

$$\frac{dM_w}{dt} = \dot{m}_{w,\text{in}} - \dot{m}_{w,\text{out}}$$, \hspace{1cm} (2-44)

where $M_w$ is the total mass of water in the storage tank, $\dot{m}_{w,\text{in}}$ and $\dot{m}_{w,\text{out}}$ are the water flow rates entering and exiting the water tank respectively. Dividing Eq. (2-44) by the water density yields,

$$\frac{dV_w}{dt} = \frac{1}{\rho}(\dot{m}_{w,\text{in}} - \dot{m}_{w,\text{out}})$$, \hspace{1cm} (2-45)

where $V_w$ is the water volume. Eq. (2-45) is used to solve for the variation in the saline water volume in the storage tank at any time. In order to determine the variation of the water temperature in the storage tank, conservation of energy can be applied on the control volume of the water in the storage tank. The current derivations assume perfect mixing of water in the storage tank such that the water temperature in the tank is uniform. The application of the conservation of energy results in,

$$\frac{dE_{c,v}}{dt} = \dot{m}_{w,\text{in}} h_{w,\text{tank,in}} - \dot{m}_{w,\text{out}} h_{w,\text{tank,out}} - h_T A_T (T_{w,\text{tank}} - T_{\text{amb}})$$, \hspace{1cm} (2-46)

where $E_{c,v}$ is the thermal energy of the saline water in the storage tank, $h_{w,\text{tank,in}}$ and $h_{w,\text{tank,out}}$ is the enthalpy of the water entering and exiting the storage tank, $h_T$ is the
heat transfer coefficient between the water tank and the surrounding, $A_T$ is the storage tank surface area. For simplicity, a perfectly insulated tank with a uniform temperature field in the tank is assumed, and Eq. (2-46) is simplified to,

$$\frac{d(M_w C_p T_{w, tank})}{dt} = m_{w, in} C_p T_{w, in} - m_{w, out} C_p T_{w, out}.$$  \hspace{1cm} (2-47)

With the assumption of uniform temperature in the tank, $T_{w, tank} = T_{w, tank}$. Eq. (2-47) is simplified to

$$\frac{dT_{w, tank}}{dt} = \frac{1}{M_w} \left( \frac{dM_w}{dt} T_{w, tank} + m_{w, in} T_{w, in} - m_{w, out} T_{w, tank} \right).$$  \hspace{1cm} (2-48)

Using Eq. (2-44) to substitute for $\frac{dM_w}{dt}$, Eq. (2-48) is simplified further to

$$\frac{dT_{w, tank}}{dt} = \frac{m_{w, in}}{M_w} (T_{w, tank, in} - T_{w, tank}).$$  \hspace{1cm} (2-49)

Eq. (2-49) is an initial value problem and can be solved using explicit finite difference scheme to obtain the temperature in the saline water storage tank. It should be noted that the same governing equation developed for the saline water tank is also used for the fresh water tank. Energy balance applied on the solar collector gives,

$$I_{solar} A_{solar} \eta_{solar} = m_{w, solar} C_p (T_{solar, out} - T_{solar, in}),$$  \hspace{1cm} (2-50)

where $I_{solar}$ is the solar intensity, the solar collector efficiency, $\eta_{solar}$, is assumed to be 100% for simplicity, $A_{solar}$ is the solar collector surface area, $T_{solar, in}$ and $T_{solar, out}$ are the temperatures of water entering and exiting the solar collector. The evaporator inlet water temperature is equal to the water temperature exiting the solar collector, which is evaluated by setting $T_{e, evap, in} = T_{solar, out}$, Eq. (2-50) gives $T_{e, evap, in} = T_{solar, in} + \frac{I_{solar} A_{solar} \eta_{solar}}{m_{w, solar} C_p}$. 

65
Solution of System of Equations

Computational Method

Equations (2-11), (2-17), (2-21) and (2-23) comprise a set of coupled differential equations that describe the heat and mass transfer process within the evaporator. These equations are solved for the evolution of the humidity ratio, water temperature, air/vapor temperature, and the packed bed temperature distributions within the evaporator. Similarly, equations (2-31), (2-34), (2-35), and (2-36) are used to solve for the evolution of humidity ratio, water temperature, air/vapor temperature, and the packed bed temperature distributions within the condenser. To enable solution of the governing equations, appropriate boundary and initial conditions must be specified. The initial and boundary conditions are applied as follows:

1. Initial conditions:

\[ T_{L,\text{evap}}(z, t = 0) = T_{\text{amb}}, \quad T_{a,\text{evap}}(z, t = 0) = T_{\text{amb}}, \quad T_{\text{pack, evap}}(t = 0) = T_{\text{amb}}, \]
\[ T_{L,\text{cond}}(z, t = 0) = T_{\text{amb}}, \quad T_{a,\text{cond}}(z, t = 0) = T_{\text{amb}}, \quad T_{\text{pack, cond}}(t = 0) = T_{\text{amb}}, \]

2. Boundary conditions:

\[ T_{L,\text{evap}}(z = H, t) = T_{l,\text{evap, in}}, \quad T_{a,\text{evap}}(z = 0, t) = T_{a,\text{evap, in}} \]
\[ T_{L,\text{cond}}(z = H, t) = T_{l,\text{cond, in}}, \quad T_{a,\text{cond}}(z = 0, t) = T_{a,\text{cond, in}} = T_{a,\text{evap}}(z = H, t) \]

The numerical scheme used to solve the governing equations is an explicit finite difference method. A uniform grid size is employed to generate the evaporator and condenser mesh. The Courant–Friedrichs–Lewy (CFL) number for the numerical scheme of the evaporator is expressed as,
\[ CFL_1 = a_1 \frac{\Delta t}{\Delta z} = \left( \frac{L}{\alpha L \rho L} \right)_{\text{evap}} \frac{\Delta t}{\Delta z}, \quad (2-51) \]

and for the condenser is expressed as,

\[ CFL_2 = a_2 \frac{\Delta t}{\Delta z} = \left( \frac{L}{\alpha L \rho L} \right)_{\text{cond}} \frac{\Delta t}{\Delta z}, \quad (2-52) \]

A proper combination of \( \Delta z \) and \( \Delta t \) were used \((\Delta t/\Delta z = 0.2)\), to ensure the accuracy and stability of the computation. For \( L = 1.0 \text{ kg/m}^2\text{-sec} \), the resulting CFL = 0.02 which is much smaller than the CFL number required to ensure stability of the numerical scheme. The grid size in the z-direction is 0.05 m, and the time step is 0.01 sec. A grid refinement study was performed using 101 and 21 grids. The study revealed that the computed air and water temperature difference between the two cases does not exceed 0.1°C.

**Typical Computational Results**

Previous studies by Klausner et al. [35] and Li et al. [36], assumed the inlet water and air temperatures to the evaporator and condenser to be constant; thus a steady state solution was appropriate for the given operating condition. In other applications, such as in solar driven desalination, the inlet water and air temperatures to the evaporator and condenser are time dependent and vary during operation. Therefore, it is of instructive to study the typical thermal behavior for varying inlet water and air temperatures. Furthermore, it is of interest to determine fresh water production produced by the DDD for a given heat input as opposed to a given inlet water temperature. The solar thermal power used in this study is for the city of Jacksonville, Florida, U.S. It is located at 30.23 latitude and 81.68 longitude. The solar thermal
power used is a typical one for cities in the southern United States, north Africa, and the Middle East.

The evaporator inlet water temperature is specified such that it varies continuously over a 12 hour period. For simplicity, a perfectly insulated tank with a uniform temperature field in the tank is assumed. It should be stressed here that changing any of the operating conditions such as the water volume in the tank, water flow rates, solar intensity, and the solar collector efficiency directly affects the inlet water temperature. The temperature history shown in Fig. 2-4 is simulated based on a 300 liter pool of water recirculated through the evaporator and heated using a 6 m² solar panel irradiated with a measured solar intensity [42] for the twelve hour duration. The temperature field results for the evaporator will be presented for a typical set of operating conditions, \( L = 1 \, \text{kg/m}^2\text{-sec}, \) gas mass flux \( G = 0.5 \, \text{kg/m}^2\text{-sec}, \) \( H = 1 \, \text{m}, \) \( T_{a,\text{evap,in}} = 25^\circ\text{C}, \) \( T_{\text{amb}} = 25^\circ\text{C}. \)

Figs. 2-5, 2-6, and 2-7 show the evolution of the water, air and packed bed temperature fields within the evaporator. The evaporator inlet water temperature has a significant influence on the water temperature profile along the evaporator. Fig. 2-8 shows the evolution of the humidity ratio in the evaporator. Comparing Fig. 2-5 and 2-7, it is clear that the packed bed temperature is very close to the water temperature. This is due to the fact that the liquid side heat transfer coefficient is an order of magnitude greater than that on the gas side.

Similarly, the solution to the condenser temperature fields are shown for a typical set of operating conditions, \( L = 2.0 \, \text{kg/m}^2\text{-sec}, \) gas mass flux \( G = 0.5 \, \text{kg/m}^2\text{-sec}, \) \( H = 1 \, \text{m}, \) \( T_{L,\text{cond,in}} = 20^\circ\text{C}. \) The air temperature exiting the evaporator in the previous example
is used as an input to the condenser. Figs. 2-9, 2-10 and 2-11 show the evolution of the water, air and packed bed temperature fields within the condenser. Fig. 2-12 shows the evolution of the humidity ratio in the condenser. The same grid size and time steps used for the evaporator are also used for the condenser. While the results shown in Figs. 2-5, 2-6, 2-7, 2-8, 2-9, 2-10, 2-11, and 2-12 are reasonable, the model requires experimental validation.

Figure 2-4. Specified evaporator inlet water temperature.
Figure 2-5. Temporal and spatial water temperature distribution along the height of the evaporator.

Figure 2-6. Temporal and spatial air temperature distribution along the height of the evaporator.
Figure 2-7. Temporal and spatial packing temperature distribution along the height of the evaporator.

Figure 2-8. Temporal and spatial humidity ratio distribution along the height of the evaporator.
Figure 2-9. Temporal and spatial water temperature distribution along the height of the direct contact condenser.

Figure 2-10. Temporal and spatial air temperature distribution along the height of the direct contact condenser.
Figure 2-11. Temporal and spatial packing temperature distribution along the height of the condenser.

Figure 2-12. Temporal and spatial humidity ratio distribution along the height of the condenser.
Experimental Apparatus

The experimental facility shown in Fig. 3-1 consists of three main components: evaporator, direct contact condenser, and electric heater. The facility is used for the distillation of seawater. The evaporator and condenser are comprised of three main sections: top, middle, and bottom. The top section is constructed from 25.4 cm (10” nominal) ID PVC pipe, and includes the water spray nozzle and air exit ducts. The top section of the evaporator is fitted with a high efficiency cellular mist eliminator to prevent aerosolized water from being carried over to the condenser with air. The mist eliminator is constructed with UV protected PVC material. The middle section is a transparent column constructed with 24.1 cm ID acrylic tubing with wall thickness of 0.64 cm and 1 m height. The bottom section is also constructed with 25.4 cm (10” nominal) ID PVC pipe, and includes the air distributor and water collection drain. The three sections are connected via PVC bolted flanges. The evaporator is connected to the condenser via flexible duct. The evaporator, condenser and connecting pipes and ducts are insulated to prevent heat loss to the ambient as shown in Fig. 3-2.

Seawater is drawn from the storage tank and circulated via a corrosion resistant centrifugal pump, and the water flow rate is controlled with a valve installed at the discharge of the saline water pump. Air is circulated through the evaporator and condenser using a 3.68 kW blower. Fresh water to the condenser is sustained at a low temperature using a heat exchanger. Water is directed to an electric heater with a maximum capacity of 15 kW. The power output from the heater is computer controlled so as to simulate heat captured from a solar collector on a typical summer day [8].
simulated solar thermal power for a collector area of 10 m² is shown in Fig. 3-3 for a 150 minute transient.

![Schematic diagram of the experimental direct contact evaporator and condenser desalination facility.](image)

Figure 3-1. Schematic diagram of the experimental direct contact evaporator and condenser desalination facility.

The computer controls the electric heater using two solid state relay (SSR) that is connected to a data acquisition system as shown in Fig.3-4. One of the SSR is installed on finned heat sink, and the other SSR is installed on a large Aluminum plat to remove the heat developed due to the nominal voltage drop. The data acquisition system sends signal to the SSR to open or close the circuit so as to match the solar thermal power needed. The two relays used are SSR240DC25 which are manufactured by Omega. They have operating temperature range of -20 to 80°C. The turn-on and turn-off time is within 0.02 and 0.03 sec, respectively.
Figure 3-2. Solar diffusion driven desalination facility.

Figure 3-3. Simulated Solar Thermal Power for 10 m² of Collector Area.
The water distributor used in the evaporator and condenser is manufactured by Allspray. They are brass full cone nozzles with 65° spray angle and are designed for a uniform solid cone spray. The spray nozzle is designed to allow a water capacity of $2.5\times10^{-4}$ m$^3$/s. The nozzles are placed at a sufficient height from the packing material of both the tower and the condenser to ensure that spray covers the entire packing area.

The polypropylene packing material, depicted in Fig. 3-5, used for the evaporator and condenser is HD Q-PAC, and was specially cut using a hotwire so that it fits tightly into the main body of the evaporator and the condenser. The specific area of the packing is 267 m$^2$/m$^3$, and the effective diameter of the packing is 17 mm when the packing is mounted in a vertical pin orientation (condenser), and the specific area and effective diameter are 236 m$^2$/m$^3$ and 20 mm when the packing is mounted in a horizontal pin orientation (evaporator). The packed bed specific heat capacity is 2.35 kJ/kg·K, the
density is 850 kg/m$^3$, and the void fraction is 87.8%. Two heavy duty water tanks, each having a capacity of 435 liters, are used as seawater and freshwater storage tanks.

Figure 3-5. Polypropylene packing material

The temperatures of water and air are measured using type E thermocouples. All of the thermocouples are manufactured by Omega and factory calibrated with an estimated uncertainty of ±0.2°C. The temperature is measured at 10 different locations within the facility. The relative humidity is measured with three HMD70Y resistance-type humidity and temperature transmitters manufactured by Vaisala Corp. The humidity and temperature transmitters have a 0-10 V output signal and have been factory calibrated. The operating range for the relative humidity is 0-100%, while the operating range for the temperature is -20 to 80°C. The measurement uncertainty is
±1.185×10⁻³ kgv/kga for the absolute humidity. The feed water flow rate is measured using a turbine flow meter constructed by Proteus Industries Inc. with a range from 0.25-2.0×10⁻⁴ m³/s, and the chilled water flow rate is measured using a magnetic flow meter constructed by Omega Inc. with a range from 0.063-6.3×10⁻⁴ m³/s. Both flow meters are calibrated using the catch and weigh method. The flow meters have a 0-5 V output, and the measurement uncertainties are ± 6.7×10⁻⁶ m³/s and ±3.15×10⁻⁶ m³/s, respectively. The air flow rate is measured with a model 620S smart insertion thermal air flow meter constructed by Sierra Instrument Inc. The flow meter has a response time of 200 ms with changes in air mass flow rate. The air flow meter has a microprocessor-based transmitter that provides a 0-10 V output signal. The flow meter operating range is 0-0.5 m³/s of air. The measurement uncertainty is ± 9.6×10⁻³ m³/s. The pressures at the inlet and exit of the evaporator are measured with two Validyne P2 static pressure transducers. All of the wetted parts are constructed with stainless steel. They are shock resistant and capable of operating in a temperature range of −20 to 80°C. The transducers operate in the range of 0-0.34 atm and the associated uncertainty is ±8.5×10⁻⁴ atm. The pressure drop across the packing is measured with a DP45-18 magnetic reluctance differential pressure transducer with a range of 0-550 Pa. The measurement uncertainty is ± 1.25 Pa. All the experimental measurements are recorded with a frequency of 5 Hz. An average value is recorded every 1 second.

Experimental and Computational Results

Evaporator Temperature Profiles for Different Air Mass Fluxes

To determine the evolution of the water, air, and packed bed temperature fields in the evaporator, the measured inlet water and air temperatures are used as an input to the numerical model. For the system operating in the re-circulating configuration, Figs.
3-6 and 3-7 compare the computed and experimental water and air temperatures entering and exiting the evaporator and humidity ratio for different air mass fluxes. The computed water and air temperatures and humidity ratio exiting the evaporator are in satisfactory agreement with those measured. The exit water and air temperatures decrease with increasing air mass flux. The deviation between the measured and computed water temperatures is within 4°C, and within 3°C for the air temperature. As shown in Figs. 3-6 and 3-7, the computed transient variations agree very well with those measured.

**Evaporator Temperature Profiles for Different Water Mass Fluxes**

Figures 3-8 and 3-9 show the computed and measured water and air temperatures and humidity ratio entering and exiting the evaporator for different water mass fluxes. The computed exit water and air temperatures and humidity ratio are in a satisfactory agreement with those measured. As can be seen, the evaporator inlet water temperature is lower for the 1 kg/m²·s water flux than that for 2.15 kg/m²·s; and this is because the same heat input, 15 kW, is applied for both experiments. Thus, the exit air temperature profile is greater for the low water mass flux. The exit water and air temperatures decrease with increasing water mass flux. The deviation between the measured and computed temperature profiles is typically within 3°C for water and 2°C for the air. The predicted air and water temperature and humidity profiles shown in Figs. 3-8 and 3-9 agree satisfactorily with those measured.

**Condenser Temperature Profiles for Different Air Mass Fluxes**

Similarly, the measured inlet water and air temperatures and humidity ratio to the condenser are used as an input to the numerical model to determine the evolution of the water, air, and packed bed temperature fields and humidity ratio within the condenser.
Figs. 3-10 and 3-11 compare the computed and experimental water and air temperatures and humidity ratio entering and exiting the condenser for different air mass fluxes. The computed water temperature profile exiting the condenser agrees, within approximately 3°C, with the measured water temperature profile. As can be seen in Figs. 3-10 and 3-11, the model prediction for the air temperature profile is approximately within 3°C.

Condenser Temperature Profiles for Different Water Mass Fluxes

Figs. 3-12 and 3-13 compare the computed and experimental water and air temperatures and humidity ratio entering and exiting the condenser for different water mass fluxes. The computed water temperature profile at the exit of the condenser agrees well with that measured. The computed condenser exit air temperature profile also agrees well with that measured and a deviation within 3°C is observed. In general the shape of the transient air and water temperature profiles and humidity ratio are well predicted with the model.
Figure 3-6. Evaporator transient temperature and humidity ratio variation (G=0.5 kg/m²-s).

Figure 3-7. Evaporator transient temperature and humidity ratio variation (G=1.02 kg/m²-s).
Figure 3-8. Evaporator transient temperature and humidity ratio variation (L=1.0 kg/m²-s).

Figure 3-9. Evaporator transient temperature and humidity ratio variation (L=2.15 kg/m²-s).
Figure 3-10. Condenser transient temperature and humidity ratio variation (G=1.02 kg/m²-s).

Figure 3-11. Condenser transient temperature and humidity ratio variation (G=0.5 kg/m²-s).
Figure 3-12. Condenser transient temperature and humidity ratio variation (L=2.0 kg/m$^2$-s).

Figure 3-13. Condenser transient temperature and humidity ratio variation (G=1.0 kg/m$^2$-s)
CHAPTER 4
SENSITIVITY AND DYNAMIC RESPONSE

It is of interest to explore the sensitivity of the direct contact evaporator performance with variable operating conditions and its dynamic response to sudden changes to the input fluid states. Here the sensitivity to the evaporator exit air and water temperatures is examined for a range of liquid volume fraction (liquid holdup), packing wettability, and packing heat capacity. In addition, the dynamic response of the evaporator exit air and water temperatures is explored for a sudden variation in inlet air and water temperatures. For the analysis that follows, the liquid mass flux is held at \( L = 2.0 \text{ kg/m}^2\cdot\text{s} \) and the air mass flux is held at \( G = 1.0 \text{ kg/m}^2\cdot\text{s} \), \( L/G = 2.0 \).

The following dimensionless parameters are introduced to nondimensionalize the governing equations of the evaporator and condenser:

\[
\begin{align*}
    z^* &= \frac{z}{H} \\
    t^* &= \frac{tG}{H \rho_a} \\
    \eta_L &= \frac{h_L}{q_{\text{max}}} \\
    \eta_{fg} &= \frac{h_{fg}}{q_{\text{max}}} \\
    \eta_Y &= \frac{h_Y}{q_{\text{max}}} \\
    T_L^* &= \frac{T_L - T_{\text{amb}}}{q_{\text{max}}/C_p_L} \\
    T_a^* &= \frac{T_a - T_{\text{amb}}}{q_{\text{max}}/C_p_L} \\
    T_{\text{pack}}^* &= \frac{T_{\text{pack}} - T_{\text{amb}}}{q_{\text{max}}/C_p_L}
\end{align*}
\]

(4-1)

where \( T_{\text{amb}} \) is the ambient temperature, and \( q_{\text{max}} \) is the maximum specific heat input into the system; it is the ratio of the maximum heat rate to the liquid mass flow rate. The governing conservation equations for the liquid, gas, and packed bed in the evaporator are cast in dimensionless form as

\[
\begin{align*}
    \frac{\partial T_L^*}{\partial t^*} &= \frac{1}{\alpha_L \rho_L} \frac{L}{G} \frac{\partial T_L^*}{\partial z^*} - \frac{1}{\alpha_L \rho_L} \frac{d}{dz} (\eta_{fg} - \eta_L) - C_1 \frac{1}{\alpha_L \rho_L} \left( T_L^* - T_{\text{pack}}^* \right) \\
    - C_2 \frac{1}{\alpha_L \rho_L} \left( T_L^* - T_a^* \right) 
\end{align*}
\]

(4-2)
Similarly, dimensionless governing equations for the condenser are given as,

\[ \frac{\partial T_{L}^*}{\partial t^*} = \frac{1}{\alpha_a} \frac{\partial T_{L}^*}{\partial z^*} - \frac{1}{\alpha_a} \frac{\partial T_{L}^*}{\partial z^*} \frac{d\omega}{dz^*} (\eta_{fg} - \eta_s) + C_1 \frac{1}{\alpha_a} \frac{1}{1 + \omega} (T_{pack}^* - T_L^*) \]

\[-C_4 \frac{1}{\alpha_a} \frac{1}{1 + \omega} (T_a^* - T_L^*) \]  

, \hspace{1cm} (4-6)

\[ \frac{\partial T_{a}^*}{\partial t^*} = \frac{1}{\alpha_a} \frac{\partial T_{a}^*}{\partial z^*} - \frac{1}{\alpha_a} \frac{\partial T_{a}^*}{\partial z^*} \frac{d\omega}{dz^*} (\eta_{fg} - \eta_s) + C_1 \frac{1}{\alpha_a} \frac{1}{1 + \omega} (T_{pack}^* - T_a^*) \]

\[ -C_4 \frac{1}{\alpha_a} \frac{1}{1 + \omega} (T_a^* - T_L^*) \]  

, \hspace{1cm} (4-7)

\[ \frac{\partial T_{pack}^*}{\partial t^*} = C_5 \frac{P_a}{\rho_{pack}} (T_{a}^* - T_{pack}^*) - C_6 \frac{P_a}{\rho_{pack}} (T_{pack}^* - T_L^*) \]  

, \hspace{1cm} (4-8)

\[ \frac{\partial \omega}{\partial z^*} = \frac{\partial T_{pack}^*}{\partial z^*} P - P_{sat}^* \omega_m \frac{q_{solar, max}}{C_p} (b - 2cT_a + 3dT_a^2) \]  

, \hspace{1cm} (4-9)

where \( C_1 = \frac{U_{L}a_w H}{C_p L G} \), \( C_2 = \frac{U_{a} H}{C_p L G} \), \( C_3 = \frac{U_{G}(a - a_w) H}{C_p L G} \), \( C_4 = \frac{U_{a} H}{C_p L G} \), \( C_5 = \frac{U_{L} a_w H}{\alpha_{pack} L G} \), and \( C_6 = \frac{U_{G}(a - a_w) H}{\alpha_{pack} L G} \). The dimensionless initial conditions for the evaporator and condenser are given, respectively, as

\[ T_{L}^*(t^* = 0, z^*) = 0, \hspace{1cm} T_{a}^*(t^* = 0, z^*) = 0, \hspace{1cm} T_{pack}^*(t^* = 0, z^*) = 0 \]  

\[ T_{L}^*(t^* = 0, z^*) = 0, \hspace{1cm} T_{a}^*(t^* = 0, z^*) = 0, \hspace{1cm} T_{pack}^*(t^* = 0, z^*) = 0 \]  

(4-10a)

(4-10b)
The dimensionless boundary conditions for the evaporator and condenser are given, respectively, as

\[
T_e^*(t^*, z^* = 1) = \frac{T_e(t, z = H_{evap}) - T_{amb}}{q_{max} / Cp_L}, \quad T_a^*(t^*, z^* = 0) = \frac{T_a(t, z = 0) - T_{amb}}{q_{max} / Cp_L}
\]

(4-11a)

\[
T_c^*(t^*, z^* = 1) = \frac{T_c(t, z = H_{cond}) - T_{amb}}{q_{max} / Cp_L}, \quad T_a^*(t^*, z^* = 0) = \frac{T_a(t, z = 0) - T_{amb}}{q_{max} / Cp_L}
\]

(4-11b)

The numerical scheme used to solve the dimensionless governing equations is an explicit finite difference method. A uniform grid size is employed to generate the evaporator and condenser mesh. If the evaporator and condenser have different heights, the dimensionless time for the evaporator and condenser is given as,

\[
t_1^* = \frac{tG}{H_{evap} \rho_a},
\]

(4-12)

\[
t_2^* = \frac{tG}{H_{cond} \rho_a} = \frac{H_{evap}}{H_{cond}} t_1^*,
\]

(4-13)

where \(t_1^*\), and \(t_2^*\) are the non-dimensional time in the governing equation of the evaporator, and condenser, respectively. Since the same dimensional time step (\(\Delta t\)) must be used to solve the governing equations for the evaporator and condenser, the non-dimensional time step for the condenser can be expressed as,

\[
\Delta t_2^* = \Delta t_1^* \frac{H_{evap}}{H_{cond}},
\]

(4-14)

where \(\Delta t_1^*\), \(\Delta t_2^*\) are the non-dimensional time step for the evaporator and condenser, respectively. The CFL condition imposed to ensure convergence of the evaporator and condenser numerical scheme are given, respectively, as,

\[
CFL_{evap} = a_e \frac{\Delta t_1^*}{\Delta z_1^*} = \left(1 - \frac{\rho_a L}{\alpha_L \rho_L G}_{evap}\right) \frac{\Delta t_1^*}{\Delta z_1^*},
\]

(4-15)
The same non-dimensional distance ($\Delta z^*$) is used for the evaporator and condenser. Therefore, $\Delta z_1^* = \Delta z_2^* = \Delta z^*$. Since the variation in the water and air densities, and liquid hold-ups in the evaporator and condenser are very small, $CFL_2$ in Eq.(4-16) can be related to $CFL_1$ as,

$$CFL_2 = CFL_1 \frac{L_{\text{cond}}}{L_{\text{evap}}} \frac{H_{\text{evap}}}{H_{\text{cond}}} \Delta t_1^*$$  \hspace{1cm} (4-17)

A proper combination of $\Delta z^*$ and $\Delta t^*$ are used to ensure the accuracy and stability of the computation. The grid size $\Delta z^*$ used is 0.05, and the time step $\Delta t^*$ is 0.01. For $L/G = 1.0$ and $L_{\text{cond}}/L_{\text{evap}} = 2$, the resulting $CFL_1$ and $CFL_2$ are 0.024 and 0.048, respectively.

**Sensitivity to Liquid Hold-up, Packing Wettability, and Packing Heat Capacity**

The liquid volume fraction (hold-up) in the packed bed for the experimental conditions considered in previous sections is approximately 1% and this value is used for the model predictions. Consider a direct contact evaporator where the initial water and air temperatures in the evaporator are 25°C. The dimensionless inlet water temperature is set to 7.54, and the transient response of the dimensionless exit air and water temperatures as the system approaches steady state is shown in Fig. 4-1 for liquid holdups of $\alpha_L = 1\%$ and 10\% (polypropylene packing). The response time for the exit air temperature is not very sensitive to the liquid hold-up. However, the response time for the exit liquid temperature is significantly shorter for $\alpha_L = 1\%$ compared with $\alpha_L = 10\%$. This is expected since the liquid within the evaporator has a greater heat
capacity with greater hold-up. The steady-state exit temperatures are independent of the liquid hold-up as expected.

For the same flow conditions considered for Fig. 4-1 and a liquid hold-up of 1%, the dynamic response of the exit water and air temperatures are considered for a sudden increase in dimensionless inlet water temperature from 0 to 7.54 for two different ratios of wetted to total specific area. Fig. 4-2 shows the response for $a_w/a=0.5$ and $a_w/a=1.0$. It is observed that better wettability does not influence the response time, but it does result in improved heat transfer with a higher exit air temperature and lower exit water temperature.

The transient response of the exit water and air temperature profile for different packed bed materials is investigated. Stainless steel and polypropylene packed beds with the same structure and specific area are considered. The Stainless steel bed has a specific heat of 0.5 kJ/kg-K and density of 8000 kg/m$^3$. The specific heat and density for the polypropylene packing is 2.35 kJ/kg-K and 850 kg/m$^3$, respectively. The exit air and water temperature responses for the two different packed bed materials and same operating conditions as above ($\alpha_L=1\%$) are shown in Fig. 4-3. Due to the high thermal inertia of the Stainless steel packing the response time for the exit water temperature is quicker for the polypropylene bed. The steady state temperatures do not depend on the heat capacity of the packing material, as expected.

**Dynamic Response to a Step Change in Temperature**

Once the evaporator system reaches steady state as shown in Fig. 4-1 ($\alpha_L=1\%$), it is of interest to examine how the water and air outlet temperatures will respond to step changes at the inlet. Fig. 4-4 displays the dynamic response for a sudden drop and
increase in dimensionless inlet water temperature from 7.54 to 5.86 and back to 7.54 for a flow condition where L/G=2. It is observed that the outlet water temperature lags the response of the outlet air temperature. Fig. 4-5 displays the response for a sudden change in dimensionless inlet air temperature from 0 to 0.84 and back down to 0. Here, the response of the water slightly leads that of the air. Fig. 4-6 displays the response for a sudden change in dimensionless inlet water temperature for L/G=0.5. Fig. 4-7 displays the response for a sudden change in dimensionless inlet air temperature for L/G= 0.5. The change in L/G from 2.0 to 0.5 does not significantly affect the system response.

Figure 4-1. Dynamic response of evaporator exit water and air temperatures for different liquid hold-up.
Figure 4-2. Dynamic response of evaporator exit water and air temperatures for \( a_w/a = 0.5 \) and \( a_w/a = 1.0 \).

Figure 4-3. Dynamic response Evaporator exit water and air temperature for different packed bed material.
Figure 4-4. Evaporator exit water and air temperature responses to a step change in the inlet water temperature, L/G=2.0.

Figure 4-5. Evaporator exit water and air temperature responses due to a step change in the inlet air temperature, L/G=2.0.
Figure 4-6. Evaporator exit water and air temperature responses due to a step change in the inlet water temperature, L/G=0.5.

Figure 4-7. Evaporator exit water and air temperature responses due to a step change in the inlet air temperature, L/G=0.5.
CHAPTER 5
SYSTEM PERFORMANCE

Water Production

In order to evaluate the economic viability of the solar DDD process, it is necessary to evaluate the potential water production for a given solar collector area and power required to yield such production. Fig. 5-1 shows a schematic diagram of the solar DDD. Here different operating conditions will be considered, and the associated fresh water production and required power will be examined.

Figure 5-1. Schematic diagram of the solar DDD
Fresh water production for the DDD process is evaluated as

\[ \dot{m}_{fw} = \dot{m}_a (\omega_{in} - \omega_{out}) , \]  

(5-1)

where \( m_{fw} \) is the fresh water produced in the condenser, \( m_a \) is the air flow rate, \( \omega_{in} \) and \( \omega_{out} \) are the humidity ratios of air entering and exiting the condenser, respectively. The distillation efficiency is defined as

\[ \eta_{fw} = \frac{\dot{m}_{fw}}{\dot{m}_{L, evap}} , \]  

(5-2)

where \( m_{L, evap} \) is the water mass flow rate in the evaporator.

**Energy Consumption**

The energy required to operate the solar DDD consists of both thermal and electrical energy. The thermal energy is obtained from the solar collector. The electrical energy is required to circulate the working fluids (air and water) through the DDD facility. The electrical energy consumption is the pumping power required by the fluid to overcome frictional and gravitational forces. The electrical energy consumption rate needed to pump the water to the top of the tower is computed as,

\[ P_{W_L} = \dot{m}_L g H = \frac{LA}{\rho L} \Delta P_L . \]  

(5-3)

\( \Delta P_L \) is equivalent to the gravitational head and is given by,

\[ \Delta P_L = \rho_L g H . \]  

(5-4)

To circulate the air, the pumping power required should overcome the pressure drop in the diffusion tower and the condenser, which is due to frictional forces between the air and the packing. A correlation provided by the manufacturer is used to predict the
pressure drop within the packing. This correlation has been experimentally validated and is given by,

\[ \frac{\Delta P_G}{H} = \frac{G^2}{\rho_G} [0.0354 + 654.48 \left( \frac{L}{\rho_L} \right)^2 + 1.176 \times 10^7 \left( \frac{L}{\rho_L} \right)^4 \left( \frac{G}{\rho_G} \right)^4], \]

(5-5)

where \( \rho_G (\text{kg/m}^3) \) is the air density, \( \rho_L (\text{kg/m}^3) \) is the water liquid density, and \( H (\text{m}) \) is the height of the packed bed. Predicted and experimental measurements of the pressure drop are shown in Fig. 5-2.

![Figure 5-2. Air pressure drop in the packed bed for different water and air fluxes.](image)

The energy consumption required for pumping the air through the facility is calculated from,

\[ P_{W_G} = V_G \Delta P_G = \frac{\dot{m}_G (1 + \omega_m)}{\rho_G} \Delta P_G = \frac{GA}{\rho_G} \Delta P_G, \]

(5-6)

where \( V_G \) is the gas volume flow rate, and \( \Delta P_G \) is given by Eq. (4-3). The total electrical energy consumption is computed as,
The specific energy consumption is, 

\[ P \equiv P_{\text{el}} + P_{\text{m}} \]  \hspace{1cm} (5-7)

The specific energy consumption is an important parameter that represents the energy consumed per unit of water produced. It can be shown in Fig. 5-2 that increasing the air flux increases the pressure drop in the packed bed, which increases the electrical energy consumption. The optimum operating condition of the DDD process suggested by Li et. al. [37] is used in this study, which is 1.5 kg/m\(^2\)-s air flux, water mass flux of 1.5, and 3 kg/m\(^2\)-s in the evaporator and condenser, respectively.

To determine water production performance, consider the solar DDD system operating in the standard configuration and operating condition as shown in Table 5-1. Fig. 5-3 shows the fresh water production rate for the given solar heat input where the condenser is operating with continuous cooling so as to maintain condenser inlet water temperature at 25\(^\circ\)C. It is clear that as the solar heat input increases the evaporator inlet water temperature increases, which improve the water production. Fig. 5-4 shows the increase in the total fresh water produced and the decline of the saline water volume in the storage tank with time. Fig. 5-5 shows the fresh water production efficiency increases with increasing the evaporator inlet water temperature. For the given system, the fresh water production rate per solar collector area is on the order of 6.5 liters per square meter of solar collector area per day, or 65 L/day. It should be emphasized that this fresh production is specific for a system with the given evaporator and condenser dimensions, solar collector area, and operating conditions shown in Table 5-1. Fig. 5-6 shows the energy consumption rate and the specific energy consumption for the
It is observed that the energy consumption rate is approximately constant for the system. However, the specific energy consumption varies significantly at the initial and final operating time due to the low water production rate with low heat input. In the midrange of operating time, the specific energy consumption is on the order of 2 kWh/m³.

Table 5-1. Standard operating conditions and configuration for the solar DDD

<table>
<thead>
<tr>
<th>System Parameter</th>
<th>Condition</th>
</tr>
</thead>
<tbody>
<tr>
<td>$L_{\text{evap}}$</td>
<td>1.5 kg/m²-s</td>
</tr>
<tr>
<td>$G$</td>
<td>1.5 kg/m²-s</td>
</tr>
<tr>
<td>$L_{\text{cond}}$</td>
<td>3.0 kg/m²-s</td>
</tr>
<tr>
<td>$A_{\text{evap}}$</td>
<td>0.0456 m²</td>
</tr>
<tr>
<td>$H_{\text{evap}}$</td>
<td>1.0 m</td>
</tr>
<tr>
<td>$A_{\text{cond}}$</td>
<td>0.0456 m²</td>
</tr>
<tr>
<td>$H_{\text{cond}}$</td>
<td>1.0 m</td>
</tr>
<tr>
<td>$V_{\text{sw}}(t=0)$</td>
<td>100 L</td>
</tr>
<tr>
<td>$V_{\text{fw}}(t=0)$</td>
<td>300 L</td>
</tr>
<tr>
<td>$T_{\text{sw}}(t=0)$</td>
<td>25 °C</td>
</tr>
<tr>
<td>$T_{\text{fw}}(t=0)$</td>
<td>25 °C</td>
</tr>
<tr>
<td>$A_{\text{solar}}$</td>
<td>10 m²</td>
</tr>
<tr>
<td>$\eta_{\text{solar}}$</td>
<td>100%</td>
</tr>
</tbody>
</table>

Figure 5-3. Solar heat input and fresh water production rate.
Figure 5-4. Total fresh water produced and saline water volume in the saline water storage tank.

Figure 5-5. Variation in fresh water production efficiency a) with respect to temperature, b) with respect to time
Figure 5-6. Energy consumption and the specific energy consumption for the solar DDD.
Fig. 5-7 shows the total fresh water produced for a system operating in the same standard configuration and operating condition shown in Table 5-1, except that the saline water tank size is varied. It is clear that the tank size does not significantly change fresh water production. Modest improvement is achieved by reducing the size of the saline water storage tank. The saline tank size can be important in some cases, such as if the condenser inlet water is not cooled which will be shown in the following sections.

Fig. 5-8 shows the total fresh water produced for a system operating in the same standard configuration and operating condition shown in Table 5-1, except that the solar collector efficiency is varied. It is clear that the solar collector efficiency influences fresh water production significantly.

Figure 5-7. Total fresh water produced for various saline water storage tank sizes.
The effect of condenser inlet water temperature on the water production performance is examined in Figs. 5-9, and 5-10. The computational results shown in Figs. 5-9, 5-10 and 5-11 are for a system operating in the standard configuration and operating conditions shown in Table 5-1, except that the condenser inlet water temperature is varied. It is interesting to note that decreasing the condenser inlet water temperature reduces the system water production efficiency. This is due to the fact that reducing the condenser inlet water temperature results in reducing the condenser exit air temperature. Since the air is re-circulated back to the evaporator, the evaporator air and water temperature reduction inhibits vapor production. Fig. 5-11 shows that reductions in the condenser water temperature results in reducing the saline water
temperature in the saline water storage tank. This conclusion holds when there is air re-circulation in the system. When there is no air re-circulation, the effect of condenser inlet water temperature on the water production performance is examined in Fig. 5-12. This computational result shown in Fig. 5-12 is for a system operating in the standard configuration and operating conditions shown in Table 5-1, except that the condenser inlet air temperature is maintained constant at 45°C, and the condenser inlet water temperature is varied. It is clear that decreasing the condenser inlet water temperature decreases the condenser exit air temperature which improves the fresh water production efficiency.

Figure 5-9. Fresh water production rate for different condenser inlet water temperatures.
Figure 5-10. Total fresh water production rate for different condenser inlet water temperatures.

Figure 5-11. Saline water tank temperature for different condenser inlet water temperatures.
Figure 5-12. Fresh water production efficiency and condenser exit air temperature for various condenser inlet water temperatures.

Cooling System

In the previous analysis, the condenser inlet fresh water temperature was assumed to be constant. In real applications, the condenser temperature is variable, and cooling is required to keep it low. For instance, the system described previously can be equipped with a heat exchanger that uses water from the sea as a cooling fluid as shown in Fig. 5-1. Assuming that $T_{sw,hx,in}$ is 25°C, $m_{sw,hx,in}$ is 0.18 kg/s, and the water to water counter–current heat exchanger effectiveness is 0.75. The condenser inlet water temperature (heat exchanger outlet water temperature) and the heat removed from the system can be determined as shown in Fig. 5-13. It can be shown that the heat removed to reduce the condenser water temperature is very close to the solar heat input.
Figure 5-13. Condenser water temperature and heat removed

Figure 5-14. Fresh water production rate and total fresh water produced.
It should be noted the difference between the solar heat input and the heat removed is due to the increased thermal energy stored in the saline and fresh water tanks. The water production rate and the total water produced are shown in Fig. 5-14. The computational results shown in Figs. 5-13 and 5-14 are for a system operating in the standard configuration and operating conditions shown in Table 5-1, except that the condenser inlet water temperature is cooled by the heat exchanger described above. The condenser inlet water temperature varies from 25°C to 37°C for the current configuration. It is clear that the water production rate for the current configuration is higher than that with 25°C condenser inlet water temperature.

**Strategy to Improve System Performance**

**No Cooling until the Heat Input is Shut Off**

It is of interest to examine how the solar DDD behaves at elevated temperatures. For a system operating in the standard configuration shown in Table 5-1, except that $A_{\text{solar}}$ is 2 m$^2$ and the condenser inlet water is not cooled until the heat input is shut off. The system essentially operates adiabatically; heat into the system is recycled. Under this operating condition, the circulating air temperature is elevated, which improves the evaporation process. However, the amount of condensation is reduced due to the increase in the condenser water temperature. This fresh water temperature rise is shown in Fig. 5-15, where the tank fresh water temperature increases until the heat input is shut off; thereafter, the condenser water is cooled. The total fresh water production rate and the total water produced per solar collector area for the given operating condition are shown in Fig. 5-16. Once the time is 12 hr, the heat input is shut off and cooling is enabled; fresh water production rate increases significantly due to the sudden decrease in the condenser inlet water temperature. After the heat input is
shut off, the circulated air temperature in the system decreases gradually which results in decreasing the water production rate. When there is no heating (the operation after 12hr shown in Fig.5-15), evaporation occurs mainly due to the thermal energy stored in the saline water storage tank. The affect of the saline water tank size on the water production with this operating condition can be examined in Figs. 5-17, 5-18 and 5-19. The results shown in Figs. 5-17, 5-18 and 5-19 are for a system operating in the standard configuration shown in Table 5-1, except that $A_{\text{solar}}$ is 2 m$^2$ and the saline water tank size is varied. It can be shown that smaller tank volumes result in higher water temperature in the saline water storage tank, as shown in Fig. 5-17; thus, higher water production can be achieved in the no cooling mode as shown in Fig. 5-18. However, for a smaller tank, the production drops much faster in the cooling mode. In general the total water produced is higher for a smaller tank size as shown in Fig. 5-19.

![Figure 5-15. Fresh water temperature and solar heat input.](image-url)
Figure 5-16. Fresh water production rates and total fresh water produced.

Figure 5-17. Saline water tank temperature history
Figure 5-18. Fresh water production rates for various saline water storage tank sizes

Figure 5-19. Total fresh water produced for various saline water storage tank sizes
Similar results are also obtained for different fresh water tank sizes shown in Figs. 5-20 and 5-21. The results shown in Figs. 5-20 and 5-21 are for a system operating in the standard configuration shown in Table 5-1, except that $A_{\text{Solar}}$ is 2 m$^2$ and the fresh water tank size is varied. The water temperature in the evaporator and condenser increases by reducing the fresh water tank size which results in improving the total water production. Fig. 5-22 shows the specific electrical energy consumption for a system operating in this mode and in the standard configuration shown in Table 5-1, except that $A_{\text{Solar}}$ is 2 m$^2$. When the system is operating in the no cooling mode, the specific energy consumption profile is similar to that shown previously in Fig. 5-6. Once the cooling is started, the specific energy consumption decreases significantly due to the sudden increase in the water production rate. As the water production decreases, due to the reduction in the evaporator inlet water temperature, the specific energy consumption increases.

![Figure 5-20. Temperature in the system for various fresh water storage tank sizes](image-url)

Figure 5-20. Temperature in the system for various fresh water storage tank sizes
Figure 5-21. Total fresh water produced for various fresh water storage tank sizes

Figure 5-22. Specific electrical energy consumption
An experimental investigation has been carried out to examine the solar DDD performance when operated with this mode. Fig. 5-23 and 5-24 show the computed and measured water and air temperatures in the evaporator and condenser, respectively. The results shown are for a system operating with the standard configuration shown in Table 5-1, except the evaporator water mass flux is 1 kg/m²-s, condenser water mass flux is 2 kg/m²-s, air mass flux is 1 kg/m²-s, and $V_{sw}(t=0)$ is 141L. As shown in Fig. 5-23 and 5-24, the computed exit water and air temperatures are in a satisfactory agreement with those measured in the evaporator and condenser. This indicates that this operating mode can be employed in real application to improve the water production of the solar DDD.

Figure 5-23. Evaporator water and air temperatures
Delay Operating Time

It is evident in the previous analysis that higher evaporator inlet water temperature results in higher water production. The option proposed here is to circulate the saline water between saline water tank and solar collector only without admitting it to the evaporator until the storage tank reaches 50°C; then the water is admitted to the evaporator and air is circulated through the system. Fig. 5-25 shows the evaporator inlet water and tank saline water temperatures, and the water production rate. As shown, the water production rate is improved significantly by increasing the evaporator inlet water temperature. This method can be readily implemented by using a thermostat to control saline water flow. Fig. 5-26 shows the total water produced, and the system efficiency using the delayed operating time method, and Fig. 5-27 shows the energy
consumption. At the initial time when the water is only circulated between the saline water tank and solar collector, there is no water production. It is evident that the system efficiency increases at an elevated water temperature. For the delayed operating time method, the water production is slightly higher than that without delaying, and the energy consumption is lower. The results shown in Figs. 5-25, 5-26 and 5-27 are for a system with the standard configuration shown in Table 5-1, 10 m² solar collector area, and operating with delayed operating time mode.

It is of interest to examine the influence of the saline water storage tank size on this operating mode. Figs. 5-28 and 5-29 show the total water produced and the electrical energy consumption for various saline water storage tank sizes. The results shown in Figs. 5-28 and 5-29 are for a system with the standard configuration shown in Table 5-1, except that the saline water tank size is varied and operating with delayed operating time mode. It can be shown that increasing the tank size does not improve the water production as shown in Fig. 5-28. However, the energy consumption reduces for larger tank size by reducing the running time of the system. For a system operating with delayed operating time mode, the specific energy consumptions are 1.97, 1.88 and 1.65 kWh/m³ for 100L, 200L, and 400L saline water tank, respectively. It is shown that increasing the saline water tank from 100 L to 400 L reduces the specific energy consumption by 16%. Furthermore, reducing the operating time reduces system maintenance. Therefore, the specific energy consumption is significantly reduced by increasing the tank size.
Figure 5-25. Water temperatures and water production rates with delayed operating time

Figure 5-26. Fresh water production and fresh water production efficiency with delayed operating time
Figure 5-27. Energy consumption with delayed operating time

Figure 5-28. Fresh Water Production with delayed operating time for various tank sizes
Water Salinity

Scaling is a problem that faces most desalination technologies and industrial processes. In desalination plants, the composition of scale depends on the feed seawater and the operating conditions. Magnesium hydroxide, calcium sulfate and calcium carbonate are most common scale compounds that are deposited on solid surfaces of desalination components. The formation of scale occurs when the solubility limits of the scaling compounds are exceeded. Scale prevention must be a consideration in the design of desalination systems. Scale can be prevented by operating the system below the solubility threshold.

The typical solubility limit of calcium sulfate and calcium carbonate are on the order of 1500 ppm, and 14 ppm, respectively. The solubility limit of calcium carbonate
is low, and since the solubility limit decreases with temperature, calcium carbonate precipitates out of solution when seawater is heated, even when no evaporation occurs. Calcium carbonate deposition occurs at relatively low temperature (approximately 60°C). The calcium ion present in seawater has a concentration of about 400 ppm, and accounts for approximately 1.2% of the total salt content by mass. Magnesium is present in seawater at about 1300 ppm, and accounts for about 3.66% of the total salt content by mass.

The water salinity in the saline water storage tank varies during the desalination process due to the variation in the saline water volume. Water salinity is expressed as

\[
C = \frac{m_{\text{salt}}}{M_{\text{sw}}},
\]

(5-9)

where \(m_{\text{salt}}\) is the salt mass in the saline water, and \(M_{\text{sw}}\) is the saline water mass. The concentration factor is defined as

\[
CF = \frac{V_{\text{sw}}(t = 0)}{V_{\text{sw}}(t)},
\]

(5-9)

where \(V_{\text{sw}}\) is the saline water volume in the tank. For thermal desalination technologies, operating the distillation processes at high temperature and concentration factor results in maximum fresh water production. Such operation also results in decreasing the solubility limit and increasing the salt concentration which is favorable for scale precipitation. Fig. 5-30 shows the variation in the salinity along with the concentration factor for the solar DDD system operating in the standard configuration shown in Table 5-1, except that \(V_{\text{sw}}\) is 400L, \(T_{\text{L,cond,in}}\) is 25°C, and the initial salinity is 35,000 ppm. As shown, the salinity and the concentration factor increase due to the decrease in the saline water volume as evaporation occurs. At the final state, water salinity is the
highest and it is of interest to examine whether scale precipitation should occur.

Considering the final condition of the saline water shown in Fig. 5-27, where CF is approximately 1.2, the calcium ion concentration is 480 ppm, and magnesium ion concentration is 1560 ppm. Assuming that the final temperature is 70°C, the alkalinity is 2 milliequivalents per liter, and the final pH value is 8.0; under these concentration conditions no calcium sulfate is expected to precipitate [43]. The concentration of calcium carbonate is 21 ppm, which is higher than the solubility limit, thus it is expected to precipitate out. The solubility limit for magnesium hydroxide under the given conditions is approximately 7210 ppm; thus no magnesium hydroxide is expected to precipitate out.

![Graph showing water salinity and CF over time](image)

Figure 5-30. Seawater salinity in the solar DDD

Calcium carbonate solubility can be greatly increased by adding acid to the concentrated seawater alkalinity. Thus calcium carbonate and magnesium hydroxide
can be prevented by controlling the seawater pH using acid for neutralization. For the system described above, reducing the pH value to 7.0 increases the solubility of calcium carbonate to approximately 170 ppm; thus no calcium carbonate would precipitate out. For 400 L saline water tank, adding 3.7 mL muriatic acid of 2 pH is enough to reduce the pH of the saline water from 8.2 to 7. Nevertheless, the DDD operating temperature and concentration factor of the DDD is less than that of other thermal distillation technologies; thus, it is easier to prevent scaling.

Conclusions

A study has been carried out to examine the transient characteristic behavior of the solar DDD. The performance of the system has been examined under different operating conditions and conclusions are (1) increasing the evaporator inlet water temperature improve the solar DDD performance; (2) re-circulating the air exiting the condenser, which is usually at higher temperature than the ambient air, back to evaporator improves the solar DDD performance. Higher water and air temperature in the evaporator increases evaporation which results in more water production; (3) water production is dependent on the solar heat input. Higher solar collector efficiency results in higher heat input and higher water temperature which improve water production; (4) modest improvement is achieved by reducing the saline water storage tank size when the condenser inlet water is held constant; (5) when operating in the re-circulation mode, reducing the condenser water temperature does not necessarily improve the water production. Even though reducing the condenser water temperature improves water vapor condensation, it reduces the system performance by decreasing the condenser exit air temperature; this results in reducing the evaporator inlet air temperature; (6) if the system is operated in a manner such that an ambient air is used
as an inlet to the evaporator instead of re-circulating the condenser exit air, reducing the condenser inlet water temperature maximizes the water production. This does not hold for re-circulated air; (7) several options have been presented to improve the solar DDD performance such as: (a) cooling the condenser water only after the heat input is shut off, and (b) delay the operating time until the tank saline water temperature is higher than 50°C; (8) when the solar DDD is operating with no cooling until the heat input is shut off, a smaller saline water tank volume results in higher saline water tank temperature; thus, higher water production can be achieved. Once the heat is shut off and cooling is enabled, the water production rate drops faster for smaller saline water tank due to the fast reduction in the water temperature. Similarly, when the system is operating with no cooling until the heat input is shut off, a smaller fresh water tank volume results in higher water production rates; (9) if no cooling is available, the water production can improved reducing the saline and fresh water tank sizes; (10) in the delayed operating time mode, the saline water is circulated between the solar collector and saline water tank only without admitting it to the evaporator until the tank saline water temperature reaches 50°C. This enables running the system only when the water production rate is high which reduces the specific energy consumption. In this mode, increasing the saline water tank size enables more thermal energy storage; thus, the run time of the system is reduced along with the specific energy consumption. Using this operating method, the specific energy consumption is reduced by 16% when the saline water tank size increased from 100 L to 400 L; (11) fresh water production rate can be maximized by operating the system at relatively higher temperature which can be accomplished by reducing the saline and fresh water tank sizes and maintaining the
condenser inlet water temperature between 30-40°C. Water production is also maximized by running the system for longer time. This however increases the specific energy consumption; (12) the optimum operating condition for the solar DDD is the condition at which the maximum water production rate and minimum specific energy consumption are achieved. The specific energy consumption can be minimized by running the system with delayed operating time and large saline water tank size. Since the total water produced by the delayed operating time method is nearly constant, the specific energy consumption can be reduced significantly by using large saline water tank. This also reduces the running time and maintenance. It is believed that solar DDD is best operated with the delayed operating time method; (13) scaling potential for the solar DDD has been examined. Since the solar DDD operates at lower temperature and concentration factor compared to other desalination processes, scaling is not a major challenge and can be readily prevented.
ECONOMIC ANALYSIS

CHAPTER 6

Economic Overview of Desalination Technologies

This chapter is aimed at investigating whether solar powered seawater desalination is cost-effective. The cost of water desalination has decreased in the last few decades due to improvements in the desalination technologies. Water production cost depends primarily on the specific energy consumption, energy cost, seawater salinity, system capacity, and plant life. The energy consumption is the most deterministic factor for selecting a desalination technology because the water production cost is strongly affected by the energy consumption. In the previous chapter, it was shown that the solar DDD specific energy consumption varies depending on the amount of water produced. The average specific energy consumption of the solar DDD is 2.1 kWh/m$^3$. The specific electrical energy consumption for RO ranges from 5-10 kWh/m$^3$ for brackish water, and 10-20 kWh/m$^3$ for seawater [44]. This is due to the fact that the osmotic pressure of seawater is significantly higher than that of brackish water; thus seawater reverse osmosis (SWRO) requires more energy than brackish reverse osmosis (BWRO). Seawater is more costly to desalinate than brackish water. Seawater contributes to more than 58% of the feed water used for desalination, followed by brackish ground water with 23%; and the rest is obtained from lakes, rivers and other sources.

Most of the large desalination plants are fueled by oil or gas; thus, the rising oil and gas prices affect the desalination water cost. The cost of electricity generated from fossil fuels has increased due to increased oil prices. In the U.S., the price for industrial electric energy price is $0.067/kWh [45]. The interest in desalination technologies,
especially those that utilize renewable energy, have increased in the last few decades. Small and medium-sized desalination plants can run partly or even entirely on renewable energies. Renewable energy utilization for desalination has strong potential; however, the exploitation is modest due to economic challenges. In the last few decades, significant progress has been made to reduce the cost of renewable energy utilization systems. A proper matching between the desalination technologies and renewable energy systems is essential to achieve a reliable supply of water at a low cost.

Solar Photovoltaics (PV) is a fast developing technology that presents an excellent choice for providing electrical energy in remote areas and isolated islands. Photovoltaics can be used independently or in conjunction with the grid to power desalination systems. The efficiency of PV panels has improved over the last few decades to approximately 15%. The average cost per installed watt for PV panel systems is about $7.50 to $9.50, including panels, inverters, mounts, and electrical items [46]. PV panels are well suited for low energy consumption desalination systems.

The coupling of solar PV with low energy consumption desalination system for small scale applications is an excellent solution for remote and arid areas. Photovoltaic powered reverse osmosis (PV-RO) is considered one of the most promising desalination technologies. The use of PV to drive water desalination is not new. Currently, there are many installations of PV-RO worldwide which are mostly of a stand alone type [44], [47], and [48]. It is well suited for brackish water. For seawater desalination, the energy consumption of RO is high compared to brackish water, and
small scale RO desalination systems are usually not equipped with energy recovery devices; thus, it is characterized by high energy consumption. Furthermore, photovoltaic seawater powered reverse osmosis (PV-SWRO) requires solar batteries to store the energy generated by the PV panels so that it can achieve high feed pressure for the system. The battery is costly and has short life span when operated in hot climates. If no battery is used, fluctuations in the solar insolation results in variable feed pressure which can lower the lifetime of the membranes. Economic assessment for different PV-driven desalination systems have been carried out and it has been reported that the cost of seawater PV-RO desalination is $7.98-29/m³ for small-scale systems [44], and [49-55].

The use of parabolic solar trough collectors for seawater desalination has been described by Kalogirou [56]. The study revealed that the use of solar troughs for medium and large scale seawater desalination systems is not cost effective to operate due to the relatively high cost of the equipment. Generally, for large scale applications, solar energy powered desalination plants are not competitive with fossil fuel powered desalination plants at the current time due to the high cost of harvesting solar energy. The price of harvesting solar energy has dropped in the last few decades, and it will continue to drop further in the future. With continuous decrease in solar energy prices, solar driven desalination systems will become a cost-effective choice as a source of fresh water in many arid and coastal regions.

Al-Hallaj et al. [57] performed an economic analysis of small scale solar HDH units. It is concluded from the study that the investment costs are highest, but energy costs are lowest in the case of renewable energy powered desalination systems.
Estimating fresh water costs from solar powered desalination systems is difficult due to the limited economic data available for the components. In the following section, an economic analysis of a small scale solar DDD plant is carried out.

**Small Scale Solar DDD and PV-RO Desalination System**

A comparison is made between the solar DDD and PV-RO systems with a 100 L/day plant capacity. To evaluate the water production cost for a small scale desalination unit, Table 6-1 shows component prices for the solar DDD unit. The PV panel size required to power the two different desalination plants is estimated by accounting for the energy consumption required for the two plants. The energy consumption of the solar DDD varies during time of operation, thus an average value of 2.1 kWh/m$^3$ is used. The energy consumption of a small scale RO plant is approximately 15 kWh/m$^3$. It is assumed that 1 kW$_p$ panel module produces an average of 4 kWh/day. The price of the PV panel used is $6.5/W_p$, where $W_p$ is the peak value of the electrical power produced by the PV panel. The PV panel costs for the solar DDD plant and the PV-RO are $289 and $2350, respectively. It is clear that the price of the PV panel required for a small scale solar DDD is significantly lower than that of PV-RO.

<table>
<thead>
<tr>
<th>Solar DDD Components for 0.1 m$^3$/day</th>
<th>Cost (U.S.$)</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>U.S.</strong></td>
<td><strong>China</strong></td>
</tr>
<tr>
<td>Flat plate solar collectors (FPSC)</td>
<td>2000</td>
</tr>
<tr>
<td>PV panel</td>
<td>289</td>
</tr>
<tr>
<td>Water tank</td>
<td>100</td>
</tr>
<tr>
<td>Nozzle</td>
<td>40</td>
</tr>
<tr>
<td>Air blower</td>
<td>100</td>
</tr>
<tr>
<td>Pumps</td>
<td>220</td>
</tr>
<tr>
<td>Packed bed</td>
<td>80</td>
</tr>
<tr>
<td>DDD structure</td>
<td>200</td>
</tr>
<tr>
<td>Mist eliminator</td>
<td>50</td>
</tr>
<tr>
<td>Heat exchanger</td>
<td>310</td>
</tr>
<tr>
<td><strong>Total Cost</strong></td>
<td><strong>3389</strong></td>
</tr>
</tbody>
</table>
The annual interest rate of the desalination unit is given as,

\[ A_{\text{fixed}} = a(DC) = a(\text{Total Cost} + \text{Assembly Cost}) , \]  

(6-1)

where DC is the direct capital cost of the desalination system, the assembly cost of the solar DDD is approximated to be 5% of the total cost, and a is the amortization factor and is given as,

\[ a = \frac{i(1 + i)^n}{(1 + i)^n - 1} , \]  

(6-2)

where i is the interest rate, n is the plant life (year). The chemical costs needed for the system is given as,

\[ A_{\text{chemical}} = (C_2)(m_{fw})(365) , \]  

(6-3)

where \( C_2 \) is the chemical cost ($/m^3) which usually ranges between $0 -0.025/m^3 for PV-RO, and $0.02/m^3 for solar DDD. The electrical energy cost required to operate the system is given as,

\[ A_{\text{electricity}} = (C_3)(p_{\text{wtr}})(m_{fw})(365) , \]  

(6-4)

where \( C_3 \) is the electricity cost per kilowatt hour. Here, \( C_3 \) is $0.067/kWh. The maintenance cost required is given as,

\[ A_{\text{maintenance}} = (C_4)(m_{fw})(365) , \]  

(6-5)

where \( C_4 \) is the maintenance cost per m^3 of water produced, and $0.02/ m^3 of water produced is used for the solar DDD. The total cost is given as,

\[ A_{\text{total}} = A_{\text{fixed}} + A_{\text{chemical}} + A_{\text{electricity}} + A_{\text{maintenance}} , \]  

(6-6)

For a PV powered system, electricity is supplied entirely by the PV panels; thus \( A_{\text{electricity}} \) is neglected. The water product cost \( C_{\text{wtr}} \) is calculated as,
The water product costs for small scale solar DDD are $7.8/m³ and $3.8/m³ for the system built in the U.S. and China, respectively. It is clear that the solar collector cost contributes to more than 50% of the total solar DDD cost. In general, RO has low capital cost compared to the solar DDD; however, it has high maintenance cost due to membrane replacement cost. The water cost of PV-SWRO reported ranges from $7.98 to 29 per m³ of fresh water product for small-scale systems [44], and [49-55]. Comparing the price for each method, it is clear that the solar DDD unit produces water at a relatively lower cost. The flat plate collector panel cost is a limiting factor. However, due to its low energy consumption and maintenance requirement, the solar DDD is projected to be a cost-effective desalination technology that is well suited for arid and remote regions. If no PV panels are used, and the electrical energy is purchased from a municipality at a cost of $0.067/kWh, the water production cost for the solar DDD is $7.19/m³ of fresh water produced.

The DDD can be coupled with a waste heat recuperation system of any industrial plant. Consider a 100 MW thermal power plant where its thermal efficiency is 40%. If the waste heat from the power plant is used to heat the seawater in the DDD system, the water production capacity is expected to be 3402 m³/day. To estimate the water production cost, the flat plate solar collector cost is omitted and the cost of heat exchanger to recuperate the waste heat is added. The cost of heat the exchanger used is $76.7 per kilowatt heat recuperated. If municipality electricity is powering the DDD instead of PV-panels, the desalinated water cost with a waste heat recuperation system...
is $2.8 /m$^{3}$ of fresh water produced. Table 6-2 shows the water cost for the solar DDD and RO process using different configurations. Table 6-3 shows the municipal water cost in some countries around the world [59].

Table 6-2. Water cost for solar DDD and RO processes

<table>
<thead>
<tr>
<th>Operating Conditions</th>
<th>Solar DDD ($/m^{3}$)</th>
<th>RO ($/m^{3}$)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>U.S.</td>
<td>China</td>
</tr>
<tr>
<td>PV panel</td>
<td>7.8</td>
<td>3.81</td>
</tr>
<tr>
<td>Municipality electricity</td>
<td>7.3</td>
<td>3.7</td>
</tr>
<tr>
<td>Municipality electricity and waste heat</td>
<td>2.8</td>
<td>1.4</td>
</tr>
</tbody>
</table>

Table 6-3. Water cost in some countries around the world

<table>
<thead>
<tr>
<th>Country</th>
<th>Water Cost (U.S. $/m^{3}$)</th>
</tr>
</thead>
<tbody>
<tr>
<td>UAE</td>
<td>2.2</td>
</tr>
<tr>
<td>Germany</td>
<td>1.91</td>
</tr>
<tr>
<td>Saudi Arabia</td>
<td>1.5</td>
</tr>
<tr>
<td>France</td>
<td>1.23</td>
</tr>
<tr>
<td>Jordan</td>
<td>1.14</td>
</tr>
<tr>
<td>Italy</td>
<td>0.76</td>
</tr>
<tr>
<td>U.S.</td>
<td>0.51</td>
</tr>
<tr>
<td>Australia</td>
<td>0.5</td>
</tr>
<tr>
<td>South Africa</td>
<td>0.47</td>
</tr>
<tr>
<td>Canada</td>
<td>0.4</td>
</tr>
<tr>
<td>China</td>
<td>0.35</td>
</tr>
</tbody>
</table>
 CHAPTER 7
CONCLUSIONS

Drinking water in remote and arid areas is a scarce commodity. While desalination technologies offer an excellent solution to meet water demand by separating salt from saline water, they are energy intensive processes. In addition, while conventional desalination technologies are well suited for large scale applications, small scale desalination systems are associated with high energy consumptions and high water costs. Since the projection is continues increase in energy cost in the future, there is a need for cost effective desalination system that is energy efficient. In remote areas, where there is no access to energy such as electricity grid, the use of conventional desalination technologies is not possible. The interests in desalination technologies, especially those that utilize renewable energy, have increased in the last few decades. Although renewable energy is abundant in many areas around the world, its use for seawater desalination is very modest due to economic constraints.

Conventional desalination technologies require continues maintenance; and this is difficult when desalination processes is operated in remote areas.

The main goal of this study is to develop a cost effective small scale desalination unit that is powered by renewable energy. The desalination unit should operate with low energy consumption and low maintenance requirement, and suitable for small scale applications in remote and arid areas. The approach used is to utilize solar energy to heat saline water; then use the hot saline water to humidify an air stream. The humidified air stream is then dehumidified in a condenser to produce fresh water. The developed distillation technology is called the solar diffusion driven desalination. A small scale solar diffusion driven desalination facility has been developed. Theoretically
and experimentally investigation is carried out to examine the heat and mass transfer in
the solar DDD process.

While the solar DDD process is driven by solar heating, the operation is
inherently transient, and a transient heat and mass transfer analysis is required to
evaluate its operation over a range of operating conditions. A rigorous theoretical
model describing the transient heat and mass transfer during the evaporation and
condensation within backed beds have been developed based on conservation
principles. The developed model takes into account the transient variation of the solar
heat input, and the thermal capacity of the packed bed. The framework of the analysis
should find utility in a broad variety of applications.

A numerical scheme has been implemented using a finite difference method to
solve for the water and air temperatures and humidity ratio profiles in the evaporator
and the condenser. It is found that the transient model has a good prediction capability
in determining the evolution of the temperature profile within the evaporator and the
condenser. It is concluded that the time required to reach a steady state depends on
the water and air mass flow rates, liquid hold-up, and packing heat capacity. It is found
that higher liquid and air mass flow rates result in lower time required to reach the
steady state. This is expected since higher mass flow rates result in more effective
heat transfer between the working fluids and the packing, allowing the system to reach
steady state in a shorter time.

From the numerical study, it is found that the water, air, and packed bed
temperature profiles in the evaporator depends on the inlet water and air temperatures
and mass flow rates. In addition, it is found that the temperature of the packing is very
close to the water temperature. This agrees with the steady state model assumption in which a fluid film was assumed to exist at the packing and have the same temperature of the packing. While the numerically obtained temperatures and humidity ratios are reasonable, the models require experimental validation.

An experimental study was undertaken to investigate the efficacy of the evaporator and condenser theoretical models. A lab-scale solar DDD has been built and used to conduct experiments under different operating conditions. A time varying solar heat input over a 2.5 hours time span is considered in the experiments. Different evaporator and condenser water and air mass fluxes are examined. It is observed that the measured outlet water and air temperatures and the outlet humidity ratio agrees satisfactorily with those predicted using the transient analysis. The theoretical models can be utilized as a design tool to analyze the performance of desalination processes and other direct contact heat and mass transfer devises that utilize a packed bed.

A numerical study was carried out to investigate the thermal response of the in the evaporator when operating in the transient mode. A set of simulation runs were performed to study the theoretical model sensitivity to the liquid hold-up, packing wettability, and packing heat capacity. It is found that the response time for the exit liquid temperature is shorter for a low hold-up compared with higher hold-up. The steady-state exit temperatures are independent of the liquid hold-up. It is found that higher wettability of the packed bed has insignificant influence on the response time, but it does result in improved heat transfer with a higher exit air temperature and lower exit water temperature. In addition, it is found that the steady state temperatures do not depend on the heat capacity of the packed bed material.
The performance of the solar DDD, dictated by the fresh water production and energy consumption of system, is investigated. Fresh water production of the solar DDD depends primarily on the operating conditions, namely: water and air flow rates, and inlet water and air temperatures. From the study, it is concluded that increasing the evaporator inlet temperature improves fresh water production. The inlet water temperature depends upon operating conditions and system configuration such as the water and air flow rates, solar heat input, and saline water storage tank size. It is found that higher solar collector efficiency results in higher heat input and higher water temperature which improve water production. In addition, modest improvement is achieved by reducing the saline water storage tank size when the condenser inlet water is held constant. If no cooling is available, the condenser inlet water temperature increases; and with this operating condition, water production can be improved reducing the saline and fresh water tank sizes.

To increase the water temperature in the evaporator, the air exiting the condenser, which is usually at higher temperature than the ambient air, is re-circulated back to evaporator. Increasing water and air temperature in the evaporator increases water evaporation which improves the system performance. It is found that if the system is operated in a manner such that an ambient air is used as an inlet to the evaporator instead of re-circulating the condenser exit air, reducing the condenser inlet water temperature maximizes the water production. This does not hold for re-circulated air. When operating in the re-circulation mode, reducing the condenser water temperature does not necessarily improve the water production. Even though reducing the condenser water temperature improves water vapor condensation, it reduces the
system performance by decreasing the condenser exit air temperature; this results in reducing the evaporator inlet air temperature.

Several options have been presented to improve the solar DDD performance such as: (1) cooling the condenser water only after the heat input is shut off, and (2) delay the operating time until the tank saline water temperature is higher than 50°C.

When the solar DDD is operating with no cooling until the heat input is shut off, a smaller saline water tank volume results in higher saline water tank temperature; thus, higher water production can be achieved. Once the heat is shut off and cooling is enabled, the water production rate drops faster for smaller saline water tank due to the fast reduction in the water temperature. Similarly, when the system is operating with no cooling until the heat input is shut off, smaller fresh water tank volume results in higher water production rates.

In the delayed operating time mode, the saline water is circulated between the solar collector and saline water tank only without admitting it to the evaporator until the tank saline water temperature reaches 50°C. This enables running the system only when the water production rate is high which reduces the specific energy consumption. In this mode, increasing the saline water tank size enables more thermal energy storage; thus, the run time of the system is reduced along with the specific energy consumption. Using this operating method, the specific energy consumption is reduced by 16% when the saline water tank size increased from 100 L to 400 L.

Fresh water production rate can be maximized by operating the system at relatively higher temperature which can be accomplished by reducing the saline and fresh water tank sizes and maintaining the condenser inlet water temperature between
30-40°C. Water production is also maximized by running the system for longer time. This however increases the specific energy consumption.

The optimum operating condition for the solar DDD is the condition at which the maximum water production rate and minimum specific energy consumption are achieved. The specific energy consumption can be minimized by running the system with delayed operating time and large saline water tank size. Since the total water produced by the delayed operating time method is nearly constant, the specific energy consumption can be reduced significantly by using large saline water tank. This also reduces the running time and maintenance. It is believed that solar DDD is best operated with the delayed operating time method.

Scaling potential for the solar DDD have been examined. Since the solar DDD operate at lower temperature and concentration factor compared to other desalination processes, scaling is not a major challenge and can be readily prevented. An economic analysis for small scale solar DDD system is carried out. A comparison is made between the solar DDD and reverse osmosis process. It is concluded that the solar DDD production cost is lower than that of PV-RO system. It is believed that the solar DDD is an excellent technology for seawater desalination with low maintenance and low energy consumption. An interesting future work can focus on recuperating the latent heat of condensation using a heat pipe.

In conclusion, the energy consumption and product water quality are the most important deterministic factors of a desalination technology. The study reveals that solar DDD is competitive and well suited for arid areas where access to drinkable water is difficult. Even though RO desalination technology is growing very rapidly compared
with other desalination technologies, RO is associated with large amounts of chemical additives that can cause pollution. The energy consumption of thermal desalination plants is independent of salt concentration, while the energy requirements for the membrane desalination are highly dependent on the salt concentration. There is no best desalination technology. Desalination technology selection criterion should be examined carefully based on the site conditions, raw water used, and the energy sources available. The utilization of waste heat from power stations, solar power plants, and industrial plants to power the DDD process presents some advantages over other desalination technologies. Proper matching between the desalination technologies and renewable energy systems is essential to achieve a reliable supply of water at low cost.
APPENDIX A
ONDA’S CORRELATION

\[ k_L = 0.0051 \text{Re}_{Lw}^{2/3} \text{Sc}_L^{-0.5} (ad_p)^{0.4} \left[ \frac{\mu_L g}{\rho_L} \right]^{1/3} \]

\[ k_G = C \text{Re}_{GA}^{0.7} \text{Sc}_G^{1/3} (ad_p)^{-2} aD_G \]

\[ C = 2 \]

\[ \frac{a_w}{a} = \left\{ 1 - \exp \left[ -1.45 \left( \frac{\sigma_{c}}{\sigma_L} \right)^{0.75} \text{Re}_{Lw}^{0.5} \text{Fr}_{Lw}^{-0.05} \text{We}_{Lw}^{0.2} \right] \right\} \]

\[ \text{R } \xi_w = \frac{L}{a_w \mu_L}, \quad \text{R } \xi_d = \frac{G}{a_{\mu_G}}, \quad \text{R } \xi_d = \frac{L}{a_{\mu_L}} \]

\[ \text{Sc}_L = \frac{\mu_L}{\rho_L D_L}, \quad \text{Sc}_G = \frac{\mu_G}{\rho_G D_G}, \quad \text{Fr}_{Lw} = \frac{L^2 a}{\rho_L^2 g}, \quad \text{We}_{L} = \frac{L^2}{\rho_L \sigma_L a}. \]
APPENDIX B
STEADY STATE THEORETICAL MODEL OF THE DDD

Evaporator:

\[
\frac{dT_e}{dz} = \frac{G}{L} \frac{d\omega}{dz} \left( h_{F_g} - h_L \right) + \frac{Ua(T_L - T_a)}{Cp_L L}
\]

\[
\frac{dT_a}{dz} = -\frac{1}{1 + \omega} \frac{d\omega}{dz} \frac{h_L(T_a)}{Cp_a} + \frac{Ua(T_L - T_a)}{Cp_a G(1 + \omega)}
\]

\[
\frac{d\omega}{dz} = \frac{k_Ga_w}{G} \frac{M_y}{R} \left( \frac{P_{sat}(T_i)}{T_i} - \frac{\omega}{0.622 + \omega T_a} \right)
\]

Condenser:

\[
\frac{dT_a}{dz} = -\frac{1}{1 + \omega} \frac{d\omega}{dz} \frac{h_L(T_a)}{Cp_a} + \frac{Ua(T_L - T_a)}{Cp_a G(1 + \omega)}
\]

\[
\frac{dT_i}{dz} = \frac{G}{L} \frac{d\omega}{dz} \left( h_{F_g} - h_L \right) + \frac{Ua(T_L - T_a)}{Cp_L L}
\]

\[
\frac{d\omega}{dz} = \frac{dT_a}{dz} \frac{P}{P - P_{sat}(T_a)} \omega_a (b - 2cT_a + 3dT_a^2)
\]
LIST OF REFERENCES


BIOGRAPHICAL SKETCH

Fadi Alnaimat was born in Jordan. He received his bachelor's degree in mechanical engineering from the University of Jordan in December 2005. After that, he moved to the U.S. and started his master's degree in mechanical engineering in 2006 at the University of Texas-Pan American. He received his master's degree in December 2007. He received his Ph.D. in mechanical engineering from the University of Florida in May 2011.