A 10 KILOWATT-THERMAL INPUT MULTIPLE EFFECT DISTILLATION PILOT FOR CONCENTRATED SOLAR POWER AND DESALINATION OF SEAWATER PLANT

BY

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THESIS

Submitted in partial fulfillment of the requirements for the degree of Master of Science in Mechanical Engineering in the Graduate College of the University of Illinois at Urbana-Champaign, 2010

Urbana, Illinois

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ABSTRACT

The use of modular parallel plate falling film heat exchangers, optimization of thermal vapor compressor (TVC) entrainment ratio, and overall process thermal management increase the flexibility and overall efficiency of Multiple Effect Distillation (MED). A generic computational MED system model with a TVC and heat addition using classical compressible gas dynamic relationships is employed. An algorithm is presented which optimizes the performance ratio (PR) thru variation of the number of TVC and entrainment ratio. The result is minimization of exergetic losses at the lowest possible inlet pressure condition to the first MED stage, reducing required motive steam pressure. Capture of unused thermal losses from low temperature sources, to heat the TVC inlet motive steam, reduces the required motive steam temperature. The results of a parametric study confirm the Brayton power cycle analysis that the use of MED-TVC thermal harvest configuration for integration into a Concentrated Solar Power - Desalination of Seawater (CSP-DSW) dual-purpose plant improves the overall performance of both systems. Furthermore, the integration of a control system related to the inlet seawater temperature and fluctuations from the thermal heat input into the MED significantly increase the performance of the system by around 20%. Preliminary modeling of the MED-TVC with heat addition shows a decrease in overall thermal losses of the dual-purpose plant; the system generates more power and similar potable water for the same solar flux. The design for a 10 kWt input, single-stage and four-stage MED design is given for use in further characterization of the MED-TVC process characteristics and for integration into a proof of principle CSP-DSW pilot.
ACKNOWLEDGEMENTS

Thank you to my family for their encouragement and support of my passions for learning and science; especially to my mother for her pragmatic guidance. My sincerest appreciation to my advisor Prof. John Georgiadis, to Prof. Mark Shannon and Prof. Rafi Semiat who have guided me during my pursuit of my Master’s Degree, supported me through the trials and triumphs of research and for being my introduction into the field of water purification.
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<thead>
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<th>Description</th>
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<tbody>
<tr>
<td>P</td>
<td>Pressure</td>
</tr>
<tr>
<td>T</td>
<td>Temperature</td>
</tr>
<tr>
<td>ρ</td>
<td>Density</td>
</tr>
<tr>
<td>M</td>
<td>Mach Number</td>
</tr>
<tr>
<td>( m )</td>
<td>Mass Flow Rate</td>
</tr>
<tr>
<td>Q</td>
<td>Heat Transfer</td>
</tr>
<tr>
<td>q</td>
<td>Heat Transfer Rate</td>
</tr>
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<td>s</td>
<td>Entropy</td>
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<td>G</td>
<td>Gibbs Free Energy</td>
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<td>W</td>
<td>Work</td>
</tr>
<tr>
<td>a</td>
<td>Activity</td>
</tr>
<tr>
<td>h</td>
<td>Enthalpy</td>
</tr>
<tr>
<td>BPE</td>
<td>Boiling Point Elevation</td>
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<tr>
<td>PR</td>
<td>Performance Ratio</td>
</tr>
<tr>
<td>GOR</td>
<td>Gain Output Ratio</td>
</tr>
<tr>
<td>c</td>
<td>Concentration</td>
</tr>
<tr>
<td>k</td>
<td>Thermal Conductivity</td>
</tr>
<tr>
<td>η</td>
<td>Efficiency</td>
</tr>
<tr>
<td>v</td>
<td>Velocity</td>
</tr>
<tr>
<td>V</td>
<td>Volume</td>
</tr>
<tr>
<td>μ</td>
<td>Kinematic Viscosity</td>
</tr>
<tr>
<td>ν</td>
<td>Dynamic Viscosity</td>
</tr>
<tr>
<td>N</td>
<td>Number of Effects</td>
</tr>
<tr>
<td>x</td>
<td>Steam vapor Quality</td>
</tr>
<tr>
<td>E</td>
<td>Total Energy</td>
</tr>
<tr>
<td>Re</td>
<td>Reynolds Number</td>
</tr>
<tr>
<td>Pr</td>
<td>Prandlt Number</td>
</tr>
<tr>
<td>Nu</td>
<td>Nusselt Number</td>
</tr>
<tr>
<td>G</td>
<td>Steam mass velocity</td>
</tr>
<tr>
<td>ξ</td>
<td>Single phase frictional coefficient</td>
</tr>
<tr>
<td>α</td>
<td>heat transfer coefficient</td>
</tr>
</tbody>
</table>
dh Equivalent hydraulic diameter
m Molality
I Ionic strength

Subscripts
sin Steam Inlet
sout Steam Outlet
p Permeate
c Condensate
b Rejectate/Brine
sw Seawater
sat Saturation
ph Preheater
F Frictional
T Total
g Gravitational
acc Viscous
l Liquid
v Vapor
## NOMENCLATURE

<table>
<thead>
<tr>
<th>Abbreviation</th>
<th>Description</th>
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<tbody>
<tr>
<td>CSP-DSW</td>
<td>concentrated solar power - desalination of seawater</td>
</tr>
<tr>
<td>MED</td>
<td>Multi-Effect Distillation</td>
</tr>
<tr>
<td>MSF</td>
<td>Multistage Flash Distillation</td>
</tr>
<tr>
<td>PR</td>
<td>Performance Ratio</td>
</tr>
<tr>
<td>GOR</td>
<td>Gain output ratio</td>
</tr>
<tr>
<td>Q_p</td>
<td>Process Thermal Power</td>
</tr>
<tr>
<td>RO</td>
<td>Reverse Osmosis</td>
</tr>
<tr>
<td>TDS</td>
<td>Total Dissolved Salts</td>
</tr>
<tr>
<td>TVC</td>
<td>Thermal Vapor Compression</td>
</tr>
<tr>
<td>LiBr</td>
<td>Lithium Bromide</td>
</tr>
<tr>
<td>LMTD</td>
<td>Logarithmic Mean Temperature Difference</td>
</tr>
<tr>
<td>ppm</td>
<td>Parts Per Million</td>
</tr>
</tbody>
</table>
CHAPTER 1: Introduction

1.1 Water Stress

When coastal areas stressed by fresh water shortage and high population growth rate are in geographical zones receiving high solar flux, it is reasonable to seek methods to drive both desalination and power production by solar energy. Recent studies indicate that thermal desalination methods, such as Multiple Effect Distillation (MED), can be efficiently coupled to a closed power cycle driven by steam generated from direct solar thermal input; a concept known as Concentrated Solar Power –Desalination of Seawater (CSP-DSW).

Water resource management is of growing concern in the US and globally. Increasing inland fresh water salinity, costal population density, and fresh water scarcity are factors in the renewed interest in seawater desalination development. Saline water sources comprise of 97.5 percent of the global water resource but the majority of potable water comes from ground and surface water sources: less than 0.4 percent in the US and less than 1 percent globally is produced from seawater sources. However, many of the fresh water resources are being depleted or polluted causing the use of seawater desalination to grow rapidly where local fresh water can no longer support the water demand [1]. Not only does this water shortage impact human consumption and agriculture but also hydroelectric and thermoelectric power generation in developing and industrialized nations [2].

As fresh water and fossil fuel sources have been named as two of the top non-renewable resources and thermoelectric power production is intrinsically linked to water consumption, for cooling needs, a solution incorporating both needs would be advantageous. Furthermore, globally, coastal regions are more populated than inland areas and are growing at a faster rate. The original gridded population model (GPW) data set and the updated GPW2, from CIESIN and NASA Socioeconomic Data and Applications Center, locates 40 percent and 78 percent respectively of the global population as near-coastal, living within 50 km of a shoreline[3].., the GPW2 shows the average population density, within 100 km of the shoreline, is approximately three times higher than the global average density [4]. The trend is increased stress in population centers of both water and power resources.

1.2 Methodology

In this work, methods to reduce the required turbine outlet temperature and pressure and thereby increase the overall efficiency of a CSP-DSW plant, with MED desalination system, are discussed. Chapter 2 describes in detail the processes being considered for desalination: MED,
multiple stage flash distillation (MSF) and reverse osmosis (RO). Previous work on process flows, advantages and disadvantages, and energy and costs are reviewed for the potential integration of the system into a CSP-DSW plant. The operation of a gas powered cycle, Brayton Cycle, is discussed thermodynamically to understand possible CSP-DSW integration methods. The modeling scheme employed for the MED system, detailed in Chapter 3, is based on a continuity approach paired with gas dynamics and viscous two-phase flow modeling. Two optimization algorithms are used. The first minimizes the compressor outlet pressure with respect to the latent heat transferred to the first stage by varying the thermal vapor compressor (TVC) entrainment ratio and the number of compressors. The second maximizes the water produced, permeate, in the MED using a simple non-linear convergence of the final condenser temperature, seawater flow rate and. These are bounded with global constraints on the maximum system salinity and boiling point elevation.

A parametric study of MED operating conditions is investigated to understand the relationship between heat transfer area, one important capitol cost in an MED plant, and performance is outlined and discussed in Chapter 4. The impact of the inlet seawater temperature to final stage temperature, and the number of stages (specifically the temperature difference between stages) are the common design parameters with the most significant influence the system performance and overall size. The MED-TVC system performance is also sensitive to perturbation of the motive steam inlet conditions performing.

Further system efficiency can be gained by decreasing the heat transfer resistance within the heat exchangers. Parallel plate falling film heat exchangers have shown to have high heat transfer coefficients and a flexible modular design; however, the characterization of these is mostly proprietary. Chapter 5 outlines the design and operating conditions of two experimental apparatus for future work. These will used to investigate the performance of a parallel plate falling film heat exchanger in a MED-TVC system. The first is a one-stage clear acrylic MED to characterize the performance of the parallel plate falling film heat exchanger the second is a four-stage 10 kWt heat input unit to demonstrate proof of operation of the CSP-DSW of principal. With respect to the desalination the four-stage unit will be used to validate integration of the low pressure TVC, an intake seawater controller design and low temperature thermal harvesting for overall improved system thermodynamic efficiency.
CHAPTER 2: Background

2.1 Desalination Methods

The DLR AQUA-SOL report [5] provides an extensive comparison of seawater desalination practices and selected reverse osmosis (RO) and Multiple Effect Distillation (MED) as being most flexible and favorable for water production. Further review of seawater desalination methods shows the advantage of using a thermal method versus RO is that thermal systems can utilize low-grade harvested heat and higher potential recovery ratios.

Table 2.1 Desalination methods by salt separation method and energy source

<table>
<thead>
<tr>
<th>Separation Method</th>
<th>Energy Type</th>
<th>Process</th>
<th>Desalination Method</th>
</tr>
</thead>
<tbody>
<tr>
<td>Water from Salts</td>
<td>Thermal</td>
<td>Evaporation</td>
<td>Multi-Stage Flash (MSF)</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>Multi-Effect Distillation (MED)</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>Solar Distillation</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Crystallization</td>
<td>Freezing</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>Gas Hydrate Processes</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Filtration/Evaporation</td>
<td>Membrane Distillation</td>
</tr>
<tr>
<td></td>
<td>Thermo-Mechanical</td>
<td>Filtration</td>
<td>Reverse Osmosis (RO)</td>
</tr>
<tr>
<td>Salts from water</td>
<td>Electrical</td>
<td>Selective Filtration</td>
<td>Electro Dialysis</td>
</tr>
<tr>
<td></td>
<td>Chemical</td>
<td>Exchange</td>
<td>Ion Exchange</td>
</tr>
</tbody>
</table>

2.1.1 Thermal Desalination

Multiple Effect Distillation (MED)

The MED process consists of several consecutive chambers, called stages or effects, maintained at decreasing levels of pressure and temperature. The operating temperature range of the hottest first stage to the coldest final stage is typically 70°C to 35°C. The system operates at sub-atmospheric pressures and the most efficient MED processes use vapor compression. This vacuum can be generated either via mechanical vacuum pumps, known as mechanical vapor compression (MVC), or thermal vapor compression (TVC).
In a basic MED system, seawater is introduced in the evaporator side of a multiphase heat exchanger and heating steam is introduced in the condenser side. The seawater evaporates as it passes across the heat exchanger surface producing concentrated brine at the bottom of each stage and an evaporated vapor. The vapor is then used as the heating medium for the next stage where the process is repeated. The temperature in each stage corresponds to the saturation pressure of the water vapor and the concentrated brine solution. The solution has a single saturation pressure and the temperature of the seawater solution is greater than that of the water vapor by the boiling point elevation. In the last stage, the produced steam condenses in a distillate condenser cooled by seawater, also referred to as a pre-heater or final condenser in the literature as it is also used to heat the influent seawater. For most MED processes more seawater is required for full condensation of the vapor from the last stage than is used in the evaporation process. This additional final condenser cooling flow is rejected back to sea.

The water production capacity of MED can be quantified by the Gain Output Ratio (GOR), the ratio of permeate mass per mass of heating steam supplied to the first stage. GOR is a convenient means to assess the performance of a simple MED system. An ideal MED single stage system without losses has GOR=1. In this ideal system, increasing the number of stages increases the GOR linearly as each stage would have a GOR of one. In actual systems processes do not perform ideally. In cases where the motive steam is not at saturation conditions or additional heat sources are used GOR is not a measure of thermodynamic, first law, performance. Another dimensionless performance measurement, the Performance Ratio (PR), better reflects the overall performances of these cases. PR is defined as the ratio of product water mass over the
mass that would be produced by condensing 1 kg of steam with a heat of vaporization of 2,326 
kJ/kg:

\[ \text{PR} = (2326 \text{ kJ/kg}) \times \text{Distillate production rate (kg/s)}/ \text{Process Thermal Power (kW)} \]  

(1)

As the system to be considered implement both superheated inlet conditions and heat addition, 
PR will be used as the performance characterization metric in this text.

The thermodynamic performance of MED depends on the process flow. The most 
efficient MED process is MED-TVC, which involves using low or medium pressure steam to take 
a portion of the vapor raised in one of the stages and recycle it into higher pressure vapor to be 
used as heating media for the first stage. TVC is an ejector or thermo-compressor driven by 
motive steam causing a normal shock that entrains a fraction of the vapor from the last stage and 
reduces the electrical energy consumption to less than 1 kWh/m³.

Figure 2.2 MED-TVC system process flow with absorption heat pump
Other methods to increase the performance of MED are using heat pumps or feed water heaters. For example, PR values up to 20 have been reported, achieved by using a LiBr heat pump between an intermediate stage and the first stage in a 14-stage solar hot water driven MED system, Figure 2.2 [6]. Instead of condensing the steam produced in the last stage with cold seawater, the steam is fed to the evaporator side of the heat pump driving the generation of steam to be absorbed by the absorber, a strong lithium bromide (LiBr)-water solution. The weak solution is then pumped to the generator where the steam is desorbed by using additional energy input. The water tank and steam input and output lines belong to the power production. This modification of the MED process allows additional extraction of thermal energy from the low temperature saturated steam produced from the last stage, which would otherwise be rejected to the additional cooling seawater.

Multiple Stage Flash Distillation
The multiple stage flash (MSF) distillation process is a series of sequential chambers, usually consisting of 24-30, with successively lower pressures that generate permeate vapor by flashing the heated seawater. In general the MSF process operates with a top brine temperature in the range of 90-110°C [7] and has a performance ratio between 7 and 8 [8]. MSF is 60% of the thermal desalination global market but this market share is reducing as MSF plants are converted to MED plants to gain overall efficiency. The exergy in the MED itself is less than in MSF as during evaporation because of the temperature difference between heating and evaporating fluid streams.
In each chamber of the MSF system seawater is flashed, or boiled at a temperature above the mixture's boiling point. The vapor generated is fed into the next chamber and condenses transferring the latent heat to the mixture. The condensed vapor is collected in distillate trays and pumped out of the system. The seawater is de-aerated before either being heated by a brine heater either at the inlet to the first chamber or before being mixed with the circulated brine in the flashing chamber. If the seawater goes directly into the flashing chamber as shown in Figure 2.4 as it is recirculated to the brine heater and acts as heat recovery mechanism as it flows through the consecutive chambers and flashes again. Excess heat in the system is rejected to the environment through overflow seawater cooling.

Figure 2.4 Schematic of a Typical Multistage Flash Desalination System

Figure 2.5 Schematic of a Typical Multistage Flash Desalination Chamber
The differences between MED and MSF are the top stage operating temperatures of 70°C versus 115°C and the oxygen concentration in this stage, 600 ppb versus 50 ppb oxygen content respectively [9]. As MED is operated at sub-atmospheric pressures the de-aeration and venting performed in MSF is not possible, the venting that does occur in MED is from the vacuum system removal of non-condensable gases. This results in the accumulation of stagnant pockets of oxygen in the first stages of MED, causing material cracking failure due to corrosion. The partial pressure of the dissolved oxygen concentration, $p_{O_2}$, of the in the evaporating brine inside the MED shell is governed by Henry’s law. It can be seen the equilibrium oxygen concentration, $c$, decreases with decreasing temperature. The for oxygen the coefficient $C$ in equation (3) is 1700 K with $k_H(T_{ref})= 769.23$ at 298 K.

$$p_{O_2} = k_Hc$$  \hspace{1cm} (2)

$$k_H(T) = k_H(T_{ref}) \exp \left[ -C \left( \frac{1}{T} - \frac{1}{T_{ref}} \right) \right]$$  \hspace{1cm} (3)

Heat Exchangers

The main component of each stage is a multiphase heat exchanger. Traditionally shell and tube heat exchangers have been used for MED, however recently plate heat exchangers (PHE) have been used due to their higher heat transfer capability and smaller size.

![Cross section of shell and tube heat exchanger for thermal desalination](image)

Figure 2.6 Cross section of shell and tube heat exchanger for thermal desalination

The heat transfer further across the multiphase heat exchanger limits system performance. The overall heat transfer coefficient, $1/(UA)$, defines the thermal resistance with which heat is transferred from the heating medium to the seawater. Decreasing this resistance or increasing the
conductivity (UA) decreases the losses of the system. The overall heat transfer coefficient is comprised of:

\[
1/(UA) = 1 / h_c + 1 / h_e + f + t / k \tag{4}
\]

where \( h_c \) is the convective resistance of the fluids, \( 1/h_c \) is related to frictional and viscous losses, \( f \) is caused by fouling, calcium or magnesium salt precipitation onto the heat exchanger surface, \( t/k \) is due to conduction across the heat exchanger plate with thickness, \( t \), and conductivity, \( k \).

### 2.1.2 Reverse Osmosis Desalination

RO is currently over 50 percent of the total desalinated water. RO is a process by which water is removed from solution by pressurizing a cross flow influent to a semi-permeable polyamid membrane. The driving force is the pressure from a high-pressure pump, typically 40 bar, to overcome the osmotic pressure of the solution and drive flux across the membrane. Product water salinity for membrane desalination has a salinity of less than 0.5 whereas desalinated water produced by thermal technologies has a salinity of \( 2.5 \times 10^{-7} \) [1].

The required driving pressure for RO can be decomposed into seven components.

\[
\Delta P = \Pi_{Osmotic|\pi} + \Pi_{Osmotic|\lambda} + \Delta P_{Reject} + \Delta P_{Foul} + \Delta P_{Polar} + \Delta P_{Membrane} + \Delta P_{Module} \tag{5}
\]

The first two terms in equation (5) are (1) the osmotic pressure and (2) the over pressure required to drive flux across the membrane. The osmotic pressure determined by the initial and final salt concentrations of the influent and effluent streams. The minimum osmotic pressure of standard
seawater, Appendix B, is 21.7 bar. The required over pressure is determined by the product to water recovery ratio, $X$,

$$X = \frac{m_p}{m_m}$$  \hspace{1cm} (6)

related to the influent and rejectate salinity. The higher the recovery ratio, the higher the salinity of the rejectate requiring a higher pressure to drive the water across membrane. On the other hand, a higher recovery ratio leads to less water needing pretreatment and smaller pump flows. The unrecovered over pressure (3) is the energy from the rejected pressurized stream not recovered by the energy recovery pump, between 10 and 20%. At the membrane surface ion and colloidal concentration polarization and fouling which increases the pressure required. The loss to the flow of product water at a given input pressure due to polarization impedance is affected by the flux: increasing the flux through the membrane, increases the concentration of salt near the membrane. The energy required to desalinate water considering osmotic pressure, osmotic over pressure, energy recovery and concentration polarization can be calculated generally, Figure 2.8.

![Figure 2.8 Theoretical RO energy consumption as a function of water recovery ratio and energy recovery, eta.](image)

The energy consumption due to membrane fouling (4), membrane (6) and membrane module pressure drops (7) along with seawater intake and return pumping are system and source water dependent. Of all the pressures the resistance due to fouling (4) is often the most significant
and difficult to characterize. Desalination plants, depending on seawater salinity that varies at different locations and the amount of pretreatment conducted, operate between a low of 42 bar to a high of 71 bar. Cleaning itself requires energy to flush or backwash, and no product water is produced during this time, which increases the energy per product water. Irreversible fouling resulting primarily from relatively small molecules and colloidal compounds has a much larger total energy cost.

2.1.3 Comparison of Desalination Methods

Scale Formation

One major technical barrier to achieving high flux desalination systems is scale formation. Scale formation is the precipitation of alkaline earth metals such as CaCO₃, Mg(OH)₂ and CaSO₄ onto to equipment surfaces. In thermal systems the scale act as an insulator to heat transfer and can form at below system level saturation conditions when the heat exchanger is not fully wetted. Anti-scalants and heat exchanger surface cleaning are used to reduce the impact of scale formation on the system performance [10]. Water recovery in RO is typically restricted to 35-40 percent of the solubility limit of these scale components. At 40°C, corresponding to the maximum operational temperature of RO membranes, CaCO₃ starts to precipitate at 2.2 times concentration, CaSO₄ at 5 times and MgSO₄ at 6.6 times [11].

The minimum rejectate volume of traditional plants is limited first by calcium carbonate scale formation. The industry general practice for prevention is the application of anti-scalants and to limit the system to 50 percent of the inlet seawater concentration. There are a number of relative indices for scale formation. For high TDS waters, 10,000 mg/L and greater, a more accurate measure is the Stiff & Davis Stability Index, S&DSI [12]. The S&DSI considers temperature, pH, alkalinity, calcium concentration, and ionic strength. It is valid from 0–100°C and for ionic strengths between 0 to 4 covering both RO and MED operating ranges. There are other indices for determining precipitation limits, however, the S&DSI is the current standard for use in seawater.

Energy Requirements

The minimum energy to desalinate water is the same regardless of the method based on the Gibbs limit. Based on the theoretical minimal separation energy is given by
where $W$ is the minimum isothermal reversible work of separation, $n_2$ is the total moles of water at the final state, $a_w$ is the water activity, $p$ is the vapor pressure of the solution assumed as an ideal gas, $p^0$ is the vapor pressure of pure water, $\Delta H$ is the change in enthalpy, $\Delta S$ is the change in entropy, and $\Delta G$ is the change of the Gibbs free energy. The minimum energy for desalination of seawater at standard conditions, equation (8), is 0.79 (kW h)/m$^3$ and 1.09 (kW h)/m$^3$ for 50 percent recovery [13]. There seems to be considerable confusion in the literature about the energetic cost of desalinating water as there is a failure to normalize the energy estimates before attempting to compare performances and discuss the relative merits of the various schemes.

The recovery ratio, volume of permeate per unit volume influent, is a critical desalination parameter. Of the primary seawater desalination processes both RO and MED have favorable recovery ratios, with MED having the greatest potential in normal operation for rejectate volume reduction.

![Comparison between Reverse Osmosis (RO) and Multi-Effect Distillation (MED) (Figure 2.9)](image)

Figure 2.9  Minimum energy requirements of RO and MED as a function of recovery ratio for standard seawater

The specific electrical energy consumption of RO is typically considered to be 2.2 to 6 kW he/ m$^3$. However, RO energy consumption increases nonlinearly with recovery ratio typically
limited to a concentrate TDS of 70 g/L. MED specific electrical and thermal energy consumption is published to be between 2 and 5 kW he/m$^3$ and 150 to 300 kWt/m$^3$, respectively. The published seawater desalination energetic cost or consumption in the literature is highly variable between reporting methods. There is not a normalization method to compare performances and discuss the relative merits of the various schemes that combines both thermal and electrical energy. One possible method would be normalize in terms of electrical energy, dividing the thermal inputs by the efficiency of the power plant (~35% is a typical value).

Table 2.2 Recovery Ratios by Desalination Process

<table>
<thead>
<tr>
<th>Recovery Ratio (%)</th>
<th>[5]</th>
<th>[14]</th>
</tr>
</thead>
<tbody>
<tr>
<td>MSF</td>
<td>10-25</td>
<td>9-20</td>
</tr>
<tr>
<td>MED</td>
<td>30-60</td>
<td></td>
</tr>
<tr>
<td>MED-TVC</td>
<td>23-33</td>
<td></td>
</tr>
<tr>
<td>MVC</td>
<td>23-41</td>
<td></td>
</tr>
<tr>
<td>RO</td>
<td>20-50</td>
<td>30-50</td>
</tr>
</tbody>
</table>

Table 2.3 Specific Energy Consumption by Desalination Process

<table>
<thead>
<tr>
<th>Source</th>
<th>Electrical [kW h$_e$/m$^3$]</th>
<th>Thermal [kJ/kg]</th>
<th>Total [kW h/m$^3$]</th>
<th>Equiv. [kW h$_e$/m$^3$]</th>
<th>Total [kJ/kg]</th>
</tr>
</thead>
<tbody>
<tr>
<td>MSF</td>
<td>2.5-5</td>
<td>250-330</td>
<td>40-120</td>
<td>10-58</td>
<td>95-288</td>
</tr>
<tr>
<td>MED</td>
<td>3-5</td>
<td>2.5-5</td>
<td>2-2.5</td>
<td>5-58</td>
<td>4-25</td>
</tr>
<tr>
<td>MVC</td>
<td>2-2.5</td>
<td>145-390</td>
<td></td>
<td></td>
<td>14-58</td>
</tr>
<tr>
<td>SD</td>
<td>1.5 - 2.5</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>ED</td>
<td>2.5-7</td>
<td>4-6</td>
<td></td>
<td></td>
<td>0.4-4</td>
</tr>
</tbody>
</table>

**Cost**

Similarly a comparison of RO and MED is challenging due to the nature of the inputs of the processes. A cost comparison by product volume is usually applied but the formulation of the
cost function is not detailed in the literature. The cost per unit volume is a beneficial method however; as commodity prices change so does this comparison. An overview of literature comparing the cost of the most common desalination systems shows there is not agreement amongst the researchers as how to discuss desalination cost as a whole, Table 2.4.

The cost to desalinate water is highly variable [14] based on fuel availability and the energy market. Depending on the energy source and material costs both RO and MED can be more economically viable, especially as fuel costs increase due to the capitol intensity of MED compared to RO. Published plant operational costs for RO are 900 to 1500 USD/m³/day with specific water production cost of between is between 0.7 and 1.5 USD/m³ and for MED are 900 to 1700 USD/m³/day with specific water production cost of between is between 0.7 and 1.5 USD/m³. Reverse osmosis (RO) and multiple effect distillation (MED) are the most developed and cost effective desalination technologies.

Table 2.4 Plant and Water Production Costs by Desalination Process

<table>
<thead>
<tr>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>[USD/m³]/d</td>
<td>[USD/m³]</td>
</tr>
<tr>
<td>MSF 1500 - 2000</td>
<td>0.7-1.5</td>
</tr>
<tr>
<td>MED 900 - 1700</td>
<td>0.27-1.49</td>
</tr>
<tr>
<td>TVC 900 - 1700</td>
<td>0.46-1.21</td>
</tr>
<tr>
<td>MVC 1500 -2000</td>
<td>0.46-1.21</td>
</tr>
<tr>
<td>RO 900 - 1500</td>
<td>0.5 avg 0.45-6.56</td>
</tr>
<tr>
<td>ED</td>
<td>0.58</td>
</tr>
</tbody>
</table>

2.2 CSP-DSW Dual-Purpose Plants

2.2.1 Brayton Cycle Turbines

Most CSP plants in use today use saturated steam turbines but it is desired to use superheated steam as the efficiency of a saturated steam turbine is significantly lower to that of a superheated steam turbine. Typical superheated turbine operation conditions are inlet pressure and temperature up to 140bar and 540°C with exhaust at 50mbar and 33°C corresponding to an exit steam quality of 0.85 [16]. Turbine efficiencies range between 30% and 40%; with efficiency increasing with power output. An average value of 35% will be used for power requirement comparisons. Two types of superheated steam turbines are discussed for use with MED: extraction and condensing. The main difference between condensing turbine and extraction
turbines is that an additional extraction port is added to the turbine, which can supply steam at varied pressure and flow rate within the appropriate design limits of the turbine though common extraction pressures are 1, 2 and 6 bar.

2.2.2 Integration of Desalination into Traditional Power Cycles

One challenge of dual-purpose power plants is the reduction in the efficiency of the power cycle due to the higher turbine outlet temperatures required to drive the thermal desalination [13]. In terms of utilizing the produced steam, a dual-purpose plant consisting of CSP and thermal desalination reaches thermodynamic efficiencies above 90%. The advantage of MED over other thermal methods is that there are less irreversible losses for both water and power production. The lower first stage temperature allows for a larger temperature difference in the power cycle.

For the gas turbine the inlet to outlet pressure ratio and the firing temperature are the critical parameters, with power output increasing with increasing pressure ratio and firing temperature [17]. Where as, the turbine outlet temperature is the most significant fact for the MED process. Reduction of system efficiency due to higher temperatures and pressure at the turbine outlet to allow for the required thermal desalination input of approximately 70°C and the correlating turbine outlet pressure of 0.35 bar. The overall power cycle efficiency would increase if it were possible to reduce the required input pressure into a MED-TVC desalination plant without lowering the first stage temperature of the desalination system.

Steam ejectors, TVC device, require steam to be extracted from the turbine at relatively high extraction pressures usually between 5 and 10 bar. For CSP-DSW a small fraction of the steam would be extracted from the turbine. The use of steam ejectors reduces the power requirement of MED significantly. There is tradeoff, however, as the power lost due to extraction at extraction pressures higher than 1 bar becomes significant.
Figure 2.10 Schematic of a CSP-DSW dual-purpose plant

Figure 2.11 Single and dual purpose plant steam cycle T-S diagram
CHAPTER 3: Theory and Modeling

3.1 General Multiple Effect Distillation Model

The MED model for each stage and the overall process is based on continuity: conservation of mass, momentum and energy. There are two models commonly used in MED modeling by Darwish [18-20] and El-Dessouky [8]. Modifications to these models have been made to account for the different heat exchanger types used and for variations in process configurations. The Darwish model is empirical where as the El-Dessouky model closer represents the continuum approach applied here. The following components are considered in the analysis of the MED-TVC system, Figure 3.1: the MED stage, final condenser, feed water heaters, TVC, and heat exchanged from a low temperature heat source.

3.1.1 Water and Seawater Properties

The thermodynamic properties of steam and water are determined using the International Association for Properties of Water and Steam Industrial Formulation 1997 (IAPWS IF97). The exception is the ideal gas assumption are used for TVC consistent with compressible fluid dynamics analysis. Correlations developed by Millero & Pierrot [21] are used for seawater enthalpy calculation. It is valid for temperature range of 0 - 200°C and salinity range of 0 – 120. The boiling point elevation BPE at a given pressure is the increase in the boiling temperature due to the salts dissolved in the water. The boiling point elevation, BPE, is calculated using the empirical formula [22]. Full equations for seawater properties are given in Appendix B.

3.1.2 Multiple Effect Desalination Stage

In the MED stage i, Figure 3.2, conservation of mass is applied to both the seawater saline component and to the water.

\[
\begin{align*}
\text{water} & : \quad \dot{m}_{b_{N-1}} + \dot{m}_{sw} = \dot{m}_{p_N} + \dot{m}_{b_N} \\
\text{salt} & : \quad \dot{m}_{b_{N-1}} c_{b_{N-1}} + \dot{m}_{sw} c_{sw} = \dot{m}_{b_N} c_{b_N}
\end{align*}
\]

In the consideration of heat transfer and overall energy balance of the system, is assumed the heating fluid is condensed at the saturation pressure without further heat transfer. For stages 2 to N, the final stage, the condensation is assumed to isothermal. For the MED stage conservation of energy:
\[ 0 = [\dot{m}_{sw} h_{sw}(T_{ph}, c_{sw}) + \dot{m}_{p,i} h_w(T_{i-1}, x = 1) + \dot{m}_{b,i} h_{sw}(T_{i-1}, c_{i-1}) + Q_i]_{in} \]
\[ -[\dot{m}_{c,i} h_w(T_i, x = 0)]_{out} \]
\[ Q_i = \eta_{therm} \dot{m}_{p,i-1} [h_w(T_{i-1}, x = 1) - h_w(T_i, x = 0)] \]

The permeate vapor and the brine temperature are assumed to be in equilibrium and saturated at the vapor pressure of each solution. The difference between the saturation temperature of the brine and the vapor is the BPE.

\[ T_i = T_{bi} - BPE(i) \]

In the general model the overall heat transfer coefficient is a fixed value based on literature of 4 kW/(m²K) [23]. The preliminary overall heat transfer value fixed value agrees with the published falling film desalination conditions [24] and are used to determine the required heat exchanger area for each stage.

\[ A_i = \frac{Q_i}{U_i LMTD_i}; \quad i = 1 \]
\[ A_i = \frac{\dot{m}_{pi}[h(v,T_{pi}) - h(l,T_{pi})]}{U_i LMTD_i}; \quad i = 2 \text{ to } N \]

The specific heat exchanger area,

\[ sA_i = \sum_{i=1}^{N} \frac{A_i}{m_{pi}} \]

is used in the parametric study, 4.1, as a cost normalization method.

3.1.3 Final Condenser
The final condenser uses the influent seawater to condense the final MED stage permeate, Figure 3.3. Additional seawater flow than is desired for the MED distillation influent is required to condense the final stage permeate vapor because the temperature difference between the fluid streams is low and there is only a phase change on the condenser side. The model is based on the log mean temperature difference (LMTD) between the two heat exchanger inlets of final condenser:

\[ 0 = [N \dot{m}_{sw} h_{sw}(T_{sw}, c_{sw}) + \dot{m}_{p,i} h_w(T_{i}, x = 1)]_{in} \]
\[ -[N \dot{m}_{sw} h_{sw}(T_{ph}, c_{sw}) + \dot{m}_{p,i} h_w(T_{ph}, x = 1)]_{out} \]

18
LMTD = \frac{(T_{sw} - T_{ph})}{\ln\left(\frac{T_N - T_{ph}}{T_N - T_{sw}}\right)} \tag{18}

A_{ph} = \frac{\dot{m}_{pN}\left[h(v,T_{pN}) - h(l,T_{pN})\right]}{U_{ph}LMTD_{ph}} \tag{19}

The specific condenser heat exchanger area,

sA_{ph} = A_{ph} / \sum_{i=1}^{N} m_{pi} \tag{20}

is also used in the parametric study, 4.1, as a cost normalization method.

3.1.4 Feed water heaters

Traditionally the feed water heaters are analyzed with a constant temperature increase across each feed water heater, such as the Amer analysis discussed below where the elevated feed water temperature, T_f, into each stage is constrained to 3°C. However as different temperature distributions and numbers of stages is considered, a feed water heater model is based on the NTU method with feed water heater effectiveness of 0.5 is used.

\[ Q_{\text{max}} = (\dot{m}C_p)_{\text{min}}(T_{i-1} - T_{ph}) \tag{21} \]

\[ Q = \varepsilon Q_{\text{max}} \tag{22} \]

\[ \varepsilon = \frac{(\dot{m}C_p)_{ph}(T_f - T_{ph})}{(\dot{m}C_p)_{\text{min}}(T_{i-1} - T_{ph})} \tag{23} \]

\[ T_f = \frac{\varepsilon(\dot{m}C_p)_{\text{min}}(T_{i-1} - T_{ph})}{(\dot{m}C_p)_{ph}} + T_{ph} \tag{24} \]

3.1.5 Thermal Vapor Compressor

The analysis of TVC and ejector models has been performed for desalination systems using a classical compressible fluid dynamics analysis. It is assumed the ejector is well insulated and adiabatic, the kinetic energy at TVC inlets and outlet are negligible and mixing of the motive
steam and entrained vapor occurs at the entrained vapor pressure. Isentropic relations with
nozzle, mixing, and diffuser efficiencies are employed to account for non-ideal effects of
frictional and mixing losses. The nozzle, mixing and diffuser efficiencies are 0.90, 0.90, and
0.95, respectively [25].

The TVC, Figure 3.4, is analyzed in four constant volume sections: an isentropic adiabatic De Laval nozzle (S to 1), a constant pressure mixing section (1 to 2), a constant area
duct with supersonic inlet condition causing a normal shock (2 to 3) and a diffuser to further
compress the motive steam through the conversion of kinetic energy into fluid pressure (3 to
exit). In the adiabatic nozzle section the enthalpy and velocity of the exit steam is found relating
the isentropic process, subscript 1, and the assumed nozzle efficiency.

\[ v_1 = \left[ 2\eta_n (h_S - h_{1s}) \right]^{1/2} \]  \hspace{1cm} (25)

\[ h_1 = h_S - \eta_n (h_S - h_{1s}) \]  \hspace{1cm} (26)

The mixing occurs at a constant pressure of the entrained vapor. To account for the frictional
losses due to the viscous interaction between the, the motive steam and entrained vapor,
the mixing efficiency is introduced into the conservation of momentum equation.

\[ \eta_m (\dot{m}_s v_1 + \dot{m}_v v_v) = (\dot{m}_s + \dot{m}_v) v_2 \] \hspace{1cm} (27)

\[ (\dot{m}_s + \dot{m}_v) \left( h_2 + \frac{1}{2} v_2^2 \right) = \dot{m}_s \left( h_1 + \frac{1}{2} v_1^2 \right) + \dot{m}_v \left( h_v + \frac{1}{2} v_v^2 \right) \] \hspace{1cm} (28)

If the inlet into the constant area section is supersonic, an irreversible normal shock
occurs causing a sudden pressure and temperature increase. Both the pressure and temperature
depend on the Mach numbers before and after the shock. The entropy after the shock increases as,

\[ s_3 - s_2 = \frac{R}{\gamma - 1} \left[ \ln \left( \frac{1 - \gamma + 2\gamma Ma_2^2}{\gamma + 1} \right) + \gamma \ln \left( \frac{2 + (\gamma - 1)Ma_2^2}{(\gamma + 1)Ma_2^2} \right) \right] \] \hspace{1cm} (29)

In the diffuser, the steam is compressed further by converting the kinetic energy into
pressure energy. The process is near isentropic deviating by the diffuser efficiency. The
conditions for the pressure, enthalpy, temperature, and entropy of the steam at the diffuser exit
are,

\[ p_{exit} = p_3 \left( 1 + \frac{\eta_d (1 - \gamma)}{2} Ma_3^2 \right) \] \hspace{1cm} (30)

\[ h_{exit} = h_3 + \frac{1}{2} v_3^2 \] \hspace{1cm} (31)
\[
T_{exit} = T_3 \left(1 + \frac{(1-\gamma)}{2} Ma_s^2 \right) \tag{32}
\]

\[
s_{exit} = s_3 + R \left[ \ln \left( \frac{p_3}{p_{exit}} \left(1 + \frac{(1-\gamma)}{2} Ma_s^2 \right)^{\frac{\gamma}{\gamma-1}} \right) \right] \tag{33}
\]

An optimization algorithm is applied to reduce the exergetic losses in the TVC and maximize the total heat input into the MED system. The total enthalpy at the TVC outlet is directly related to the entrainment ratio, the motive steam to entrained vapor flow ratio, \( r = m_s/m_v \).

For a given motive steam inlet condition the optimal entrainment ratio maximizes the heat transfer to the first stage of the MED, \( Q_1 \). A tandem two-ejector system is considered if the motive steam and the vapor entrained from the last stage does not achieve required pressure with one steam ejector. It is assumed that 20\% of steam flows into the first ejector and remaining amount of the steam drives the second ejector. For given motive steam and entrained vapor properties, the solver examines if the maximum achievable pressure, \( p_{max} \), is higher than the minimum required pressure, \( p_{req} \). If not, two steam ejectors are examined to increase \( p_{max} \). When \( p_{max} \) is greater than \( p_{req} \), the entrainment ratio that maximizes the available energy of the mixed steam output is found. This optimization allows for lower pressure motive steam. A flow charge for the optimization algorithm is given in Appendix A.

3.1.6 Low Temperature Heat Harvesting

A heat exchanger allows the addition of heat (\( Q_{H} \)) harvested from the solar collection system to the steam extracted from the turbine before it enters the TVC. This is added to the process thermal power (\( Q_{P} \)), Figure 2.10, which is the dominant energy input to the MED, while electrical energy for fluid pumping is minimal.

The collection and storage of heat in a CSP system at least 550°C, a standard steam turbine inlet temperature, but there are conductive system losses to the environment which can be captured at a lower temperature. The energy harvested from the solar collection system is added to the motive steam reducing the required turbine outlet temperature. For the operating conditions of a typical MED system the inlet, steam inlet mach number of 0.5, temperature of 73°C, and 50 kPa-a pressure, the steam temperature can be increased by 100°C, Figure 3.5.

The heat addition is modeled using the Rayleigh line; constant heat addition in a constant diameter pipe without friction. The downstream conditions of the motive steam after the harvested heat addition is based on the change in stagnation temperature, \( T_0 \). The increase of the stagnation temperature and decrease of the stagnation pressure superheats the motive steam.
\[ Q_H = \dot{m}_s C_p (T_{02} - T_{01}) \]  

(34)

3.1.7 Concentrated Solar Power and MED Integration Model

In order to determine the operating conditions with the highest permeate output, the following algorithm is performed for the simulation of MED-TVC with harvested heat addition. It performs an nonlinear analysis of the system using the seawater inlet temperature after the final condenser, \( T_{ph} \), as convergence criteria and employs global constraints on the brine salinity and stage temperatures.

Step 1. Compute harvested heat input to motive steam

Step 2. Set initial guess for TVC entrainment (\( m_{v,TVC} \)), heat transfer to the first stage (\( Q_1 \)), preheated seawater temperature (\( T_{ph} \)), and seawater inlet flow rate, \( m_{sw} \)

Step 3. Perform TVC optimization algorithm, Appendix A

Step 4. Compute MED performance through continuity, conservation of energy and thermodynamic state for stage \( i \). The heat transfer to the next stage \( Q_{i+1} \) is the energy to fully condense the vapor permeate vapor of the previous stage. The brine from stage \( i \) is first mixed with the distributed seawater feed and is the saline feed to stage \((i+1)\) repeat until \( i=N \)

Step 5. - Compute \( T_{ph} \) from the amount of vapor produced in the last stage (\( m_{pN}=m_{v,TVC} \))
   - Compute maximum brine concentration, \( \text{max}(c_{b,i}) \)
   - Compute total permeate mass flow, \( \Sigma m_{po,new} \)
   if \( \text{max}(c_{b,i})>c_{max} \), increase seawater flow rate, \( m_{sw,i} \), into each stage
   return to Step 4

Step 6. If \( T_{ph} \) does not converge within 0.01°C return to Step 4

Step 7. if \( \Sigma m_{po,new}<\Sigma m_{po,old} \)
   decrease seawater flow rate, \( m_{sw,i} \), into each stage
   return to Step 4

Step 8. if variation in \( m_{po,i,new} \) to \( m_{po,i,old} \) is less than \( 10^{-3} \)
   return to Step 4

Step 9. Return to Step 2 until convergence of \( Q_1 \)

Step 10. Compute overall process performance \( s_{Ai}, s_{Ac}, PR \)
3.2 Dimensional Modeling

In addition to the system model, dimensional correlations for the specific experimental design are employed to better predict the pressure drop and overall heat transfer coefficient of the PHE. Experimental data regarding falling film herringbone-type, also called chevron or corrugated, PHE can be extracted from the scientific literature for water two phase heat exchangers. The total pressure drop is comprised of gravitational, acceleration and frictional terms. The largest contribution comes from the frictional pressure drop; calculated using the Lockhart-Marinelli model. These predictions were found to vary within 20% [26]. The water model predicts a total pressure drops between 5-100 kPa/m, strongly dependant on the Reynolds number.

\[ \Delta p_t = \Delta p_g + \Delta p_{acc} + \Delta p_F \]  \hspace{1cm} (35)

\[ \Delta p_g = \int_0^L \left[ \rho_v \alpha + \rho_l (1 - \alpha) g \right] dz \]  \hspace{1cm} (36)

\[ \frac{1}{\alpha} = 1 + \frac{1 - x}{x} \left( \frac{\rho_v}{\rho_i} \right)^{2/3} \]  \hspace{1cm} (37)

\[ \Delta p_{acc} = G^2 \left[ \left( \frac{(1 - x_2)^2}{\rho_i (1 - \alpha_2)} + \frac{x_2^2}{\rho_v \alpha_2} \right) - \left( \frac{(1 - x_1)^2}{\rho_i (1 - \alpha_1)} + \frac{x_1^2}{\rho_v \alpha_1} \right) \right] \]  \hspace{1cm} (38)

G is the steam mass velocity, \( \alpha_1 \) and \( \alpha_2 \) are the inlet and outlet void fractions respectively, and \( x_1 \) and \( x_2 \) are the inlet and outlet steam qualities.

\[ \Delta p_{L,F} = \frac{\dot{m} L \xi}{2 \rho_i d_h} \]  \hspace{1cm} (39)

\[ \xi = 0.56 \text{Re}^{-0.12} \]  \hspace{1cm} (40)

The equivalent diameter \( d_h \) is defined as \( dh = 2h \), where \( h \) is the average channel gap.

\[ \Delta p_{CO,F} = \Phi^2 \Delta p_{L,F} \]  \hspace{1cm} (41)

\[ \Phi^2 = F X_{LM}^{-2} \]  \hspace{1cm} (42)

\[ X_{LM} = \left( \frac{1 - x}{x} \right)^{0.9} \left( \frac{\rho_v}{\rho_i} \right)^{0.5} \left( \frac{\eta}{\eta_v} \right)^{0.1} \]  \hspace{1cm} (43)

An empirical correlation for the overall heat transfer coefficient is used based on falling film steam condensation in corrugated PHE, was found to be between 2 and 30 kW/(m\(^2\)K) and with 30% variation [27]. The non-dimensional parameters for PHE heat transfer analysis are:
\[
Re_h = \frac{G_{eq}d_h}{\eta_L}
\]  
(44)

\[
G_{eq} = \bar{m}\left[(1 - x) + x\left(\frac{\rho_t}{\rho_v}\right)^{\frac{1}{k}}\right]
\]  
(45)

\[
Nu_{CO} = CRe^m_Pr_{L}^{0.33} = \frac{\alpha_{CO}d_h}{\lambda_L}
\]  
(46)

\[
Pr_{L} = \frac{C_p\eta_L}{k_{h\text{em\text{cond}}}}
\]  
(47)

Which can then be re-dimensionalized using the standard heat transfer relationship. As the pressure drop and overall heat transfer coefficient are linked an iterative solution approach is applied to verify convergence of both models.

### 3.3 Validation

The general version of the MED-TVC code has been verified for several desalination plants. The plant data [20, 28-30] is used to compare the performances of the simulation program. Four cases are examined; Mirfa 4-stage plant, Al-Taweelah A1 6-stage system, Sayyaadi-Model 10-stage system all employ MED-TVC at different motive steam inlet conditions, and the Nafey 7-stage model for a MED system, Table 3.2.

The model has been correlated to the published plant data within 20%. The model performs consistently for all MED-TVC and MED conditions with estimated outputs 19.5 to 21.5% lower than the actual system for the PR and the total distillate production. The TVC model because of the consistency of the error appears to agree with plant performance and the ejector outlet pressure and entrainment to compression ratios have been compared with literature with significantly better agreement than the MED-TVC model. The MED stage and feed water heater models appear to be inconsistent. Feed water heaters were not used in the for the comparison simulation model and the heating steam is assumed to exit the condensers at saturated conditions. Updating the model to include the different feed water heat schemes employed in each case and including a second single phase condenser heat exchange correlation are recommended.
3.4 Figures

Figure 3.1 Process flow diagram for MED with TVC and harvested heat

Figure 3.2 Schematic of MED stage

Figure 3.3 Schematic of final condenser
Figure 3.4 TVC model schematic

Figure 3.5 Theoretical motive steam temperature increase as a percent increase over of the motive steam enthalpy

3.5 Tables

Table 3.1 The PHE two phase evaporation modelling constants C, m and k dependence
corrugated plate geometry

<table>
<thead>
<tr>
<th>Combination of the plates</th>
<th>C</th>
<th>m</th>
<th>k</th>
<th>F</th>
</tr>
</thead>
<tbody>
<tr>
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<td>3.77</td>
<td>0.43</td>
<td>0.14</td>
<td>0.5</td>
</tr>
<tr>
<td>h/l</td>
<td>3.2</td>
<td>0.46</td>
<td>0.3</td>
<td>0.1</td>
</tr>
<tr>
<td>l/l</td>
<td>0.325</td>
<td>0.62</td>
<td>0.4</td>
<td>0.1</td>
</tr>
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Table 3.2 Comparison of model predictions and actual systems

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</thead>
<tbody>
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<td></td>
<td>Model</td>
<td>Actual</td>
<td>Model</td>
<td>Actual</td>
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<tr>
<td>TVC</td>
<td>1</td>
<td>1</td>
<td>0</td>
<td>1</td>
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<tr>
<td>Feed water heaters</td>
<td>Y</td>
<td>Y</td>
<td>Y</td>
<td>N</td>
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<tr>
<td>Motive steam flow rate [kg/s]</td>
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<td>57.9</td>
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<td>180</td>
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<td>Motive steam pressure [bar]</td>
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<td>2.8</td>
<td>0.35</td>
<td>2</td>
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<td>46</td>
<td>45</td>
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<td>Feed seawater temperature to the 1st stage [°C]</td>
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CHAPTER 4: Design for the integration of MED into a CSP-DSW plant

4.1 Parametric study

A parametric study of the operating and input conditions was performed to determine the configuration for the integration of MED into a CSP plant. The design variables investigated in the study are the number of MED stages, the first and final stage temperatures, motive steam pressure and temperature and the use of a secondary low temperature heat source. The environmental condition considered is the seawater inlet temperature as it fluctuates throughout the year. For each case both TVC and MVC were considered. The baseline conditions for the parametric study are inlet seawater with salinity of 40 at 25°C and motive steam input at 150°C, 2 bar and Mach number 0.5. The baseline configuration consists of a 10 stage MED with seawater inlet flow distributed equally into each stage without low temperature heat addition or feed water heaters. The operating condition of the MED process is set to a first stage temperature of 70°C and final stage temperature 35°C, however, when TVC is considered the first stage temperature may be decreased.

The PR is used as the measure of the system efficiency and the heat exchanger areas are used a comparison of cost. Only the specific stage and condenser heat exchanger areas are considered in the cost evaluation as the heat input is maintained across each parametric case and the electrical power consumption is small relative to the thermal heat input. Also the heat exchanger area per stage is allowed to vary. For some input cases in the study a solution was not found and the results are as any solution either without convergence or with ill-conditioned inputs was removed from the simulation results. An more robust non-linear solver, such as a dog-leg method would reduce the instances of this particular error type.

4.1.1 Seawater Temperature

A seawater inlet temperature range of 10 and 45°C is considered. The seawater inlet temperature is a significant factor in determining the condenser size. Three cases are investigated: final stage temperature of 35°C, final stage temperature 5°C above seawater inlet temperature and final stage temperature 10°C above the seawater inlet temperature. For all cases the PR and specific heat exchanger area increase with increasing seawater inlet temperature. Significant performance gains can be made using the fixed final stage to seawater inlet temperature difference but at the cost of total heat exchanger area while the condenser heat exchanger area remains constant. The fixed final stage temperature, as is the configuration of most plants, maintains a relatively low but constant performance and both heat exchanger areas.
The optimal configuration is dependent on the ratio of the specific heat exchanger area to the PR and Figure 4.6 depending on the inlet temperature, with this control being more variable for the MED system where as the MED-TVC performs best with constant 10°C difference. Specific control strategies could be developed to exploit this potential performance improvement.

The seawater temperature does not impact the TVC performance. For a constant final stage temperature the relationship between the PR and the seawater temperature is linear for both system configurations. Furthermore, there is a constant PR difference of 1 between the MED and MED-TVC. There is performance improvement between MED-TVC and MED at higher temperatures for the constant temperature difference control but is a function of the temperature difference between the stages, not the actual seawater inlet.

4.1.2 Number of Stages

The number of stages has a significant impact on the amount of permeate production. Distillate production increases with the number of stages, Figure 4.7, and there is a diminishing return in terms of distillate production with increasing number of stages used. The upper limit on the number of stages with the default first and final stage temperatures is 25 where the performance ratio asymptotically approaches a maximum value.

With the variable heat exchanger area per stage the benefit of increasing the number of stages is clearly shown by the similarly asymptotically decreasing stage heat exchanger area with increased number of stages, Figure 4.8. The heat exchanger area required to produce the same permeate volume decreases, by a factor of 10, with the increase in number of stages from one to ten. The advantage significantly decrease after the addition of 15 or more stages approaching a steady value so the addition of more stages does not increase the performance gain. The condenser heat exchanger area increases with the number of stages as the required inlet seawater flow also increases.

4.1.3 Final Stage Temperature

Increasing the final stage temperature increases the PR for a fixed number of stages. Final stage temperature between 25 and 60°C were considered with a fixed seawater inlet temperature of 25°C. As the condenser size is a function of the permeate produced in the last stage, not the seawater flow, it is not significantly impacted by changing the seawater inlet condition. The stage heat exchanger area increases with increasing final stage temperature due to the increase in the temperature difference between the stages decreases.
4.1.4 First Stage Temperature and Top Brine Temperature

The top brine temperature, varied from 50 to 90°C, is the limiting temperature for the steam inlet can be and is the temperature of the first stage plus the boiling point elevation. The PR decreases with increasing top brine temperature as expected with a constant the number of stages. The heat exchanger area and PR relation, PR increasing directly with surface area, the top brine temperature is related to the overall thermal system such that this general variation does not yield performance to cost beneficial design configurations. The perturbation of the motive steam pressure and temperature better represent the system limits.

4.1.5 Motive Steam Pressure and Temperature

The motive steam temperature was varied from 120°C, near saturation, and 200°C. At a high pressure and temperature relative to the first stage saturation condition further superheating of the steam does not have a significant influence on the system performance or heat exchanger area, Figure 4.17. This relationship is discussed below for low temperature heat harvesting.

The inlet pressure has a significant impact on the performance between the simulations with and without TVC. Inlet pressures between 0.33 and 5 bar were considered. At pressures greater than 1.2 bar the TVC system performance increases with increasing motive steam pressure. Where as, the system performance without TVC decreases with increasing motive steam pressure because the MED only system assumes the first stage outlet condition is saturated liquid so the full heat input is not utilized. The TVC drops the inlet temperature to the first stage and with higher pressures the system approaches the optimal MED inlet conditions.

At low motive steam pressures, less than 1.2 bar, the performance of the system varies significantly. From the detailed parameterization from 0.4 to 2 bar steam inlet conditions, the system operates best at steam inlet pressures closest to the saturation condition of the top brine temperature. However to achieve this result the first stage temperature is decreased to ensure full condensation of the motive steam.

4.1.6 TVC

The PR of the TVC system is consistently higher than that of the system without. This is because the TVC acts as a heat recovery device. The energy from the steam entrained in the TVC is added to that of the motive steam, increasing the overall heat input into the first stage. The smaller heat exchanger area follows a similar argument; the inlet temperature of the TVC is lower than that of the system without and the first stage heat exchanger exhibits better thermal performance. A general observation from the results in the TVC system is less sensitive to input variations and has a higher overall performance.
4.1.7 Harvested Heat Input

For the harvested low temperature heat addition variation the total inlet energy into the system was varied from no additional heat into to the an equal amount of energy input into the system from the harvesting mechanism as was in the motive steam. Multiple cases are considered here as this determines the minimum inlet pressure and temperature of the system when both harvested heat input and TVC are employed. The second set of variations is the motive steam inlet pressure and temperature to standard turbine outlet values.

The low temperature of the harvested heat source allow for boosting of the MED to close to nominal conditions for a low total percent additional heat input. As this heat input is low, only requiring a superheat of 20°C for significant reductions the required in turbine outlet pressures. It follows that water produced from the harvested energy versus energy that could be produced from configuration appears to be the most viable low grade energy harvesting for the combined plant.

4.2 Recommended plant configuration

A comparison of the performance ratio to the total specific heat exchanger area, both the stage and condenser, gives insight into which system configurations have the greatest thermodynamic benefit. The configurations, which have the greatest impact on performance, are the number of stages and the top operating temperature. However, using an influent seawater-to-final stage temperature difference control strategy can make low cost performance gains. The performance can further be improved without adding heat transfer media by operating the TVC over favorable inlet pressure ranges: for the reference conditions above 2 bar or below 0.7. As it is desired to operate a low pressure the combination of low temperature heat addition and optimization for the lowest TVC inlet temperature is advantageous. This configuration allows for the turbine to output as close to the bottom of the cold cycle as possible while maintaining the same MED inlet superheat condition. Increasing the motive steam temperature more than 20°C above the saturation point does not increase the system efficiency or reduce the cost significantly.
4.3 Figures

Figure 4.1 PR for different seawater influent and final stage temperature control strategies

Figure 4.2 Stage specific heat exchanger area for different seawater influent and final stage temperature control strategies
Figure 4.3 Final condenser specific heat exchanger area for different seawater influent and final stage temperature control strategies.

Figure 4.4 Overall system performance, PR, as a function of seawater intake temperature for final stage temperature constraint of 5°C above the intake temperature.
Figure 4.5 Heat transfer area per product volume as a function of Seawater intake temperature for final stage temperature constraint of 35°C.

Figure 4.6 Comparison of seawater intake temperature control strategies on system performance and total heat transfer area per product volume
Figure 4.7 Overall system performance, PR, as a function of number of MED stages

Figure 4.8 Required heat exchanger area per unit product as a function of number of MED stages
Figure 4.9 Effect of last stage or final condenser temperature on system performance, PR, for intake temperature constraint of 5°C below the final stage vapor temperature.

Figure 4.10 Effect of last stage or final condenser temperature on heat transfer area seawater for intake temperature constraint of 5°C below the final stage vapor temperature.
Figure 4.11 Effect of first stage or top brine temperature on overall system performance

Figure 4.12 Effect of first stage or top brine temperature on heat transfer area
Figure 4.13 Motive steam inlet pressure versus system performance for constant heat input at steam inlet condition of 180°C maintained by variation of steam inlet flow rate

Figure 4.14 Motive steam inlet pressure versus specific heat exchanger area for constant heat input at steam inlet condition of 180°C maintained by variation of steam inlet flow rate
Figure 4.15 Motive steam inlet pressure versus first stage temperature change from TVC algorithm for constant heat input at steam inlet condition of 180°C maintained by variation of steam inlet flow rate.

Figure 4.16 Motive steam inlet temperature versus performance ratio for constant heat input at steam inlet condition of 2 bar maintained by variation of steam inlet flow rate.
Figure 4.17 Motive steam inlet temperature versus specific heat exchanger area for constant heat input at steam inlet condition of 2 bar maintained by variation of steam inlet flow rate.

Figure 4.18 Effect of the ratio of harvested heat to motive steam input heat performance with overall energy input maintained through variation of the motive steam flow rate.
Figure 4.19 Ratio of harvested heat to motive steam input heat required heat exchanger area with overall energy input maintained by variation of the motive steam flow rate.

Figure 4.20 Inlet temperature variation as a function of the ratio of harvested heat to motive steam input heat performance for a constant overall energy input.
Figure 4.21 MED-TVC performance and total heat transfer area required per water flux for the study parameters for a constant overall energy input and baseline input conditions.

Figure 4.22 Detail of baseline condition for MED-TVC performance and total heat transfer area required per water flux.
Figure 4.23 MED performance and total heat transfer area required per water flux for the study parameters for a constant overall energy input and baseline input conditions

Figure 4.24 Detail of baseline condition for MED performance and total heat transfer area required per water flux
### 4.4 Tables

Table 4.1 Parametric study baseline condition inputs

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Table 4.2 Parametric study baseline condition simulation results

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<td>2664</td>
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<td>( m_{po} ) [kg/s]</td>
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CHAPTER 5: 10 kW-thermal Multiple Effect Distillation System Design

5.1 Design

The design of a small scale, 10 kWt, system to demonstrate proof of principle of the CSP-DSW system integration is discussed. The heat exchanger is modular so the number of plates in each stage can be varied in to achieve the desired performance in each stage. To design and characterize the MED system for this purpose an understanding of the heat exchanger performance (accurate estimates of the pressure drop and heat transfer coefficient), permeate vapor path, and overall system steady state and transient performance is required. Due to variations in solar radiation, transient response of the MED system between 10 and 20 kWt heat input is considered. Off the shelf components for use in home potable water distribution systems and sanitary/chemical processing have been employed in the apparatus construction, as many components traditionally used in MED are not commercially available in this scale.

5.1.1 Heat Exchanger

A seawater-compatible plate heat exchanger design rated for up to 20 kW heat input by Alfa Laval similar to their state-of-the art large-scale MED units was selected. Parallel plate falling film heat exchangers have been reported to exhibit evaporative heat transfer coefficients up to 4,000 W/m² K (Tonner, 2001). Modifications to the sealing gaskets to allow for three-phase flow are made, Figure 5.1(c).

Although the performance correlations of the heat exchangers used in MED are proprietary, two methods to predict the pressure drop and overall heat transfer coefficient for the four-stage design have been used. First for the M3-FG heat exchanger, Figure 5.1(a), Alfa Laval provided overall heat exchanger performance estimates for the conditions of a three-stage system with equal stage temperature distribution of 80°C to 35°C. The heat exchanger has a surface area of 0.353 m² comprised of 13 plates in an alternating pattern of alternating chevrons with a 60° corrugation angle and fluid passage gap of 2.2mm. The performance Alfa Laval provided is significantly smaller than what is published in the case of the EasyMED system (Renaudin, 2005). However, this discrepancy is explained in terms of the difference in Reynolds number (Re) between the small scale MED and the literature.

5.1.2 Material Section

Materials were selected to meet the steam and vapor temperatures, for seawater and potable water compatibility. CPVC was selected for the low temperature lines as it has a good
heat and chemical resistance. Though it is used for water and steam applications, corrosion of galvanized steel has been observed allowing for biological fouling buildup in the system. Additionally these iron deposits effect the flow instrumentation. Instead for the seawater high temperature components an aluminized brass or stainless steel grade 316 is recommended. Different materials were not used for each stage, with decreasing temperature, but for a larger scale system this would be more cost beneficial. The heat exchanger surfaces were selected to be a titanium coated stainless steel plate. The steel support plates of the purchased heat exchanger were substituted for aluminum as the stage temperatures are less than 80°C. Aluminum corrosion at higher temperatures with the system maximum salinity of 80 TDS has been observed in the literature.

5.1.3 Single Stage

A transparent single stage MED has been constructed for heat exchanger performance characterization, sensitivity to input heat, flow visualization of the permeate vapor flow path within the vacuum vessel and wetting of the heat exchanger plates. Differential pressure transducers between the heat exchanger inlet and outlet on the steam and seawater sides measure the pressure drop across the condenser and evaporator, respectively. The overall heat transfer coefficient is determined by the change in temperature of the seawater and ratio of permeate to seawater inlet mass flow rates. The single stage experimental setup with instrumentation locations is given in Figure 5.2. The thermal measurement system uses cold junction compensation and was calibrated at the system level using a high precision thermometer. Temperatures can be measured within 0.1°C accuracy, enabling better understanding of the boiling point elevation.

The sealing of the acrylic vessel has been a major technical challenge as acrylic-acrylic and acrylic metallic bonds are not well suited for the levels a vacuum required. A self-sealing lid with an axial seal and a backup o-ring was used and has proved to be the best method for sealing the acrylic-acrylic connections. Fusing of acrylic was attempted on the base of the housing and though it is structurally stable it is not a suitable sealant. Basic grade silicone sealant was applied to the leaking connections successfully, however the use of mechanical o-ring positive seals is recommended for future designs requiring vacuum tight connections with the acrylic.

The acrylic vessel was selected to observe the flow path of the water vapor along with visual observations of the dynamic system. Low frequency oscillations of the vapor outlet have been observed and real time monitoring of the brine pool with in the stage has been useful in setting stable experimental inputs.
The amount of steam lost to vacuum generation for mechanical vacuum, is of interest and flow visualization is planned to validate the predicted flow path of the vapor with in the vessel. The simulation considered two possible steam production paths were considered independently. Path A is evaporation from the outlet of the heat exchanger and Path B is evaporation from the brine pool surface. The steam production will be a combination of these two paths and two phase interactions were not considered. A summary of results for different configuration is given in Table 5.1 showing the port diameter has approximately a squared effect on the volumetric flow rate. A 5 mm diameter port provides acceptable flow distribution at 11 times more flow into the vapor port. A KF10 to ¼” NPT fitting would be acceptable with ID 4.8 mm. For 10 kg/hr permeate production with outlet velocity from the heat exchanger of 0.31 m/s and at a stage pressure of 30 kPa-a, the predicted mass flow distribution in each outlet is: 0.077 m²/s vapor and 0.0068 m²/s to the vacuum. The streamlines of the vapor flow within the vessel, Figure 5.3, show thought recirculation occurs the majority of the exits the permeate port.

5.1.4 Four-Stage

The 10 kWt CSP-DSW MED system is comprised of four stages and will be integrated into a laboratory setup for overall steady state and transient response characterization before being integrated into the full CSP-DSW system. The system was designed for maximum operating conditions between 80 and 35°C, linear temperature difference between stages, and a maximum pressure drop in the evaporator of 4 kPa. The decreasing pressure from one effect to the next one allows brine to be drawn to the next effect where it flashes releasing additional amounts of vapor at the lower pressure. A common configuration is to place the stages vertically to use gravity set the pressure difference with a barometric leg. Brine and distillate are collected from effect to effect, up the last one from where they are extracted by pumps. The schematic of the four-stage MED differs slightly from commercial and more desirable process configuration due to laboratory space limitations. These differences are that the rejectate will flow from the first to the last stage and the pressure differential will be set with needle valves instead of by gravity. A flanged stainless steel pipe was modified to form the vacuum vessel and is identical for each stage.

The predicted performance of the four stage system with MED only is 1 litter of distilled water per minute (1440 kg per day) with a PR of 3.29. The process configuration of the MED experiment is shown in Figure 5.4. A TVC will be added to the laboratory configuration after heat exchanger performance characterization is complete.
5.2 Figures

Figure 5.1 (a) Alpha Laval heat exchanger; (b) Flow paths in a co-current PHE (c) Modified seawater-side gasket.

Figure 5.2 Single stage MED experimental setup
Figure 5.3 CFD results of flow distribution inside the MED stage
Figure 5.4 Four-stage MED schematic with proposed operational conditions
5.3 Tables

Table 5.1 Results from single phase CFD flow path distribution analysis for varied flow path and vacuum port outlet diameter

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<th>Dia. (mm)</th>
<th>Flow (m^2/s)</th>
<th>Volumetric Flow (m^3/s)</th>
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Table 5.2 MED four-stage system input conditions

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### Table 5.3 MED four-stage system simulation results

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<td>0.86</td>
<td>74</td>
<td>29.0</td>
<td>12.8</td>
<td>47.3</td>
<td></td>
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<tr>
<td>Stage 4</td>
<td>35.0</td>
<td>0.157</td>
<td>0.79</td>
<td>74</td>
<td>29.0</td>
<td>13.5</td>
<td>62.9</td>
<td></td>
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</table>

### Table 5.4 Operating Conditions from heat exchanger supplier

<table>
<thead>
<tr>
<th>Flow Path</th>
<th>Effect #</th>
<th>1</th>
<th>5</th>
<th>10</th>
<th>Condenser</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td>CoCurrent</td>
<td>CoCurrent</td>
<td>CoCurrent</td>
<td>CounterCurrent</td>
</tr>
<tr>
<td>Cold Side: Seawater Inlet</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Mass flow rate</td>
<td>kg/h</td>
<td>20.57</td>
<td>57.26</td>
<td>116.55</td>
<td>205.71</td>
</tr>
<tr>
<td>Temperature</td>
<td>°C</td>
<td>31.2</td>
<td>52.9</td>
<td>38.5</td>
<td>28</td>
</tr>
<tr>
<td>Salinity</td>
<td>ppt</td>
<td>42</td>
<td>75.45</td>
<td>74.13</td>
<td>42</td>
</tr>
<tr>
<td>Phase</td>
<td></td>
<td>Liquid</td>
<td>Liquid</td>
<td>Liquid</td>
<td>Liquid</td>
</tr>
<tr>
<td>Specific heat</td>
<td>kJ/kg</td>
<td>3.989</td>
<td>3.864</td>
<td>3.851</td>
<td>3.986</td>
</tr>
<tr>
<td>Max T difference achievable</td>
<td>°C</td>
<td>97.8</td>
<td>12.0</td>
<td>1.3</td>
<td>5.9</td>
</tr>
<tr>
<td>Max heat load</td>
<td>kW</td>
<td>2.23</td>
<td>0.74</td>
<td>0.17</td>
<td>1.34</td>
</tr>
<tr>
<td>Hot Side: Steam Inlet</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Mass flow rate</td>
<td>kg/h</td>
<td>17.0000</td>
<td>9.5319</td>
<td>9.5111</td>
<td>10.2619</td>
</tr>
<tr>
<td>Temperature</td>
<td>°C</td>
<td>130</td>
<td>65</td>
<td>40</td>
<td>35</td>
</tr>
<tr>
<td>Pressure</td>
<td>bar</td>
<td>1.00</td>
<td>0.25</td>
<td>0.07</td>
<td>0.06</td>
</tr>
<tr>
<td>Phase</td>
<td></td>
<td>Vapor</td>
<td>Vapor</td>
<td>Vapor</td>
<td>Vapor</td>
</tr>
<tr>
<td>Enthalpy Hlv Kj/Kg</td>
<td></td>
<td>2,012</td>
<td>2,348</td>
<td>2,409</td>
<td>2,421</td>
</tr>
<tr>
<td>Max heat available</td>
<td>kW</td>
<td>9.50</td>
<td>6.22</td>
<td>6.36</td>
<td>6.90</td>
</tr>
<tr>
<td>OHTC clean conditions</td>
<td>W/(m²*K)</td>
<td>624.00</td>
<td>896.00</td>
<td>1126.00</td>
<td>1424.00</td>
</tr>
<tr>
<td>OHTC service</td>
<td>W/(m²*K)</td>
<td>415.20</td>
<td>702.40</td>
<td>720.00</td>
<td>1401.00</td>
</tr>
<tr>
<td>LMTD</td>
<td>°C</td>
<td>15.50</td>
<td>2.90</td>
<td>0.60</td>
<td>2.70</td>
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<tr>
<td>Press. Drop Hot Side</td>
<td>kPa</td>
<td>2.70E-03</td>
<td>6.78E-03</td>
<td>4.05E-03</td>
<td>2.56E-01</td>
</tr>
<tr>
<td>Press. Drop Cold Side</td>
<td>kPa</td>
<td>3.22E-03</td>
<td>1.77E-02</td>
<td>6.18E-02</td>
<td>1.63E-01</td>
</tr>
</tbody>
</table>
CHAPTER 6: Conclusions

Two performance improvements to CSP-DSW with MED desalination have been shown: the decrease in required motive steam pressure into a TVC vacuum system though optimization of the entrainment ratio and the reduction in required motive steam temperature by means of utilizing thermal losses from the solar collection system. These improvements increase the power production of CSP-DSW dual-purpose plant while maintaining the same total heat input into the MED-TVC. A model incorporating MED, TVC and thermal harvest has been correlated within 20% to available MED-TVC plant data and used to design a 10 kWt four-stage MED system for end use in a CSP-DSW proof of principle apparatus.

The recommended overall system layout for MED in a CSP-DSW plant is a MED-TVC design utilizing low temperature thermal energy harvesting consisting of between 15 and 20 stages. Second, the use of a control system to modify they system operating conditions between hold and cold seasons, in addition to with regards to fluctuations from the solar and/or power system significantly increases the overall system output, and additional gain of 2 to 4 times the inlet steam mass. By operating the TVC away from the minimum performance: between 0.7 and 2 bar or below 0.7 and heating it with another heat source thermodynamic CSP and MED system efficiency increases are possible with possible reduction in.

Future work on the one-stage MED includes characterization and parameterization of a single stage with permeate vapor flow visualization, correlating these results with the existing water based analytical and CFD models, quantitative optical measurement of liquid phase falling film thickness and flow distribution along the heat exchanger plate. These results will provide a basis for the characterization of the performance of the four-stage MED with predicted PR of 3.29 to begin after the laboratory assembly is completed. Replacing the mechanical vacuum with low-pressure motive steam TVC MED, and finally system integration into a CSP-DSW plant will follow.
REFERENCES


APPENDIX A: TVC Optimization

Figure A.1 Flowchart of the TVC optimization algorithm.
Figure A.2 Result of the effects of compression ratio on the entrainment ratio based on input values given by Aly [25].

Figure A.3 Comparison of the exit temperatures calculated using the TVC code and found on IAPWS IF97 table.

Table A.1 Input values for the TVC code and IAPWS IF97 comparison

<table>
<thead>
<tr>
<th>ER</th>
<th>$T_s$ [°C]</th>
<th>$T_v$ [°C]</th>
<th>$p_v$ [bar]</th>
<th>$\eta_n$</th>
<th>$\eta_m$</th>
<th>$\eta_d$</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.2949</td>
<td>210.85</td>
<td>100.05</td>
<td>1.016</td>
<td>0.9</td>
<td>1</td>
<td>0.85</td>
</tr>
</tbody>
</table>
APPENDIX B: Seawater Properties

B.1 Standard Seawater Composition and Solubility

Table B.1 Ionic composition of standard seawater: 35 g/L TDS at pH 8.1 and 25°C [31].

<table>
<thead>
<tr>
<th>Ion</th>
<th>Concentration [g/L]</th>
</tr>
</thead>
<tbody>
<tr>
<td>Cl⁻</td>
<td>19</td>
</tr>
<tr>
<td>Na⁺</td>
<td>10.5</td>
</tr>
<tr>
<td>SO₄²⁻</td>
<td>2.7</td>
</tr>
<tr>
<td>Mg²⁺</td>
<td>1.35</td>
</tr>
<tr>
<td>Ca²⁺</td>
<td>0.4</td>
</tr>
<tr>
<td>HCO₃⁻</td>
<td>0.142</td>
</tr>
<tr>
<td>K⁺</td>
<td>0.38</td>
</tr>
<tr>
<td>CO₃⁻</td>
<td>0.0035</td>
</tr>
<tr>
<td>Br⁻</td>
<td>0.065</td>
</tr>
<tr>
<td>Total</td>
<td>34.54</td>
</tr>
</tbody>
</table>

Figure B.1  Common seawater mineral scalant precipitation as a function of temperature
B.2 Seawater Enthalpy

For temperature in Kelvin and a molar mass of seawater of $M_{SS}=62.793$ and an average charge of +2 the following correlations are used for seawater enthalpy. The molarity, $m$, and ionic strength, $I$, are related to the salinity:

\[
m = \frac{16.011S}{1000 - 1.00488S} \quad (48)
\]
\[
I = \frac{19.9243S}{1000 - 1.00488S} \quad (49)
\]
\[
A_L = 1.533198 \times 10^6 + 9.928964 \times 10^2 T - \frac{2.998991 \times 10^7}{T} - 2.988147 \times 10^5 \ln T + \frac{4.5587696 \times 10^2}{T - 263} - 6.372429 \times 10^{-1} T^2 + \frac{1.241133 \times 10^7}{680 - T} \quad (50)
\]
\[
x = 2I^{0.5} \quad (51)
\]
\[
g = \frac{2(1 - (1 + x)e^{-x})}{x^2} \quad (52)
\]

\[
\beta_{SS0}^L = \frac{19681.228}{T^2} - 2.78154 \times 10^{-2} T + 3.09899 \times 10^{-4} T^2 - 1.481241 \times 10^{-6} T^3 + 3.672820 \times 10^{-9} T^4 - 4.403776 \times 10^{-12} T^5 + 8.557563 \times 10^{-16} T^6 + 3.252522 \times 10^{-18} T^7 - 2.35678 \times 10^{-21} T^8 \quad (53)
\]
\[
\beta_{SS1}^L = \frac{809.0}{T^2} - 6.383 \times 10^{-5} T + 1.250 \times 10^{-7} T^2 \quad (54)
\]
\[
C_{SS}^L = \frac{29.06}{T^2} - 3.133 \times 10^{-6} T + 6.10 \times 10^{-9} T^2 \quad (55)
\]
\[
B_{SS}^L = \beta_{SS0}^L + \beta_{SS1}^L g \quad (56)
\]
\[
\Delta h_{SS} = \frac{S}{M_{SS}} \left( \frac{A_L I}{1.2 M} \log(1 + 1.2 I^{0.5}) - 2RT^2 M (B_{SS}^L + MC_{SS}^L) \right)
\]
B.3 Boiling Point Elevation

Where $T$ is the temperature in °C and $c$ is the seawater concentration in kg/kg. The equation below is valid for the boiling point elevation of seawater with accuracy: ±0.018°C over the ranges: 0<S<0.120, 0<T<200°C.

$$BPE = c(B + cC)10^{-3} \quad (57)$$

$$B = (6.71 + 6.43 \times 10^{-2}T + 9.74 \times 10^{-5}T^2) \times 10^{-3} \quad (58)$$

$$C = (22.238 + 9.59 \times 10^{-3}T + 9.42 \times 10^{-5}T^2) \times 10^{-8} \quad (59)$$
APPENDIX C: Parametric Study Results

Figure C.1 Overall system performance, PR, as a function of Seawater intake temperature for final stage temperature constraint of 10°C above the intake temperature.

Figure C.2 Heat transfer area per product volume as a function of Seawater intake temperature for final stage temperature constraint of 10°C above the intake temperature.
Figure C.3 Heat transfer area per product volume as a function of seawater intake temperature for final stage temperature constraint of $5^\circ$C above the intake temperature.

Figure C.4 Heat transfer area per product volume as a function of Seawater intake temperature for final stage temperature constraint of $35^\circ$C.
Figure C.5 Detailed view of PR versus motive steam pressure for regions where first stage temperature was lowered in the TVC optimization algorithm

Figure C.6 Detailed view of specific heat exchanger area versus motive steam pressure for regions where first stage temperature was lowered in the TVC optimization algorithm
Figure C.7 Detailed view of first stage temperature reduction by the TVC optimization algorithm during motive steam parameterization.
APPENDIX D: 10 kWt Stage Drawings

Figure D.1 Four-Stage MED Flange Bottom Drawing
Figure D.2 Four-Stage MED Flange Top Drawing
Figure D.3 Four-Stage MED Stage Housing Nipple Drawing
Figure D.4 Four-Stage MED Stage Assembly Drawing
Figure D.4 (Cont.)