

ABSORPTION COOLING FOR DIFFUSION DRIVEN DESALINATION PROCESS

By

UDAY KIRAN MAHAKALI

A THESIS PRESENTED TO THE GRADUATE SCHOOL
OF THE UNIVERSITY OF FLORIDA IN PARTIAL FULFILLMENT OF THE
REQUIREMENTS FOR THE
DEGREE OF MASTER OF SCIENCE

UNIVERSITY OF FLORIDA

2011

© 2011 Uday Kiran Mahakali

This work is dedicated to my parents and sister who have supported me in all my endeavors.

ACKNOWLEDGMENTS

First, I would like to thank Dr. James F Klausner, the chairman of my graduate committee, for allowing me to work under his guidance and for granting me the opportunity to be a part of his research team on the diffusion driven desalination process. I am grateful to Dr. Klausner for his constant guidance and patience throughout the two years of my master degree studies. I would also like to thank Dr. H A Ingley, member on my graduate committee, for his technical assistance on the absorption cooling system. I also extend my thanks to Dr. S A Sherif for being part of my graduate committee and for his valuable time and attention.

I would also like to thank all my friends who provided me with moral and academic support throughout my graduate studies. I extend my special thanks to Dr. Fadi Alnaimat for his crucial technical assistance.

Above all, I would like to thank my family for their unwavering faith in me, love and support. Their blessings and encouragement gave me the motivation to achieve my goals.

TABLE OF CONTENTS

	<u>page</u>
ACKNOWLEDGMENTS.....	4
LIST OF TABLES.....	7
LIST OF FIGURES.....	8
LIST OF KEY SYMBOLS	9
ABSTRACT	12
CHAPTER	
1 INTRODUCTION	14
2 DIFFUSION DRIVEN DESALINATION.....	19
2.1 Description of Conventional DDD Plant	19
2.2 Diffusion Desalination Process with Ammonia Absorption System.....	22
2.2.1 Requirement of Ammonia Absorption System.....	22
2.2.2 Description of DDD Plant with Ammonia Absorption System	23
3 REFRIGERATION SYSTEM.....	25
3.1 Conventional Vapor Compression Refrigeration.....	25
3.2 Ammonia Absorption Refrigeration System	26
3.2.1 Principle of Operation.....	27
3.2.2 Characteristics of Ammonia Absorption.....	31
3.2.3 Factors Affecting COP of Ammonia Absorption.....	32
3.2.4 Advantages of Ammonia Absorption Refrigeration System	32
4 MATHEMATICAL MODELLING OF DDD PROCESS.....	34
4.1 Flow in the Diffusion Tower.....	36
4.2 Flow in the Condenser Tower	37
4.3 Numerical Procedure	40
5 RESULTS AND DISCUSSION	41
5.1 Diffusion Tower Analysis.....	41
5.2 Condenser Tower Analysis	45
6 INDUSTRIAL APPLICATION OF DDD PLANT WITH AAR SYSTEM	56
7 CONCLUSIONS	63

APPENDIX

A	ONDA'S CORRELATION	65
B	CO-GENERATION PLANT DETAILS	66
C	NUMERICAL ANALYSIS RESULTS.....	69
D	INDUSTRIAL APPLICATION NUMERICAL ANALYSIS RESULTS.....	73
	LIST OF REFERENCES	74
	BIOGRAPHICAL SKETCH.....	76

LIST OF TABLES

<u>Table</u>		<u>page</u>
C-1	List of results from diffusion tower analysis	69
C-2	List of results from condenser tower analysis (without AAR system).....	70
C-3	List of results from condenser tower analysis (with AAR system).....	71
C-4	List of results from condenser tower analysis	72
D-1	List of results from diffusion tower analysis	73
D-2	List of results from condenser tower analysis	73

LIST OF FIGURES

<u>Figure</u>	<u>page</u>
2-1 Typical diffusion driven desalination plant.	20
2-2 DDD plant with ammonia absorption refrigeration system.	24
3-1 Ammonia absorption refrigeration system.	28
4-1 Differential control volume for liquid/gas heat and mass transfer within diffusion tower.	34
4-2 Differential control volume for liquid/gas heat and mass transfer within counter current condenser.	35
5-1 Diffusion tower exit air temperature variation with air to feed water mass flow ratio.	43
5-2 Diffusion tower exit feed water temperature variation with air to feed water mass flow ratio.	43
5-3 Diffusion tower exit humidity ratio variation with air to feed water mass flow ratio.	44
5-4 Condenser tower exit air temperature variation with fresh water feed to air mass flow ratio.	48
5-5 Condenser tower exit fresh water temperature variation with fresh water feed to air mass flow ratio.	50
5-6 Condenser tower exit humidity ratio variation with fresh water feed to air mass flow ratio.	51
5-7 Fresh water production variation with fresh water feed to air mass flow ratio.	53
5-8 Percentage increase in fresh water production variation with fresh water feed to air mass flow ratio.	55
6-1 Variation of fresh water produced with x	60
6-2 Variation of required area for diffusion and condenser towers with x	62
B-1 Co-generation plant.	66

LIST OF KEY SYMBOLS

A	Cross sectional area	m^2
a	Specific surface area	m^2/m^3
a_w	Wetted specific surface area	m^2/m^3
C_p	Specific heat	kJ/kg
D	Molecular Diffusion coefficient	m^2/s
d_p	Diameter of packing bed	m
G	Air mass flux	$\text{kg/m}^2\text{-s}$
h	Enthalpy	kJ/kg
h_{fg}	Latent heat of vaporization	kJ/kg
k	Mass transfer coefficient	m/s
L	Water mass flux	$\text{kg/m}^2\text{-s}$
m_f	Mass flux of fresh water produced	$\text{kg/m}^2\text{-s}$
M_v	Molecular weight of vapor	kg/kmol
m	Mass flow rate	kg/s
P	Pressure	kPa
Q	Power / Rate of energy	kW
R	Universal gas constant	kJ/kg-K
T	Temperature	$^{\circ}\text{C}$ or K
U	Heat transfer coefficient	$\text{W/m}^2\text{-K}$
x	The fraction of exhaust air going into refrigeration system	
z	Height of packing material	m

Greek letters

μ	Dynamic Viscosity	Pa-s
ρ	Density	kg/m ³
σ_c	Critical surface tension of the packed bed	N/m
σ_L	Liquid surface tension	N/m
ω	Humidity ratio	
ϕ	Relative Humidity	

Subscripts

a	Air	
ab	Absorber	
AAR	Ammonia absorption Refrigeration	
cf	Cooling fresh water	
cn	Condenser	
CL	Cooling load	
DDD	Diffusion driven desalination	
ea	Exhaust air	
ev	Evaporator	
fw	Fresh water	
ge	Generator	
G	Air/Vapor mixture	
GA	Gas side parameter based on the specific area of the packed bed	
i	Liquid/Vapor interface	
in	Inlet	

<i>L</i>	Liquid
<i>LA</i>	Liquid side parameter based on the specific area of the packed bed
<i>LW</i>	Liquid side parameter based on the specific wet area of the packed bed
<i>out</i>	Outlet
<i>s</i>	Available for refrigeration system
<i>sat</i>	Saturated state
<i>sw</i>	Saline water

Abstract of Thesis Presented to the Graduate School
of the University of Florida in Partial Fulfillment of the
Requirements for the Degree of Master of Science

ABSORPTION COOLING FOR DIFFUSION DRIVEN DESALINATION PROCESS

By

Uday Kiran Mahakali

August 2011

Chair: James F. Klausner
Major: Mechanical Engineering

In order to make the diffusion driven desalination (DDD) process commercially more attractive, the fresh water production from the process must be increased. This research investigates the possible increase in the fresh water production from the DDD process when the condenser fresh water inlet temperature is reduced. The reduction in the temperature of fresh water is achieved by a single effect ammonia absorption refrigeration system which can be run on waste heat like the DDD process. The heat and mass transfer analysis for the DDD process is utilized and simulations are carried out for the heated air / heated water case for different air and water mass fluxes. Simulations are performed for two condenser feed water inlet temperatures of 25°C and 1°C, for the same diffusion tower outlet conditions. The results show an increase in the fresh water production. A conceptual design of the DDD plant coupled with an ammonia absorption refrigeration system working on waste heat from a power plant is conceived. A parametric study is made in which the DDD plant and the ammonia absorption refrigeration system are powered by the waste heat available at 93°C from a 271 MW solar combined cycle power plant. The *COP* of the ammonia absorption refrigeration system operating on the waste heat under the available conditions is calculated. Results

indicate that due to the low *COP* of the refrigeration system, there is a higher fresh water production when the entire available waste heat is utilized by the DDD plant itself, and the ammonia absorption refrigeration system is not used.

CHAPTER 1 INTRODUCTION

Water is an indispensable part of everyday human life. Fresh water is needed for the biological activities of living organisms. Agriculture requires fresh water for crop cultivation. Industries require fresh water for washing, cooling, fabrication and processing. Approximately, 70% the Earth's surface is covered by a water body and 30% is composed of land. Unfortunately, of water available on the Earth, 97% is salt water and only 3% is fresh water [1]. The salt water has very limited use for mammalian life support. The 3% of fresh water is not completely available for ready use by mammals since 69% is trapped in the form of ice and 30% resides under the ground as ground water. The remaining 1% is available on the surface. Of this 1%, only 0.3% is present in the rivers and lakes, which can be easily used by mammals. The remaining 0.7% is either present in the atmosphere as vapor or some other unusable form. Given the limited availability of fresh water and its role in mammalian life, it should be treated as a very precious resource.

The increase in population of humans and the proportional increase in their need for fresh water have led to shortages of fresh water supply around the world. The scientific and engineering communities are actively developing new technologies to address this shortage. One such popular method is Desalination.

Many countries such as Saudi Arabia, United Arab Emirates, and Kuwait depend on desalination technologies to meet their fresh water requirements. According to an IDA Worldwide Desalting Plant Inventory Report [2], in 2009 approximately 1.7 billion gallons per day of fresh water was being produced by desalinating salt and brackish water in the desalination plants commissioned all over the world.

Desalination involves the conversion of the abundantly available salt and brackish water to consumable fresh water. This is achieved by removing the salt and minerals from the water. The desalination process sometimes yields sea salt as a by-product. There are many different processes available for desalinating salt or brackish water [3] such as processes based on phase change, processes using membranes, and processes based on modifying the chemical bonds. Some of the commercially popular methods [3] used for desalination are Vacuum Distillation, Low Temperature Thermal Desalination, Multistage Flash Distillation, Multiple Effect Distillation, Electrodialysis and Reverse Osmosis. Multistage Flash Distillation is the most widely used method while the Reverse Osmosis method is gaining in popularity. The Low Temperature Thermal Desalination method is a relatively new one. Some other methods of desalination are solar humidification-dehumidification, freezing, renewable energy powered conventional desalination, among others.

Although Multistage Flash Distillation and Reverse Osmosis are commercially popular and reliable methods, there are certain disadvantages. For example, thermal distillation is economically feasible only for very large scale production, typically more than 300-400 kL/day [3]. Thermal distillation is very energy intensive and requires a high level of technical knowledge to build, operate and maintain efficient plants. Reverse Osmosis technology is gaining in popularity as the cost for equipment has reduced in recent years due to mass production. However, the performance and longevity of a Reverse Osmosis plant is significantly dependent on the pre-treatment of the feed water, which is expensive. Also, there is a danger of contamination of the product due to the growth of bacteria on the membranes. High pressures are required to operate the

Reverse Osmosis plant, and the high pressures lead to an increase in frequency of plant shut down due to failures in the mechanical equipment providing the high pressures.

In an attempt to find a lower cost desalination process, Humidification - Dehumidification (HDH) was studied by Bourouni et al. [4]. According to Bourouni, it is a simple and flexible process utilizing low grade heat energy. It is described as a process involving low initial cost, lower operation and maintenance costs. Many novel methods like the Multiple Effect Humidification process, Mechanically Intensified Evaporation process which are based on the principle of HDH process, have been developed to carry out desalination. However, each of them has its own disadvantages. For example, the HDH process is unsuitable for the economical fresh water production on a large scale when compared against Reverse Osmosis or Multistage Flash Distillation methods because it requires a very large area of land.

In order to overcome the shortcomings of the HDH desalination method mentioned above, an innovative desalination method known as the Diffusion Driven Desalination (DDD) process was studied by Klausner et al. [5]. The DDD process provides an economically feasible desalination method suitable for desalinating saline water on a large scale. The DDD process, like the HDH method, can be run on low grade waste heat. In the DDD process, air is pumped into a diffusion tower and is made to contact saline or brackish water flowing in the opposite direction along the height of the tower. Heat transfer and mass transfer occur between the air and saline or brackish water and as a result only the fresh water vaporizes leaving behind salts and other minerals. The air exits the diffusion tower humidified. This humidified air is then pumped

into another similar tower known as the condenser tower. In this tower, it is made to contact fresh water flowing in opposite direction, which is at a lower temperature than the air entering the condenser tower. Heat transfer occurs and the water vapor condenses out of the air stream and joins the fresh water flowing in the tower. Thus the mass of fresh water at the tower exit is greater than that at the tower inlet, the difference being the fresh water production. Also, the air exits the condenser tower dehumidified. The DDD process which has been described in detail by Li et al. [6] is presented in the chapter 2 of this report.

It is known that as the temperature of air decreases, its capacity to hold water vapor also decreases, and the water vapor condenses out. This is the principle utilized in the condenser tower of the DDD process to extract fresh water from the air stream. Currently, the amount of fresh water that can be extracted from the air stream is limited by the temperature of the fresh cooling water being pumped into the condenser tower. This temperature is the ambient temperature of fresh water available, which is typically 25°C. In the current work, this temperature is brought down to 1°C, which is the lowest possible temperature for the safe circulation of water without forming ice. Utilizing the heat and mass transfer analysis available for the DDD process which is developed by Klausner et al. [7], the amount of fresh water produced when the fresh cooling water temperature is 1°C has been calculated for various mass flux combinations of air and water. Also, the increment in the amount of fresh water produced due to the reduction in the fresh cooling water temperature from 25°C to 1°C is also presented.

The main advantage of the DDD process compared with commercially available processes is that it can run on low grade heat energy. Therefore, in an effort to keep up

with this advantage, the fresh cooling water is cooled to 1°C by an Ammonia Absorption Refrigeration (AAR) plant, which can also be run on low grade energy. The results of a study involving a 271 MW solar combined cycle electric power generation plant, together with the DDD system and an AAR plant running on shared waste energy from the power generation plant are presented in this work. The optimum operating condition for the DDD process as part of this co-generation plant is also discussed.

CHAPTER 2 DIFFUSION DRIVEN DESALINATION

2.1 Description of Conventional DDD Plant

The conventional diffusion driven desalination process has been developed by Klausner et al. [5]. A laboratory scale facility is currently in operation at the University of Florida, Gainesville. A brief description of the operation of the DDD plant is given below.

It is possible to run the DDD plant in three modes:

1. Ambient Air and Heated Water
2. Heated Air and Ambient Water
3. Heated Air and Heated Water

In the Ambient Air and Heated Water mode, the air that is passed into the diffusion tower is at atmospheric temperature while the sea water that is passed into the diffusion tower is heated to a temperature greater than that of the atmospheric air. In the Heated Air and Ambient Water mode, the air that is passed into the diffusion tower is heated to a higher temperature than that of the sea water that is passed into the diffusion tower, which is at atmospheric temperature. In the Heated Air and Heated Water mode, both air and water that are passed into the diffusion tower are heated to higher temperatures than the atmospheric temperature at which they are available. The following description on the operation of the DDD plant is based on the Ambient Air and Heated Water operating mode.

Figure 2-1 shows a process flow diagram for a typical DDD facility. There are two towers and three fluid circulation systems: Diffusion Tower, Condenser Tower, Saline Water, Air/Vapor and Fresh Water.

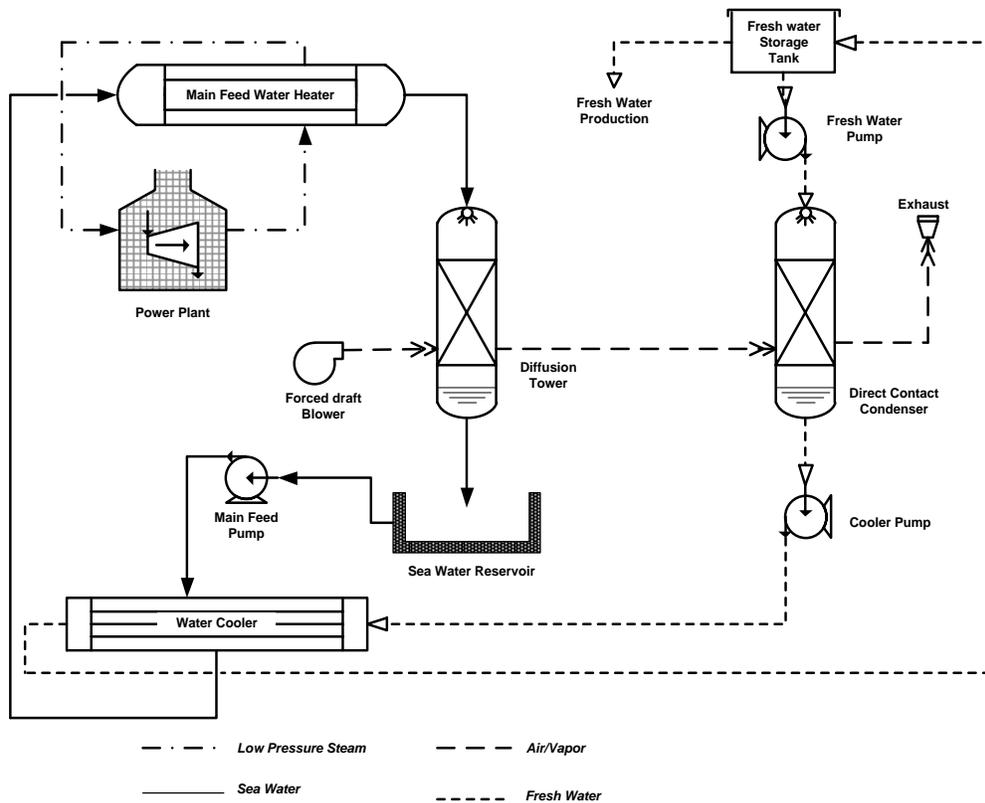


Figure 2-1. Typical diffusion driven desalination plant.

Sea water from the surface or from shallow depths is taken, as it is warmer than the water which is deeper. This water is pumped into the water cooler which is a heat exchanger. The sea water is preheated in the water cooler by the fresh water discharge from the condenser. It is further heated to a higher temperature using waste heat of the low pressure condensing steam from a thermal power plant or from another source of waste heat. Waste heat can be utilized because the required feed water inlet temperature into the diffusion tower can be as low as 50°C for the DDD process. This heated water is then sprayed in the diffusion tower from the top. Simultaneously atmospheric air is forced into the diffusion tower from the bottom by a forced draft blower. A portion of the sea water evaporates and diffuses into the air rapidly. The evaporation in the diffusion tower is governed by the concentration gradient at the

liquid/vapor interface and bulk air, according to Fick's law. The diffusion tower consists of a low pressure drop and high surface area packing material. The packing material is made from polypropylene. Polypropylene has a very low cost and is inexpensive to replace [8]. A great portion of the heat and mass transfer occurring in the diffusion tower takes place in the packed bed. The sea water sprayed from the top of the diffusion tower travels downwards through the tower by means of gravity and passes over the packing material forming a thin layer of saline water on it. This film of water contacts the air which is flowing upwards in the diffusion tower facilitating the heat transfer between the saline water and air. The height and diameter of the diffusion tower is chosen so that the air entering the diffusion tower leaves it in a saturated condition. The portion of saline water that is not evaporated is now at a lower temperature due to evaporation and heat transfer with air. This saline water is collected at the bottom of the diffusion tower and discharged into a sea water reservoir. The saturated air exiting the diffusion tower is passed through a piping system and enters the counter current condenser tower from the bottom where it is cooled and dehumidified.

The condenser tower is a direct contact condenser with counter-current flow [9]. As there is a large fraction of air/vapor mixture that is non-condensable [8], direct contact condensation is more effective than film condensation as concluded by Bharathan et al. [10]. In the DDD plant, a packed bed condensation approach is utilized in the direct contact condenser as it is found to be more effective than droplet direct contact condensation [9]. The packing material in the condenser tower is similar to that in the diffusion tower. The fresh water is collected at the bottom of the condenser tower and is pumped by a cooler pump through the water cooler where it is cooled to a lower

temperature than the air/vapor mixture from the diffusion tower exhaust. This cool fresh water is sent into the fresh water storage tank. A certain mass of this fresh water in the storage tank is pumped by a fresh water pump and is sprayed at the top in the condenser tower. The remaining water in the storage tank is taken out as fresh water production. The saturated air/vapor mixture from the diffusion tower exhaust, which is at a higher temperature than the fresh water feed is forced from the bottom of the condenser tower. It meets the fresh water feed in the condenser tower and most of the heat and mass transfer occurs within the packed bed. The water vapor in the air/vapor mixture condenses out due to lowering its temperature along the saturation line, and the condensed fresh water is taken out along with the fresh water feed at the bottom of the condenser tower. The fresh water is collected and sent through the water cooler again to be used as fresh water coolant.

2.2 Diffusion Desalination Process with Ammonia Absorption System

2.2.1 Requirement of Ammonia Absorption System

Thermal analysis of the DDD process suggests that one way to improve the fresh water production is to lower the fresh water temperature into the condenser as low as possible. The lowest possible temperature of the fresh water feed that is possible is just above 0°C. Reducing the fresh water temperature further will result in freezing. So, a safe and stable operating fresh water feed temperature of 1°C is chosen. In order to bring about the large reduction in the fresh water temperature, a refrigeration system is required. In line with the objective of running the entire DDD plant on waste heat, an Ammonia Vapor Absorption Refrigeration System is used in place of the water cooler.

2.2.2 Description of DDD Plant with Ammonia Absorption System

The functioning of the DDD plant remains exactly the same as that already described. The waste heat from a combined cycle solar power plant is divided between the Desalination plant and the Ammonia Absorption Refrigeration system. The fresh water sent into the condenser tower is passed through the evaporator of the ammonia absorption refrigeration system in order to reduce its temperature to 1°C. The sea water, exiting the diffusion tower can be used as a coolant for the condenser or absorber in the ammonia absorption refrigeration system. This way the sea water feed is preheated and it reduces the amount of heat to be added in the main feed water heater. Alternatively, the exhaust air from the condenser tower can also be used as a coolant. The process flow design depicted in Figure 2-2 uses the latter scheme. A choice between the two possibilities is made depending on the operating conditions. In some cases, even the fresh water that is produced and stored in the tanks can be utilized as a coolant. Figure 2-2 shows a diagram of the DDD plant with an ammonia absorption refrigeration system, depicting the different components and fluids involved in the working of a DDD plant. A detailed description of the working of ammonia absorption refrigeration system is presented in the next chapter. Details regarding the completely solar combined cycle power generation plant are also discussed in this report.

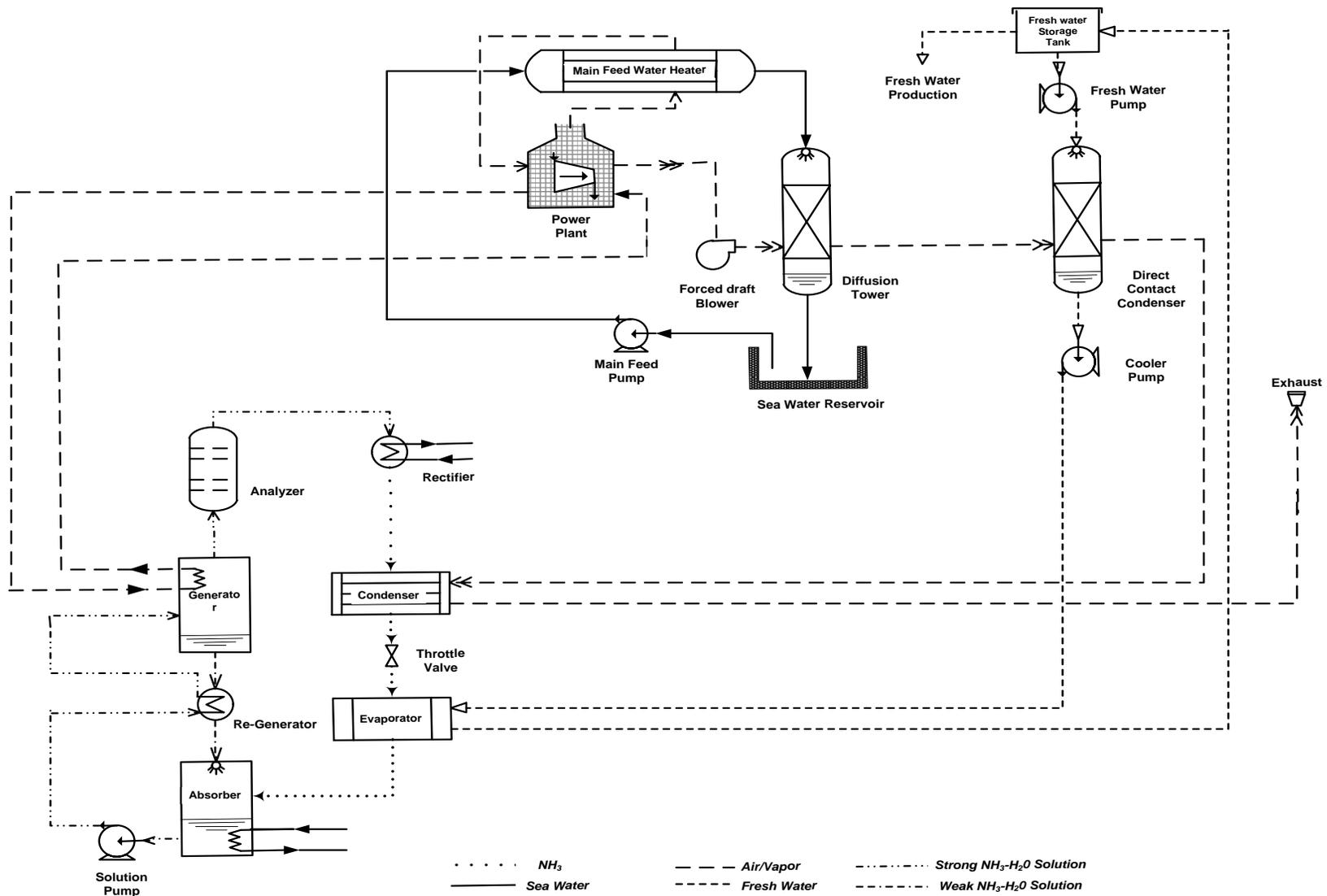


Figure 2-2. DDD plant with ammonia absorption refrigeration system.

CHAPTER 3 REFRIGERATION SYSTEM

There are various refrigeration methods available for rejecting heat at low temperature, such as Vapor Compression, Vapor Absorption, Gas Cycle, and Stirling Cycle. The Vapor Compression system is the most widely used system for refrigeration. The Gas cycle is not very efficient compared to Vapor Compression. The Stirling Cycle is too complex to be implemented at a competitive cost. The Vapor Compression system requires a compressor to function, and the compressor requires high quality energy to run it. The objective of this study is to run the entire desalination plant on waste heat. Of these currently available refrigeration methods, the Vapor Absorption system can be run on waste heat commonly discharged by industrial users. It can also be run on solar energy. Its use with the DDD process will be investigated in further chapters.

3.1 Conventional Vapor Compression Refrigeration

A brief description of the conventional vapor compression refrigeration system is provided here in order to emphasize the main differences in construction and operation between this cost effective and relatively efficient refrigeration system and the ammonia absorption system, which is the subject of importance in this thesis.

The conventional vapor compression refrigeration system usually consists of a compressor, a condenser, an expansion device, an evaporator and a working fluid called refrigerant. The compressor requires mechanical energy to drive the shaft. It converts this mechanical energy into an increase in the pressure potential and thermal heat storage in the refrigerant. It also helps to circulate the refrigerant through the entire system. The compressor can be a reciprocating or rotary type. The reciprocating type

provides higher pressure ratios than the rotary type but is less efficient and noisier. The compressed refrigerant which is at a higher pressure and temperature enters the condenser from the compressor. The condenser is a heat exchanger. The refrigerant rejects heat to a coolant and this brings down its temperature, but the pressure is maintained almost the same. However, the refrigerant pressure might reduce to some extent in the condenser owing to flow losses. The condenser is typically a shell and tube type or finned tube type. The coolant is usually air or water. The refrigerant then flows into an expansion device. The refrigerant loses its pressure as it expands across the device and thus the temperature significantly reduces.

The rate of refrigerant flow in the system can be controlled by this expansion device. It is typically a small orifice, capillary tube, or a thermostatic expansion valve. The cooling load taken by the refrigeration system i.e. the amount of cooling depends on the rate of refrigerant flow. The minimum temperature that can be reached by using the system also depends on the type of refrigerant and the amount of expansion. There are various types of refrigerants available for different applications and to reach different temperatures. A refrigerant should be non-toxic, less damaging to the environment, economically viable and easily available. This low pressure and low temperature refrigerant enters into evaporator which is the refrigerated space. This component is a heat exchanger.

3.2 Ammonia Absorption Refrigeration System

Ammonia has a great affinity for water. This property is the basis for the working of an ammonia vapor absorption system. The ammonia vapor absorption refrigeration (VAR) system was invented in 1850 by Ferdinand P E Carre. It utilizes ammonia and water as operating fluids. It has been patented in the USA in 1860. In the early days,

machines based on this basic design were used in industrial refrigeration for storing food and making ice. In the 1950s a new vapor absorption refrigeration system using lithium bromide and water as operating fluids was introduced. As concluded by Horuz [11], the VAR system operating on lithium bromide and water is more efficient than that operating on ammonia and water. However, the danger of crystallization and the impossibility of operating at sub-zero temperatures, due to the usage of water as a refrigerant render the lithium bromide VAR system unsuitable for the present application.

The coefficient of performance is a measure of a cycle's ability to transfer heat between different temperatures. The coefficient of performance (*COP*) [12] of a VAR is:

$$COP = \frac{\text{Cooling capacity of evaporator}}{\text{Heat input for the generator} + \text{Work input for the pump}} \quad (3.1)$$

The *Carnot COP* for the system is the maximum possible performance that can be achieved and for the VAR system:

$$Carnot\ COP = \frac{(T_{ge} - T_{ab})}{T_{ge}} \frac{T_{ev}}{(T_{cn} - T_{ev})} \quad (3.2)$$

where, T_{ge} (K) is the temperature in the generator, T_{ab} (K) is the temperature in the absorber, T_{cn} (K) is the temperature in the condenser and T_{ev} (K) is the temperature in the evaporator of the VAR system.

3.2.1 Principle of Operation

In an ammonia-water vapor absorption refrigeration system, ammonia is the refrigerant and water is the absorbent. The vapor absorption refrigeration system consists of an absorber, a solution pump, a re-generator, a generator, an analyzer, a rectifier, a condenser, a receiver, an expansion device and an evaporator as shown in Figure 3-1.

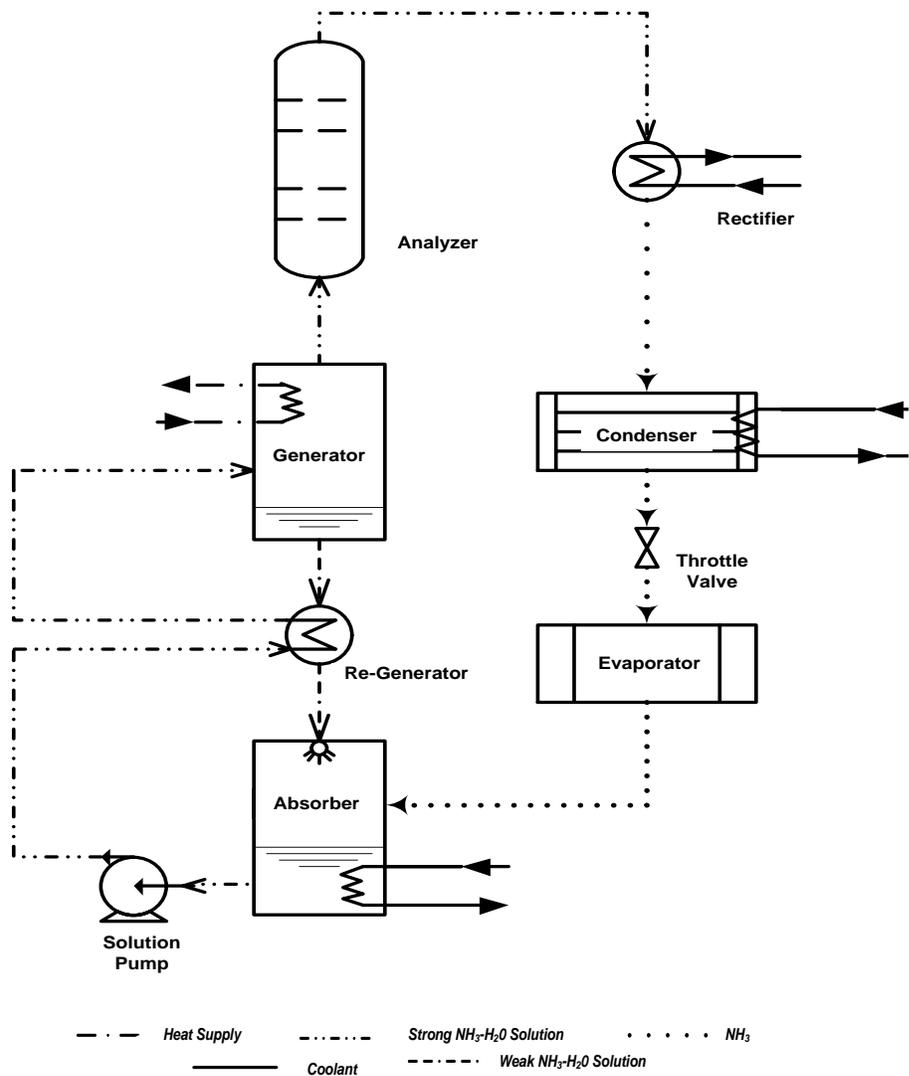
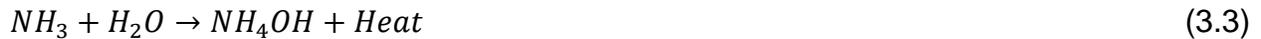


Figure 3-1. Ammonia absorption refrigeration system.

Pure ammonia in a gaseous state is sent into the absorber from the evaporator. In the absorber, the pure gaseous ammonia comes into contact with the water which absorbs ammonia. This absorption process is exothermic. However, the concentration of ammonia in water increases with decreasing temperature. Therefore in order to have maximum absorption of ammonia by water, the absorber temperature is maintained low

by removing the heat released due to the reaction by circulating cooling water. The following reaction occurs in the absorber:



The strong solution of ammonia in water exiting the absorber is pumped into the generator via the re-generator. Generally a centrifugal pump or a diaphragm pump is used for this purpose. The energy consumed by this pump is very low compared to the heat energy supplied to the generator, usually on the order of 0.25%. The pump also raises the pressure of the Ammonia -Water strong solution. This strong solution is passed through a re-generator which is a heat exchanger so as to preheat the solution before entering into the generator. This reduces the amount of thermal energy supplied to the generator and helps in improving the system *COP*. This strong solution at higher pressure is then sent into the generator which is also a heat exchanger. Here, the heat energy is supplied from sources like waste heat from a thermal power plant or a renewable energy source like solar energy. The temperature at which heat is supplied is generally above 85°C. In the generator, the heat energy supplied is used to raise the temperature of the strong Ammonia-Water solution. The solubility of ammonia in water decreases with increase in temperature and the pure ammonia vapor separates out from the solution leaving a weak solution of Ammonia-Water behind. Ideally only pure ammonia should leave the generator, but in practice the heat energy supplied also vaporizes some water. So, a mixture of Ammonia-Water vapor leaves the generator. When water vapor is also carried into the evaporator, it reduces the performance of the system. In order to remove the water vapor in the mixture, it is passed through an analyzer and then through a rectifier. The analyzer is placed on top of the generator and

it generally consists of a distillation column. It contains a number of horizontal plates along its length. When the Ammonia-Water Vapor mixture enters the analyzer, it rises up and cools down. The boiling point of water being higher than that of ammonia, it condenses first and is collected in the bottom of the analyzer. The ammonia is still in gaseous phase. A stronger mixture of Ammonia-Water vapor mixture exits the analyzer from the top and passes into the rectifier in which further removal of water particles from the mixture occurs and nearly pure ammonia in the gaseous state exits the rectifier. The rectifier is a heat exchanger which is cooled externally by a coolant. This further reduces the temperature of the Ammonia-Water vapor mixture, and water particles condense out along with few ammonia particles. This mixture is then sent into the analyzer and is collected at the bottom along with the analyzer exhaust. In both the rectifier and the analyzer, the weak Ammonia-Water mixture drains down the system into the generator by gravity. This is mixed with the strong Ammonia-Water mixture in the generator and further heated to generate more ammonia vapor. Finally, the weak Ammonia-Water mixture is drained down from the generator by gravity and is sprayed into the absorber from the top. The weak mixture is passed through the re-generator before being sprayed into the absorber, where it is pre-cooled losing its heat to pre-heat the strong Ammonia-Water mixture. From the re-generator, it is passed through a valve where it is further expanded lowering its temperature and finally into the absorber. The pure ammonia gas which is at a high pressure and temperature at the exit of the rectifier now enters into the condenser which is a heat exchanger similar to that in the vapor compression system. In the condenser pure ammonia gas rejects heat to a coolant and it condenses into liquid. The pressure remains almost constant barring minor flow

losses. The liquid ammonia is then sent through an expansion device where it expands suddenly and significantly drops in pressure, thereby further lowering its temperature. It is through this valve that the flow rate of ammonia and the amount of expansion is controlled which in turn determines the amount of cooling load and the minimum temperature that can be attained through the refrigeration system. The liquid ammonia which is at a very low pressure and temperature is then sent into the evaporator which is the space to be refrigerated. The ammonia absorbs heat from the refrigerated space and changes its phase into vapor. This low pressure but higher temperature gaseous ammonia is then sent into the absorber for absorption by water and the cycle repeats.

3.2.2 Characteristics of Ammonia Absorption

The ammonia absorption refrigeration system has many characteristics that make it suitable for use with low grade waste heat. Some of these characteristics are discussed here.

The mechanical compression present in the conventional vapor compression refrigeration (VCR) system is replaced by chemical-thermo compression in the VAR system. As mentioned earlier, the VAR system can run on waste heat, unlike the VCR system which requires electrical energy to operate. The VAR system converts the vapor from the evaporator exit back into a liquid using a method that only requires heat, which is low grade energy compared to electrical energy which is required for the VCR system. To operate the VAR system, the source temperature can be relatively low, although a higher source temperature improves the system *COP*. The mechanical compressor from a conventional vapor compression refrigeration system is replaced by an absorber, a pump and a generator in the VAR system. The absorber and generator have no moving parts like the mechanical compressor utilized in the VCR system. Also

the pump employed in the VAR system operates between lesser pressure differential compared to the compressor employed in the VCR system. A simple apparatus such as re-generator and a rectifier can be incorporated to improve the system performance. A sub-cooler can be placed in between the evaporator and absorber to cool the gaseous ammonia before entering the absorber there by improving the system performance. Multi-stage vapor absorption refrigeration systems, which are relatively expensive, are also available. They have a higher COP than single stage vapor absorption refrigeration systems.

3.2.3 Factors Affecting COP of Ammonia Absorption

The COP of the ammonia absorption refrigeration system is affected by the generator temperature (T_{ge}), evaporator temperature (T_{ev}) and condenser temperature (T_{cn}) [11]. These effects are presented below:

The VAR system COP increases with an increase in the generator temperature (T_{ge}).

The system COP increases as the evaporator temperature (T_{ev}) increases and the system COP increases as the condenser temperature (T_{cn}) decreases. It is also to be noted that the cooling capacity increases as the condenser temperature (T_{cn}) decreases [11].

3.2.4 Advantages of Ammonia Absorption Refrigeration System

The ammonia absorption system offers certain advantages over the conventional vapor compression refrigeration system:

It has no moving parts except the solution pump. So there is less wear and tear leading to less frequent maintenance and less shutdown time. This implies that the maintenance cost of the VAR system will be less compared to the VCR system. The

VAR system can be installed out doors or on roof tops as is done for the VCR system. It has a wide operating range of $+5^{\circ}\text{C}$ to -55°C as the freezing point of ammonia is -77°C . The VAR system typically has a long life of at least 25 years.

There are some advantages of using ammonia as a refrigerant [13]. Ammonia has zero potential for global warming and ozone layer depletion unlike the CFCs. Ammonia has a higher latent heat of vaporization and hence is slightly more efficient as a refrigerant than CFCs. Therefore the system runs on lesser energy. Ammonia is cheaper than CFCs. Also, there exists tremendous amount of practical experience in the handling of ammonia as a refrigerant. Ammonia has an unpleasant odor and this property of it makes it easy to detect any leaks in the system. There is an unlimited availability of ammonia which makes it an economical refrigerant.

However, there are few disadvantages of using ammonia as a refrigerant [13]: It is poisonous to humans if inhaled beyond a certain amount. If the concentration of ammonia in the air reaches 25 parts per million (ppm), it can cause headaches, nausea, and intense burning of the eyes, nose, throat, and skin. Ammonia is combustible, but it is very difficult to combust ammonia in air in the absence of a catalyst. Ammonia has an auto ignition temperature of 651°C . Ammonia is a corrosive substance and it is incompatible with copper, which is the most common material used in the current refrigeration systems.

CHAPTER 4 MATHEMATICAL MODELLING OF DDD PROCESS

The diffusion tower and the direct contact condenser are the main components of the DDD system. Inside the diffusion tower, evaporation occurs when heated saline water is sprayed from the top leading to the formation of a thin film of saline water on the packing material and is contacted by a low humidity turbulent air stream flowing in the opposite direction to that of saline water. The principles of heat transfer and mass are utilized to quantify rate of evaporation of water and the subsequent increase in the humidity of the air. It is desired that for high production of fresh water from the DDD system, the humidity ratio of the air stream coming out of the diffusion tower should be as high as possible. Ideally, the air exiting the diffusion tower should be in a saturated state. The equations governing the evaporation and humidification processes in the diffusion tower have been developed by Klausner et al. [7]. It is based on a two-fluid film model in which one dimensional conservation equations for mass and energy are applied to a differential control volume shown in Figure 4-1.

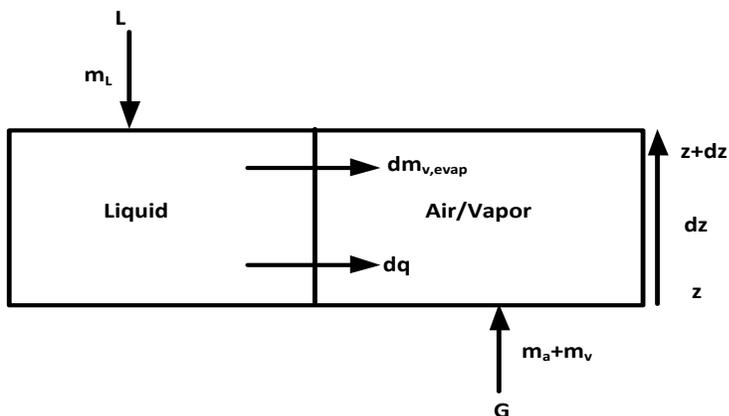


Figure 4-1. Differential control volume for liquid/gas heat and mass transfer within diffusion tower.

In the condenser tower, cool fresh water contacts the high humidity ratio turbulent air stream in the packing material and the heat given up to the water film leads to the

condensation of water vapor, and thus fresh water production. Thus, inside the direct contact condenser, the fresh water mass increases, and the humidity ratio of the air stream decreases. The formulation for the direct contact condenser has been developed by Klausner et al. [8]. A one dimensional two fluid condensation model is used to represent the change in the humidity ratio of the air inside the direct contact condenser tower. The mass and energy conservation equations are applied to a differential control volume shown in Figure 4-2.

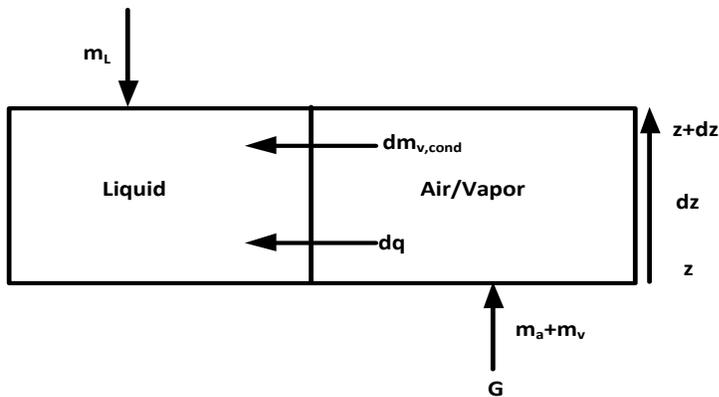


Figure 4-2. Differential control volume for liquid/gas heat and mass transfer within counter current condenser

The air temperature variation along the transverse direction in the condenser tower is important for the condensation process [10]. The local humidity based on the local transverse air temperature is averaged and the mean humidity is used in the one dimensional conservation equations.

The following assumptions [9] are made for the one-dimensional model reported here:

1. The process is at steady state.
2. Air and Water Vapor display perfect gases behavior.
3. The changes in kinetic and potential energies are neglected.
4. Water is to be pumped only against the gravitational force.
5. Heat lost to the surroundings is negligibly small.

4.1 Flow in the Diffusion Tower

For the mathematical modeling of the flow inside the diffusion tower, the conservation of mass principle is applied to the differential control volume shown in Figure 4-1. The conservation equations are applied separately to both the liquid and vapor phases in the differential control volume. Utilizing these conservation equations, the convective law of mass transfer and the relationship between relative humidity (ϕ) and the humidity ratio (ω), the following first order ordinary differential equation for the gradient of the humidity ratio in the diffusion tower is obtained:

$$\frac{d\omega}{dz} = \frac{k_G a_w}{G} \frac{M_V}{R} \left(\frac{P_{sat}(T_i)}{T_i} - \frac{\omega}{0.622 + \omega} \frac{P}{T_a} \right) \quad (4.1)$$

Here, $G = \left(\frac{m_a}{A}\right)$ is the air mass flux, k is the mass transfer coefficient, a_w is the wetting area of packing, R is the Universal gas constant, M_V is the vapor molecular weight, P is the system pressure and $P_{sat}(T)$ is the vapor saturation pressure at temperature . The solution for Equation 4.1 yields the variation of humidity ratio along the height of the diffusion tower. By assuming that the energy convected from the liquid is approximately equal to that convected to the gas, the interfacial temperature (T_i) is calculated using the Equation 4.2 :

$$T_i = \frac{T_L + \frac{U_G}{U_L} T_a}{1 + \frac{U_G}{U_L}} \quad (4.2)$$

Here, U_G and U_L are the heat transfer coefficients on the gas and liquid respectively.

Applying conservation of energy to the liquid phase in the differential control volume, a first order ordinary differential equation for the gradient of the liquid temperature (T_L) is obtained:

$$\frac{dT_L}{dz} = \frac{G}{L} \frac{d\omega}{dz} \frac{(h_{fg} - h_L)}{c_{pL}} + \frac{Ua(T_L - T_a)}{c_{pL}} \quad (4.3)$$

Here, $L = \left(\frac{m_L}{A}\right)$ is the water mass flux, U is the overall heat transfer coefficient h_{fg} is the latent heat of vaporization, h is enthalpy, a is the specific area of packing material and C_p is the specific heat. This first order ordinary differential equation can be solved for T_L , yielding the water temperature distribution along the height of the diffusion tower.

Similarly, applying conservation of energy to the air/vapor phase and neglecting the heat loss from air [14], the following equation is obtained:

$$\frac{dT_a}{dz} = -\frac{1}{1+\omega} \frac{d\omega}{dz} \frac{h_L(T_a)}{Cp_G} + \frac{Ua(T_L-T_a)}{Cp_G \cdot G(1+\omega)} \quad (4.4)$$

Equation 4.4 is also a first order ordinary differential equation which yields the air/vapor mixture temperature, T_a through the diffusion tower.

Together, the Equations 4.1, 4.3 and 4.4 constitute a set of coupled ordinary differential equations that can be solved for the humidity ratio, water temperature and air/vapor mixture temperature variations along the height of the diffusion tower. These coupled sets of equations require closure relationships since a one dimensional formulation is used here. This implies that the overall heat transfer coefficient and gas side mass transfer coefficient are required. The heat transfer coefficients for the air and vapor are evaluated using the heat and mass transfer analogy presented by Klausner et al. [7]. Onda's correlation [15] is used to calculate the mass transfer coefficients, k_G and k_L , in the diffusion tower.

4.2 Flow in the Condenser Tower

The flow modeling in the counter-current direct contact condenser is described in this section. The humidity ratio ω , which is dependent on the air temperature T_a , is calculated using the relationship between the relative humidity (ϕ) and the humidity ratio (ω) utilizing Equation 4.5:

$$\omega = \frac{m_V}{m_a} = \frac{0.622\phi P_{sat}(T_a)}{P - \phi P_{sat}(T_a)} \quad (4.5)$$

where, P (kPa) is the total system pressure and P_{sat} (kPa) is the water saturation pressure corresponding to the local air temperature, T_a . P_{sat} (kPa) can be calculated using the following empirical representation of the saturation line:

$$P_{sat}(T) = a \exp(bT - cT^2 + dT^3) \quad (4.6)$$

where, the empirical constants are: $a = 0.611379$, $b = 0.0723669$, $c = 2.78793e-7$, $d = 6.76138e-7$ and T ($^{\circ}\text{C}$) is the temperature. For the DDD application, the temperature range across any given cross-section is small and so the area averaged humidity ratio ω_m can be approximated by $\omega_m = \omega(T_a)$ in Equation 4.5 [6].

However, the relative humidity of air remains 100% during the condensation process. This condition implies that the absolute humidity is only a function of air temperature T_a . Taking this into consideration and differentiating Equation 4.5 and combining with Equation 4.6, the following first order ordinary differential equation shown in Equation 4.7, which expresses the gradient of humidity along the height of the direct contact condenser, is obtained,

$$\frac{d\omega}{dz} = \frac{dT_a}{dz} \frac{P}{P - P_{sat}(T_a)} \omega (b - 2cT_a + 3dT_a^2) \quad (4.7)$$

Applying the conservation of energy to the liquid phase in the differential control volume shown in Figure 4-2, the gradient of water temperature in the condenser tower can be expressed as:

$$\frac{dT_L}{dz} = \frac{G}{L} \frac{d\omega}{dz} \frac{(h_{fg} - h_L)}{c_{pL}} + \frac{Ua(T_L - T_a)}{c_{pL}L} \quad (4.8)$$

By applying energy conservation to the gas phase in the differential control volume shown in Figure 4-2, the gradient of the air temperature in the condenser tower is expressed as:

$$\frac{dT_a}{dz} = -\frac{1}{1+\omega} \frac{d\omega}{dz} \frac{h_L(T_a)}{Cp_G} + \frac{Ua(T_L-T_a)}{Cp_G G(1+\omega)} \quad (4.9)$$

Similar to the evaporation model, a one dimensional approach is used for the condensation model (Equations 4.7 and 4.8) and thus it requires closure relationships (Equations. 4.4 and 4.6). Onda's correlation and the heat and mass transfer analogy presented in the Appendix A are used to calculate the mass transfer and heat transfer coefficients. In this model the gas side mass transfer coefficient is taken as (see Appendix A).

$$k_G = 2.0 Re_{GA}^{0.7} Sc_G^{\frac{1}{3}} (ad_p)^{-2} a D_G \quad (4.10)$$

Onda suggests that the effective packing diameter affects the mass transfer coefficient on the gas side and he recommended the use of 5.23 for the co-efficient in Equation 4.10 for the cases where the effective packing diameters are larger than 15mm and 2.0 for those less than 15mm. In the DDD system, the packing material used in the direct contact condenser has an effective packing diameter of 17mm, which is close to the limit reported by Onda. The use of 2.0 for the coefficient in Equation 4.10 is justified in Knight et al. [14].

The fresh water production rate is calculated as follows:

$$m_f = GA(\omega_{in} - \omega_{out}) \quad (4.11)$$

Here, the subscripts f , in , out respectively refer to the fresh water, condenser inlet and condenser outlet.

4.3 Numerical Procedure

The numerical procedure to compute the exit humidity ratio, exit water temperature and exit air temperature from the diffusion tower and the condenser tower is presented by Klausner et al. [9], and is detailed below:

1. Specify the inlet water temperature, air temperature and humidity.
2. Guess the exit water temperature.
3. Compute the temperature distributions and humidity distribution through the packed bed using Equations 4.1, 4.3 and 4.4.
4. Check whether the computed inlet water temperature agrees with the specified inlet water temperature and stop the computation if agreement is achieved, otherwise repeat from step 2.

For the counter-current direct contact condenser analysis, the numerical procedure to calculate the exit water temperature, exit air temperature and exit humidity ratio, is reported here:

1. Specify the inlet water temperature, air temperature and bulk humidity.
2. Guess the exit water temperature.
3. Compute the temperatures and bulk humidity at the next step change in height using Equations 4.7, 4.8 and 4.9.
4. Proceed to new height and restart the computation from step 3 until the computed air exit temperature matches the specified air exit temperature.
5. Check whether computed inlet water temperature agrees with the specified inlet water temperature and stop the computation if agreement is found, otherwise repeat the procedure from step 2.

CHAPTER 5 RESULTS AND DISCUSSION

The main aim of the discussion presented in this section is to investigate the improvement in the performance of the DDD plant due to the introduction of AAR system. The AAR system reduces the fresh water inlet temperature into the condenser tower to 1°C from 25°C, and It is expected that there would be an increase in the amount of condensation in the condenser tower. Thus, due to the increased condensation, there will be an improvement in the amount of fresh water produced by the DDD plant.

As mentioned earlier, the DDD plant can be run in three modes namely, the heated air / ambient water mode, the heated air / heated water mode and the ambient air / heated water mode. In the current analysis, the heated air / heated water mode is chosen. The application of interest involves running the DDD plant with the waste heat energy available from a solar driven combined cycle power plant, and both heated air and heated water are readily available. Also, a preliminary study, which is not mentioned in this report, indicates that the heated air / heated water case yields the highest fresh water production among all the three cases for the same mass fluxes.

5.1 Diffusion Tower Analysis

The equations that were developed in the theoretical modeling of the diffusion tower are solved using the numerical procedure that is described in the previous chapter. A Runge-Kutta 4 numerical method was used to solve the equations developed by Yi Li et al. [6]. A code is written by Yi Li in Fortran 77 for the same. The same code is used to solve the heated air / heated water case. The coefficient C used in Onda's correlation is fixed as 2.0 for this case as is justified by Knight et al. [14]. In the

numerical analysis results presented , the specific area of packing is taken as $267\text{m}^2/\text{m}^3$ and the diameter of the packing material is taken as 0.018m. These are the specifications corresponding to HD QPACK, a commercial packing material manufactured by Lantec.

Certain temperatures, such as the saline water inlet temperature and air inlet temperature into the diffusion tower, used in this numerical analysis are calculated from the energy balance of a 271 MW power plant described in Appendix B.

In the heated air / heated water case, the temperature of air is taken as 93°C , since it is the temperature of exhaust air available from the combined cycle power plant which has been mentioned in [16]. It is assumed for the purpose of the numerical analysis that saturated air at ambient i.e. at 25°C , is taken and heated to 93°C . Therefore, the inlet humidity ratio of air at 93°C entering the diffusion tower remains unchanged and is equal to the humidity ratio of saturated air at 25°C i.e. 0.0201. The temperature of the saline water entering the diffusion tower of the DDD plant is assumed to be 80°C for the reason that it is possible to heat the pre-heated saline water, coming out of the steam turbine condenser at 35°C , to 80°C utilizing the energy available in the exhaust air stream, which is at 93°C . The diffusion tower analysis is performed in order to obtain the input data into the condenser tower like the temperature and humidity ratio of air entering the condenser tower.

In the diffusion tower analysis, the numerical simulations were run for different air to feed water mass flow ratios for a particular feed water mass flux. Results are obtained in a similar way for different feed water mass fluxes. A sample of the numerical values thus obtained are tabulated and presented in the Appendix C. The results are

plotted and are presented here. The results have been verified against the results presented in Knight et al. [14] and it is observed that similarity exists in the trends observed in both the results.

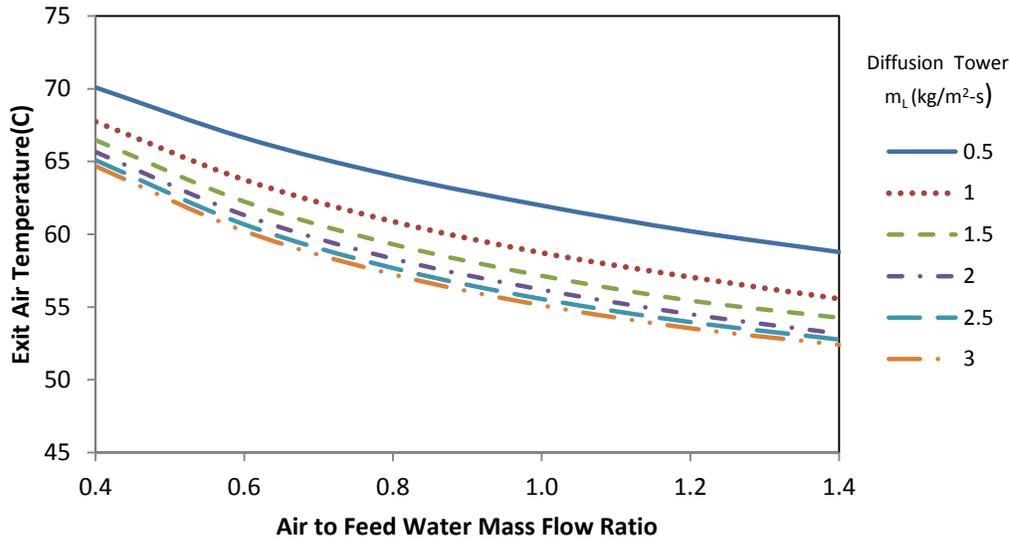


Figure 5-1. Diffusion tower exit air temperature variation with air to feed water mass flow ratio

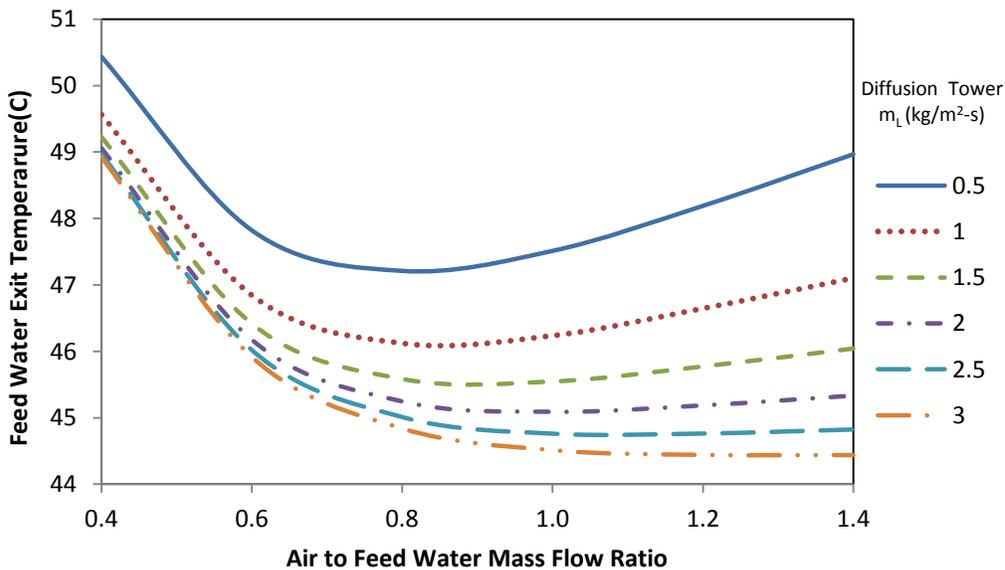


Figure 5-2. Diffusion tower exit feed water temperature variation with air to feed water mass flow ratio.

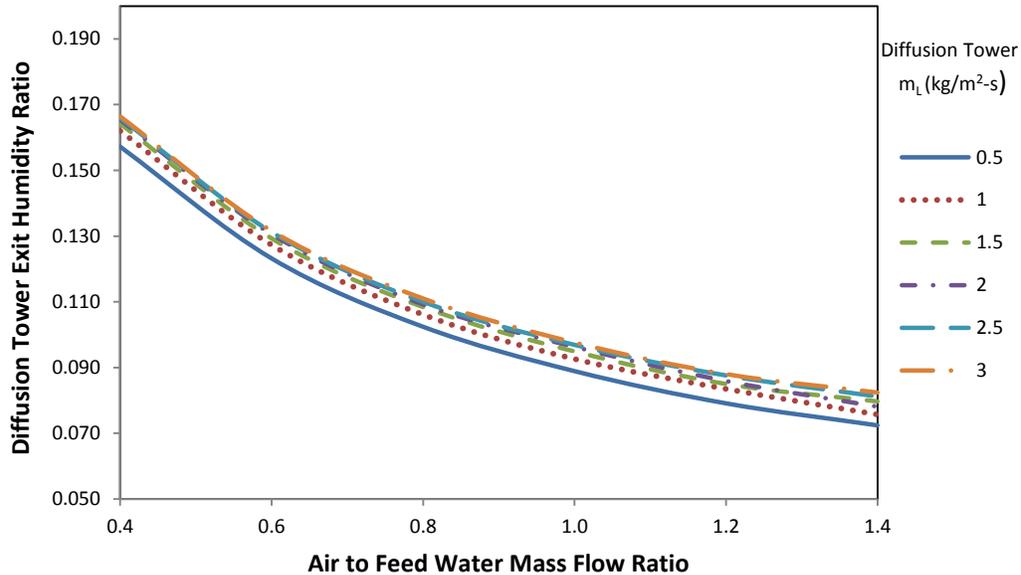


Figure 5-3. Diffusion tower exit humidity ratio variation with air to feed water mass flow ratio.

Figure 5-1 represents the variation of the exit air temperature from the diffusion tower with air to feed water mass flow ratios, for different feed water mass fluxes. It is seen that the exit air temperature from the diffusion tower is decreasing with increase in the air mass flux, for a given feed water mass flux.

In Figure 5-2, the variation of feed water exit temperature with air to feed water mass flow ratio, for various feed water mass fluxes is shown. It is seen that there is initially a dip in the feed water exit temperature from the diffusion tower as the air to feed water mass flow ratio increases. Further increase in the air to feed water mass flow ratio causes a small increase in the feed water exit temperature. The plot shows that as the water feed mass flux increases, the feed water exit temperature from the diffusion tower decreases.

The exit feed water from the diffuser cannot be used as a coolant for either the steam turbine condenser or the absorber in the ammonia-water vapor absorption

refrigeration system utilized in the DDD plant. This is because it is clear from the graph in Figure 5-2 that the feed water exit temperature from the diffusion tower is high, making it unsuitable for use as a coolant in the present case. However, energy required to heat the feed water for the diffusion tower can be reduced by re-circulating this exit feed water.

Figure 5-3 shows the variation in the exit humidity ratio from diffusion tower as the air to feed water mass flow ratio changes. A high exit humidity ratio is desired from the diffusion tower as it indicates better mass transfer. However, it should be noted that a high exit humidity ratio alone does not necessarily indicate a high fresh water production. It can be observed from the graphs that a change in water feed mass flux has no significant effect on the exit humidity ratio of air from the diffusion tower. It can also be seen from the graphs that low air to feed water mass flow ratios yield higher exit humidity ratios.

The arguments justifying the observed behavior of the different parameters presented in the Figures 5-1, 5-2 and 5-3 have been mentioned in detail by Knight et al. [14].

5.2 Condenser Tower Analysis

The results for the condenser tower are discussed next. Based on the results of exit humidity ratio from the diffusion tower, the rate of water vapor evaporated into the air stream, $m_{fw, evap}$ is computed as:

$$m_{fw, evap} = G(\omega_{out, d} - \omega_{in, d}) \quad (5.1)$$

where, G is the mass flux of air, $\omega_{in, d}$ and $\omega_{out, d}$ are respectively the humidity ratio of air at the inlet and at the outlet of the diffusion tower. From these results, exit humidity ratio

from the diffusion tower which yields the highest fresh water output is chosen and is considered for the analysis of the condenser tower. The exit temperature, exit humidity and mass flux of air from the diffusion tower of the highest exit fresh water output case become the inlet temperature, inlet humidity ratio and inlet mass flux of air respectively, in the condenser tower analysis. Interestingly, the highest fresh water output is obtained for the lowest exit humidity case instead of the highest exit humidity case. This is due to the fact that the lowest exit humidity from the diffusion tower is obtained for the highest air to feed water mass flow ratio and hence, a large amount of air is available to condense out the fresh water, thereby yielding a high fresh water production. The condenser analysis is similar to the diffusion tower analysis. In the diffusion tower analysis presented previously, the air to feed water ratio is varied for different feed water mass fluxes and the behavior of different parameters is observed. However, in the condenser tower analysis, the fresh water feed to air mass flux ratio is varied for different air mass fluxes, and the variation of different parameters such as the condenser tower exit air and water temperatures and condenser tower exit humidity ratio are computed. In addition, in the condenser tower analysis, the behavior of the different parameters is compared for two inlet feed water temperatures, the temperatures being 25°C and 1°C. As mentioned in the earlier section, the reason for comparing the two temperatures is, it is expected that by reducing the inlet feed water temperature in the condenser tower to 1°C, more condensation would be possible leading to higher fresh water production. The choice of the fresh water feed inlet temperature is made as 25°C because it is assumed that the fresh water at ambient is available at this temperature. The other feed water inlet temperature is chosen as 1°C

as it is the lowest possible temperature below which water freezes and causes operational problems for the DDD plant.

The same Runge-Kutta 4 numerical method used for the diffusion tower analysis is utilized in the condenser analysis. Onda's correlation coefficient C is fixed as 2.0 for this analysis. The other parameters that were involved in the numerical analysis are mentioned here: The specific area of packing is taken as $267\text{m}^2/\text{m}^3$ and the packing diameter is taken as 0.017m.

Figures 5-4A and 5-4B show the variation of exit air temperature from the condenser tower when fresh water inlet temperature is 25°C and 1°C respectively, for different fresh water feed to air mass flow ratios. The variation in the exit air temperature is also shown for various air mass fluxes. The change in the behavior of the exit air temperature for the two inlet feed water temperatures is also presented in the graphs. The exit air temperature from the condenser tower tends to remain nearly constant for large fresh water feed to air mass flow ratio. It is observed that this constant temperature value in both cases is approximately equal to the respective inlet feed water temperatures i.e. 25°C and 1°C . However, in both cases, at low fresh water feed to air mass flow ratios, the exit air temperature is elevated, indicating poor heat transfer which can be attributed to the lack of a sufficient amount of fresh water for cooling. The air mass flux is found to show a small influence on the exit air temperature from the condenser tower for the 25°C inlet water case. However, for 1°C inlet water, the air mass flux has a significant influence at low feed water to air mass flow ratios.

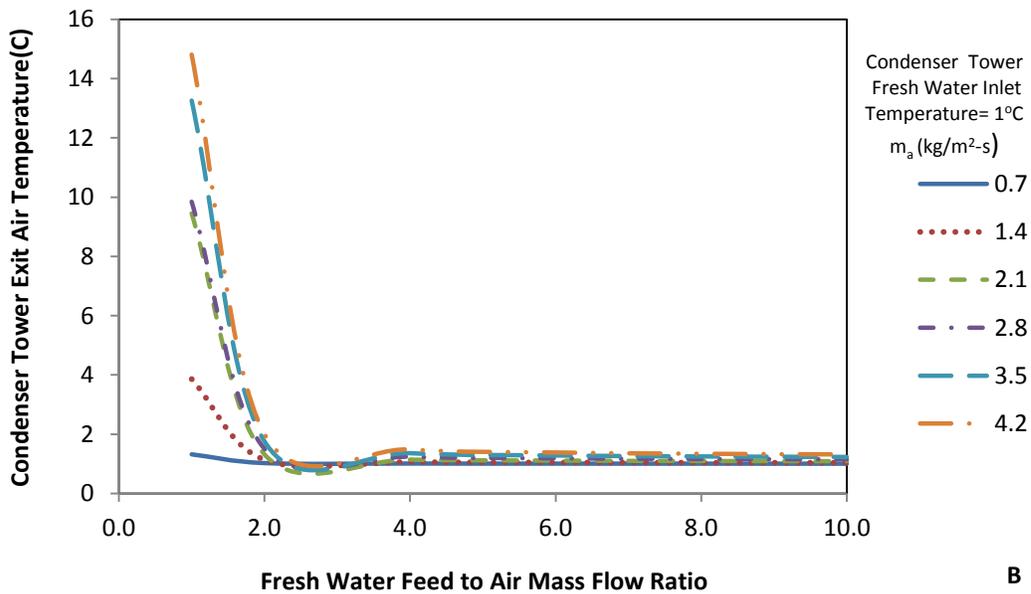
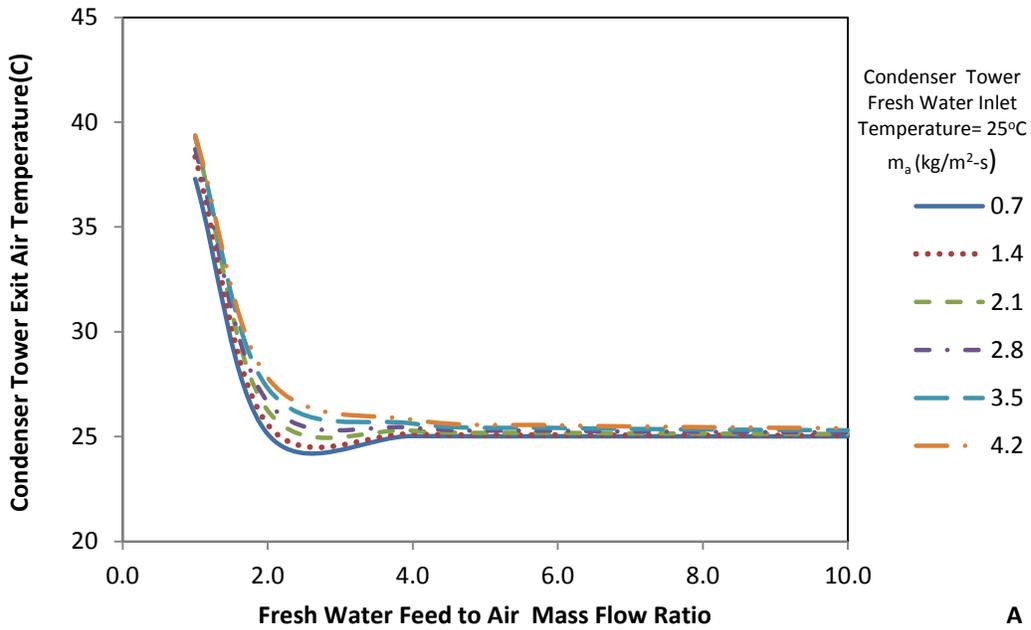


Figure 5-4. Condenser tower exit air temperature variation with fresh water feed to air mass flow ratio. Fresh water inlet temperature is: A) 25°C. B) 1°C.

It is desired that the exit air temperature from the direct contact condenser be as low as possible because this air is intended to be used as a coolant for the condenser in the AAR system. The lower the coolant temperature, the lower will be the temperature

of the ammonia vapor cooled in the condenser and higher will be the efficiency of the AAR system.

Figures 5-5A and 5-5B show the variation of the exit feed water temperature from the direct contact condenser tower when fresh water inlet temperature is 25°C and 1°C respectively. The graphs show that the exit feed water temperature from the direct contact condenser decreases with an increase in the fresh water feed to air mass flow ratio. It can be observed that in both cases, the exit feed water temperature decreases rapidly up to a fresh water feed to air mass flow ratio of 4.0, and the decrease is gradual with a further increase in the fresh water feed to air mass flow ratio. It can be observed from the graphs that, in both cases, the exit feed water temperature from the direct contact condenser tower is unaffected by the air mass flux in the condenser tower. In general, the reason for the decrease in the feed water exit temperature with increase in fresh water feed to air mass flow ratio is that, the amount of air available decreases and the amount of heat carried and so there is lower amount of heat being transferred to the feed water thereby decreasing its exit temperature. Specifically, when the feed water temperature is 1°C, its exit temperature is desired to be as low as possible. This is because, in the DDD plant, the exit feed water from the direct contact condenser is cooled to 1°C by the AAR system and is re-circulated as inlet feed water to the condenser tower. The lower this exit temperature is, the less cooling load will be required from the AAR system.

Figures 5-6A and 5-6B show the variation of the exit humidity ratio from the direct contact condenser tower when fresh water inlet temperature is 25°C and 1°C

respectively, for different fresh water feed to air mass flow ratios. This variation is presented for the two fresh water feed inlet temperatures of 25°C and 1°C.

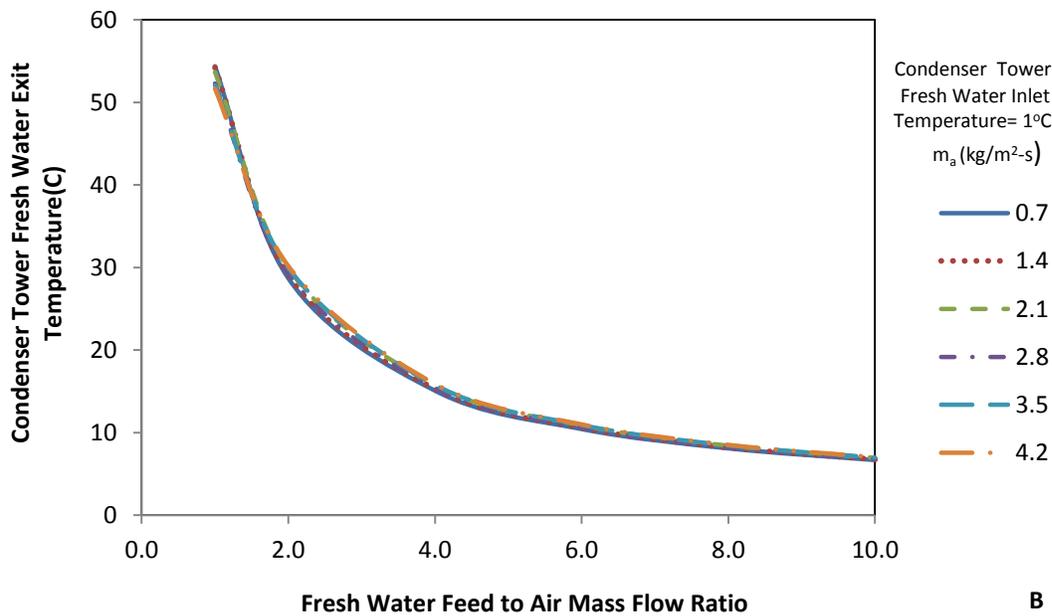
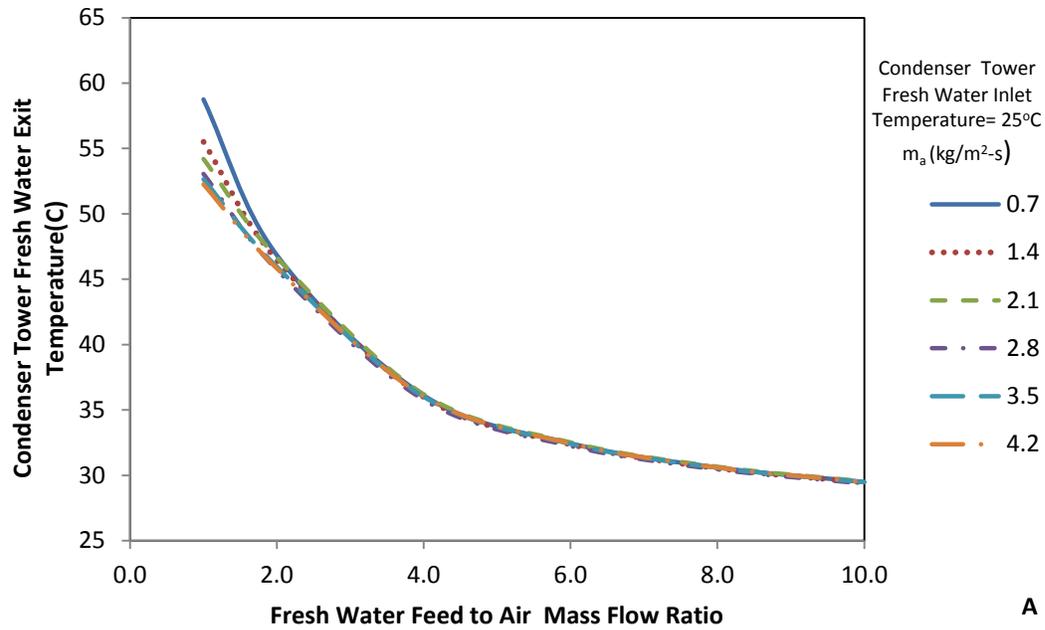
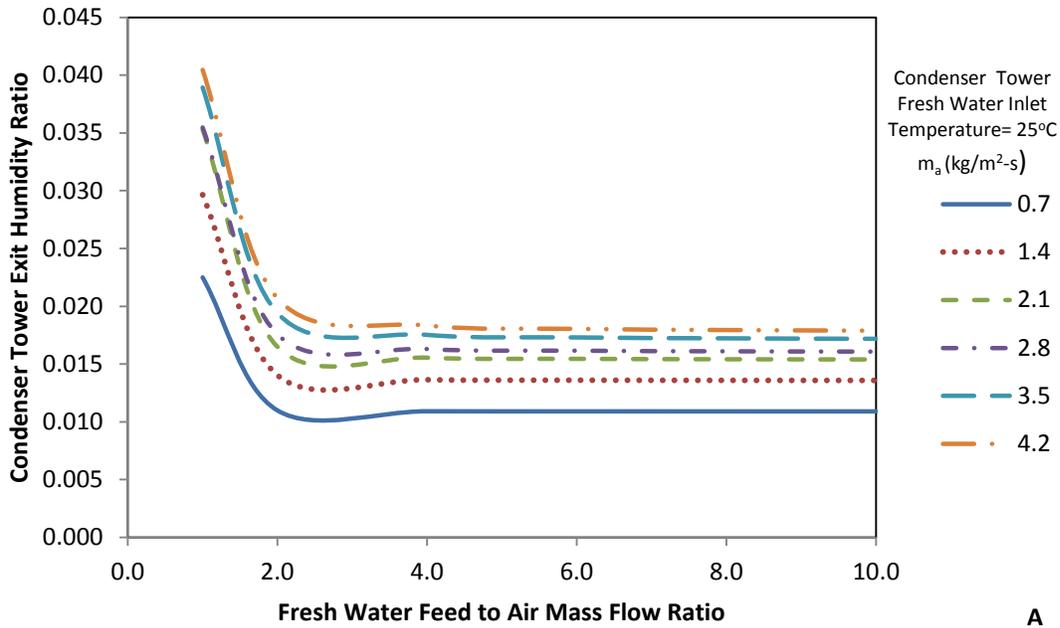
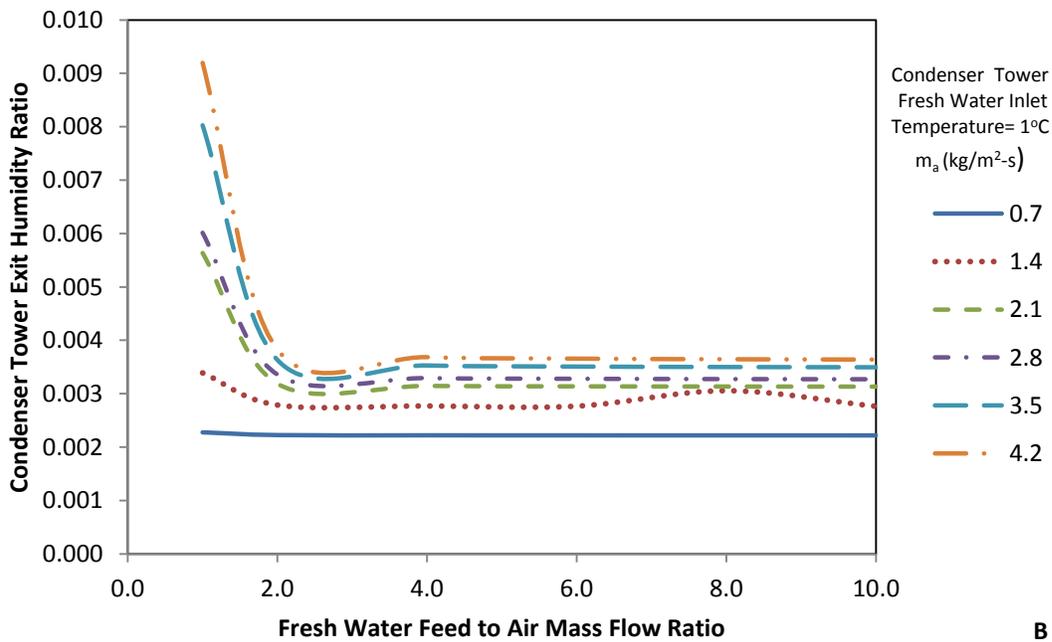


Figure 5-5. Condenser tower exit fresh water temperature variation with fresh water feed to air mass flow ratio. Fresh water inlet temperature is: A) 25°C. B) 1°C.



A



B

Figure 5-6. Condenser tower exit humidity ratio variation with fresh water feed to air mass flow ratio. Fresh water inlet temperature is: A) 25°C. B) 1°C.

In general, it can be observed from the graphs that the exit humidity ratio from the condenser tower remains largely unaffected by larger fresh water feed to air mass flow

ratios. It is desired that the difference between the inlet and exit humidity ratios of air from the direct contact condenser tower is as high as possible. This is because, this difference in the humidity ratio directly relates to the amount of fresh water that can be condensed. It can be observed from the graphs that when the fresh water feed inlet temperature is 1°C, the exit humidity ratios are significantly lower, nearly 10 times less, than their corresponding values when the fresh water feed inlet temperature is 25°C. This implies that the difference between inlet and exit humidity ratios is significantly higher when the fresh water feed inlet temperature is 1°C than when the fresh water feed inlet temperature is 25°C. Therefore, it can be concluded from the graphs that by reducing the fresh water feed inlet temperature to 1°C from 25°C, there can be a significant increase in the production of fresh water from the DDD plant.

Figures 5-7A and 5-7B show the fresh water mass flux produced in the condenser tower when the fresh water feed inlet temperature is 25°C and 1°C respectively, for various fresh water feed to air mass flow ratios and also for different air mass fluxes in the condenser tower. The rate of fresh water mass flux produced in the condenser tower, L_{fwp} is computed as,

$$L_{fwp} = G(\omega_{in} - \omega_{out}) \quad (5.2)$$

where, G is the mass flux of air, ω_{in} and ω_{out} are respectively the humidity ratio of air at the inlet and at the outlet of the condenser tower. As observed from the graphs, in both cases, there is no significant increase seen in the mass of fresh water produced with an increase in the fresh water feed to air mass flow ratio beyond 2.0. It is economical to operate the DDD system with the least amount of electric power. The electric power is

required in the DDD system to run the pumps and blowers which circulate air and water throughout the system.

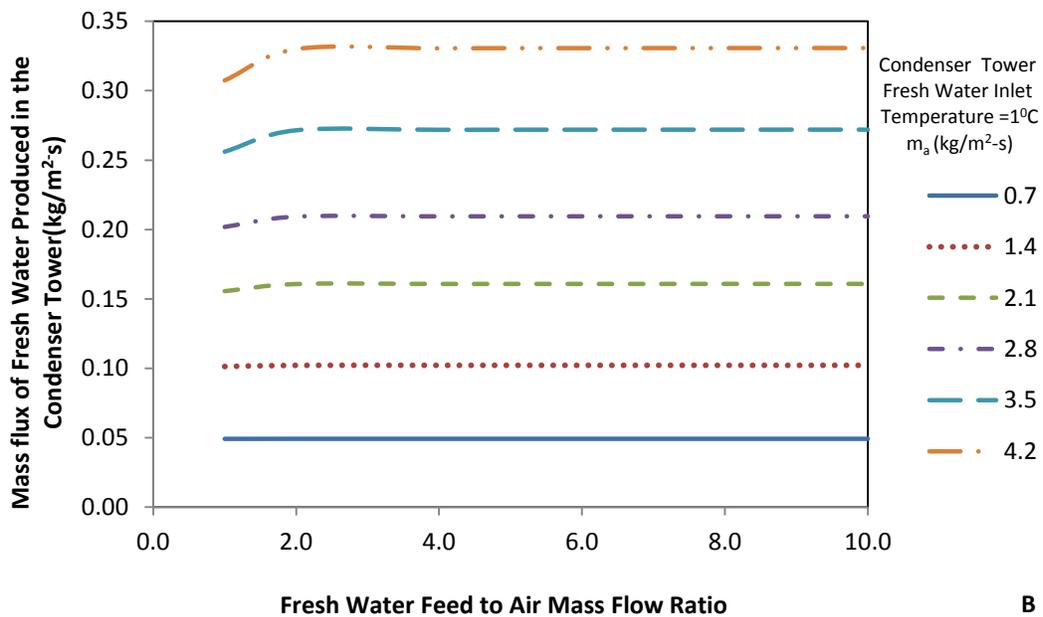
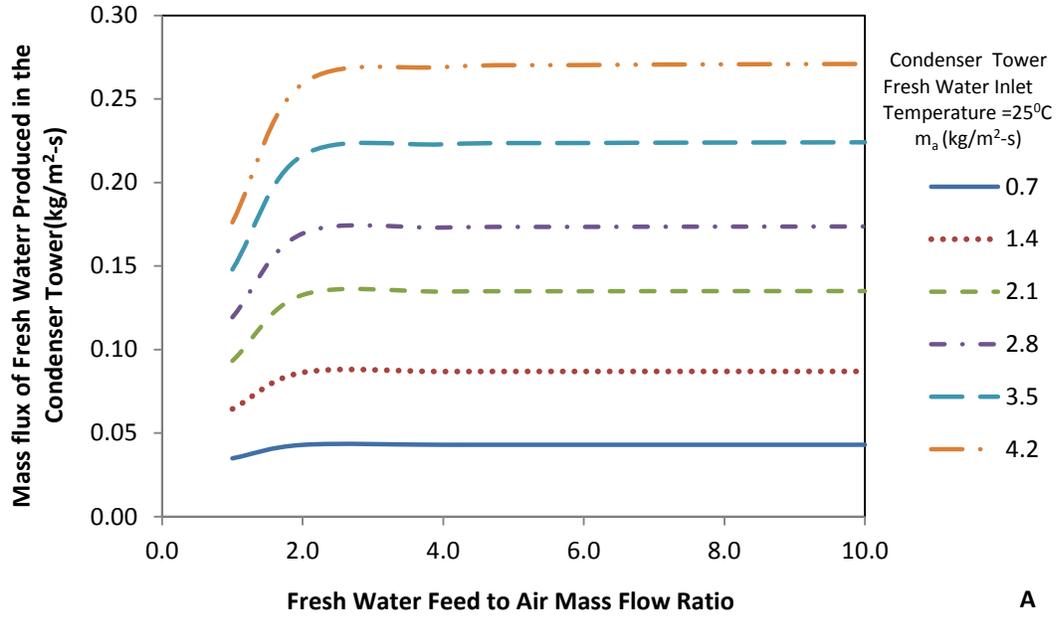


Figure 5-7. Fresh water production variation with fresh water feed to air mass flow ratio. Fresh water inlet temperature is: A) 25°C. B) 1°C.

The electric power consumed by the pumps varies directly with the mass of fluid it can pump. Yi Li's work suggests that the DDD plant consumes least electricity for the amount of fresh water produced at a fresh water feed to air mass flow ratio of 2.0. Therefore, in keeping with the above argument, from the Figures 5-5 and 5-6, it can be concluded that it is economical to operate the condenser tower of DDD plant at a fresh water feed to air mass flow ratio of 2.0, since it yields the maximum fresh water production with least electricity consumption.

The percent increase in water production by incorporating the AAR system is shown in Figure 5-8. It is observed from Figure 5-8 that for higher air mass fluxes, there is an increase in the mass flux of fresh water produced from the condenser tower due to the reduction in the fresh water feed inlet temperature from 25°C to 1°C, for all corresponding operating conditions. It is also clear from Figure 5-8 that for fresh water feed to air mass flow ratios below 2.0, the improvement in the performance of the DDD plant due to the addition of AAR system is significantly higher than that for fresh water feed to air mass flow ratios above 2.0. Also, the increase in the performance of the DDD plant remains nearly constant beyond a fresh water feed to air mass flow ratio of 2.0.

From the analysis presented in this section, it can be concluded that there is an improvement in the performance of the DDD plant due to the reduction of the fresh water feed inlet temperature into the condenser tower. Hence, the addition of a refrigeration plant, which in this case is the AAR plant, to the DDD plant to reduce the fresh water feed inlet temperature gives better performance. Whether or not the increased performance can be justified based on cost requires an economic assessment.

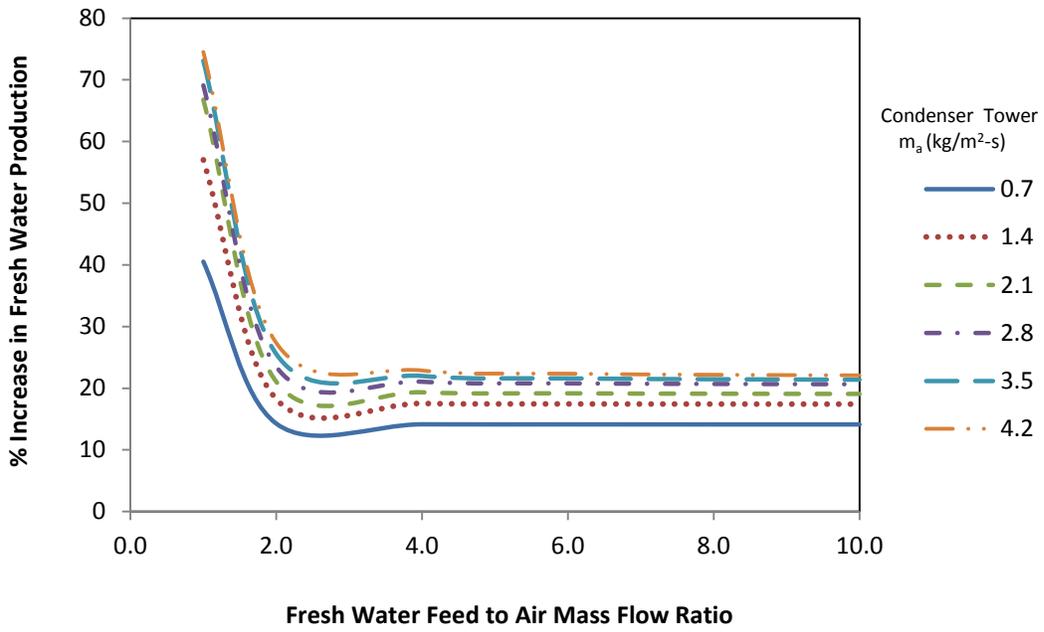


Figure 5-8. Percentage increase in fresh water production variation with fresh water feed to air mass flow ratio.

CHAPTER 6 INDUSTRIAL APPLICATION OF DDD PLANT WITH AAR SYSTEM

In this section, a preferred operating condition for the DDD plant with an AAR system is investigated where it is powered by the waste heat from a 271MW solar combined cycle power plant. The operating conditions for the 271 MW solar power plant are discussed in detail in Appendix B. The approach that is followed in arriving at a preferred operating condition is described. m_{AAR} is the mass flow rate (kg/s) of exhaust air from the combined cycle power plant that is diverted to the AAR system to supply heat energy to it, and m_{ea} is the total mass flow rate (kg/s) of exhaust air from the combined cycle power plant. Therefore, $x = \frac{m_{AAR}}{m_{ea}}$ defines the fraction of exhaust air (exhaust or waste energy) that is utilized to power the AAR plant. $(1 - x)$ denotes the fraction of exhaust air that powers the DDD plant. The ambient saline water temperature and the ambient air temperature are taken as 25°C. The ambient sink temperature is also taken as 25°C. The aim of this analysis is to arrive at a preferred value of x which yields the highest fresh water production with the available energy.

The most efficient operating conditions for the condenser tower were identified by Yi Li et al. [6] as:

1. The areas of the diffusion and condenser towers are assumed to be equal.
2. Mass flux of air in the diffusion and condenser towers, $G = 1.5 \text{ kg/m}^2\text{-s}$
3. Mass flow ratio of fresh water feed to air in the condenser tower, $\frac{L_{fw}}{G} = 2$ where, L_{fw} is the mass flux ($\text{kg/m}^2\text{-s}$) of fresh water in the condenser tower.

Consider the AAR system,

$$Q_{AAR} = x * m_{ea} * C_{p,air} * (T_{ea} - T) \text{ kW} \quad (6.1)$$

is the amount of waste heat available for the AAR system, where $C_{p,air}$ is the specific heat of air at constant pressure (kJ/kg K), T_{ea} is the exhaust temperature of air from the combined cycle plant which is 93°C, and T is the temperature to which this exhaust air can be cooled to. The value of T depends on the temperature of the ammonia-water mixture entering the generator of the AAR system and is taken to be approximately 5°C greater than the temperature of the ammonia-water mixture entering the generator of the AAR system.

Let the coefficient of performance of the AAR plant as defined in earlier sections be denoted by COP . By definition of COP , the cooling load, Q_{CL} , that can be achieved by the AAR system with the available input energy, Q_{AAR} , is calculated as,

$$Q_{CL} = (COP * Q_{AAR}) \text{ kW} \quad (6.2)$$

Let m_{cf} denote the mass flow rate (kg/s) of fresh water that can be cooled by an AAR system with a cooling capacity of Q_{CL} . Therefore,

$$m_{cf} * C_{water} * (T_{f,out} - T_{f,in}) = Q_{CL} \quad (6.3)$$

where, C_{water} is the specific heat (kJ/kg K) of water, $T_{f,out}$ is the cooling fresh water temperature entering the evaporator of the AAR system. Its value depends on the cooling fresh water temperature exiting the condenser tower which is influenced by the specified operating mass fluxes and temperatures of the DDD plant. The value of $T_{f,out}$ is taken as 30°C in the cases where the cooling fresh water exit temperature from the condenser tower is greater than 30°C. This is because, it is expected that with the available sink temperature of 25°C, the cooling fresh water exiting the condenser tower can be cooled to 30°C, thereby reducing the load on the AAR system. In the other cases, $T_{f,out}$ is taken to be equal to the temperature of the cooling fresh water exiting

the condenser tower. In the cases of $T_{f,out}$ being less than 30°C, first an appropriate value of $T_{f,out}$ is guessed and simulation is performed. The obtained value of $T_{f,out}$ is compared against the guess value and if they both are different, another value of $T_{f,out}$ is guessed and the procedure is iterated until the obtained value and guess value of $T_{f,out}$ are equal. $T_{f,in}$ is the inlet temperature of fresh water feed in the condenser tower. In the present application $T_{f,in}$ can be either 1°C or 25°C, depending on whether AAR plant is utilized or not. m_{cf} is calculated from Equation 6.3, since all the other parameters involved are specified. Also, the fresh water mass flux, $L_{cf} = \frac{m_{cf}}{A}$, is known, where A is the area (m²) of the diffusion tower and the condenser tower. Therefore, the area of the towers is determined from knowing L_{fw} and m_{cf} .

Since, the air mass flux, $G = \frac{m_a}{A}$, is known, the mass flow rate (kg/s) of air flowing through the DDD system, m_a , is determined. From these variables, the saline water mass flow rate (kg/s), m_{sw} , is determined as follows.

$$Q_{sw} = [(1 - x) * m_{ea} - m_a] * C_{p,air} * (T_{ea} - T') \text{ kW} \quad (6.4)$$

Here, Q_{sw} is the amount of waste heat diverted to the DDD plant to supply heated feed water. T' is the temperature to which the exhaust air can be cooled to and it depends on T_h , which is the steam turbine condenser cooling water outlet temperature. T' is taken to be approximately 5°C greater than T_h . The mass of saline feed water, m_{sw} that can be heated to 80°C utilizing Q_{sw} is calculated using,

$$m_{sw} * C_{sw} * (T_{sw,in} - T_h) = Q_{sw} \quad (6.5)$$

where, C_{sw} is the specific heat (kJ/kg K) of saline water, $T_{sw,in}$ is the saline water inlet temperature into the diffusion tower, which is equal to 80°C in the present case. Since

the area of the diffusion tower and m_{sw} are known, the mass flux of the saline feed water into the diffusion tower, $L_{sw} = \frac{m_{sw}}{A}$, is calculated. Simulations for values of x ranging from 0 to 1.0 are carried out based on the values of L_{sw} , G , L_{fw} and A obtained from the above calculations.

Based on the area of the towers, which depends on the parameter x , the mass flow rate of fresh water that is produced in the condenser tower, m_{fwp} , is evaluated for each value of x as,

$$m_{fwp} = GA(\omega_{in} - \omega_{out}) \text{ kg/s} \quad (6.6)$$

For some larger values of x , the value of Q_{sw} becomes negative, indicating that there is no waste heat available for heating saline water feed. In such a case, the saline feed water temperature is taken to be equal to T_h . Also, since m_{sw} cannot be obtained by the approach mentioned previously, a value of the ratio $\frac{L_{sw}}{G}$ is chosen. In this application, the ratio $\frac{L_{sw}}{G}$ is taken as 1.0 for $x=0.9, 1.0$.

For $x=1$, the entire waste heat is utilized for supplying cooling fresh water at 1°C to the condenser tower by the AAR system. In this case, the temperature of air entering the diffusion tower is equal to T and the saline feed water temperature is equal to T_h .

For $x=0$, the entire waste heat is utilized for supplying heated air and heated water to the DDD plant. Therefore, the cooling fresh water temperature entering the condenser tower is taken to be equal to the sink temperature which is 25°C . The value of $\frac{L_{sw}}{G}$ is chosen as 1.0 in the simulation.

A sample calculation has been performed by taking the COP as 0.4, m_{ea} as 634 kg/s, T and T' as 40°C and T_h as 35°C . The remaining inlet conditions and the

results of the simulations for various values of x are tabulated and presented in Appendix D. The COP is taken as 0.4 as a result of an approximate theoretical analysis performed on a single effect AAR system operating under the available conditions. The results of the analysis are presented graphically below:

Figure 6-1 shows the volume of fresh water produced in one day by the DDD plant, in US gallons, as x varies from 0 to 1.0.

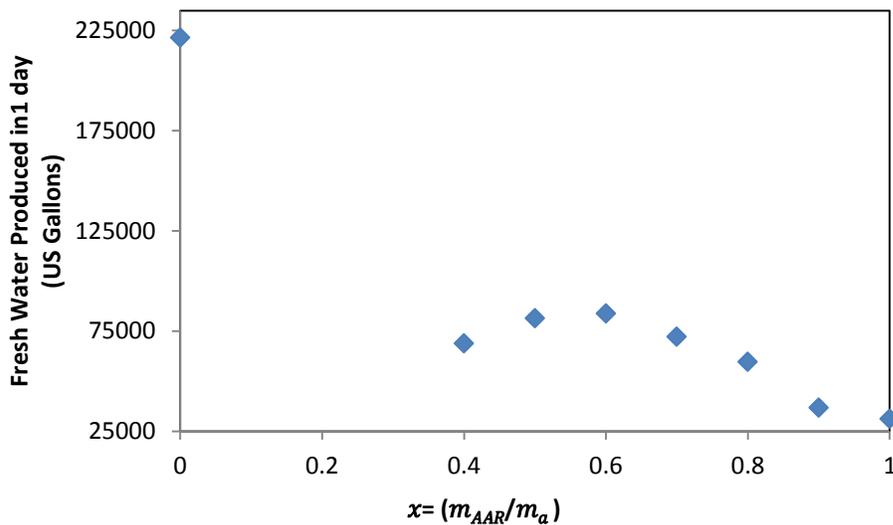


Figure 6-1. Variation of fresh water produced with x .

It is observed in Figure 6-1 that there is a maximum production of fresh water when x is 0. There is also a local maximum occurring when x is approximately 0.6. It is observed that the fresh water produced when x is 0.6 is significantly lower (approximately 2.65 times) than that when x is 0. The condition $x = 0$ virtually represents the absence of the AAR system. It can therefore be concluded from the graph in Figure 6-1 that the addition of AAR system, though improves the amount of condensation in the condenser tower and thereby the amount of fresh water produced

from the DDD plant, it does not increase fresh water production when both the DDD plant and the AAR system are running on shared energy from the same source.

In other words, more fresh water can be produced from the DDD plant if the entire available waste heat from the source is utilized by the DDD plant alone without the AAR system. The reason for this is the low *COP* of the AAR system under the available conditions. It has been determined that under the same operating conditions, if an AAR system with a *COP* of approximately 4.0 can be designed, only then the fresh water production from the DDD plant with an AAR system at least equals to that from the DDD plant without an AAR system. It is estimated that it would be extremely difficult to achieve a *COP* of approximately 4.0, even with a multiple effect AAR system, under the available conditions. It is understood from the above discussion that an AAR system with a *COP* greater than 4.0 is required in order to obtain higher fresh water production rates from the DDD plant than that from a DDD plant without the AAR system. It is also observed from the numerical calculations of the above analysis, which are not included in this report, that as the *COP* increases, the second highest fresh water production point shifts very slowly towards left.

The variation of the area of the diffusion tower and the condenser tower is shown in Figure 6-2. It is observed from the graph that the area of the diffusion and condenser towers is highest for $x = 1.0$. As the *COP* increases, it is observed from the theoretical analysis that only the area corresponding to $x = 0$, remains the same but the area corresponding to all other values of x increases, with the area corresponding to $x = 1.0$ being the highest.

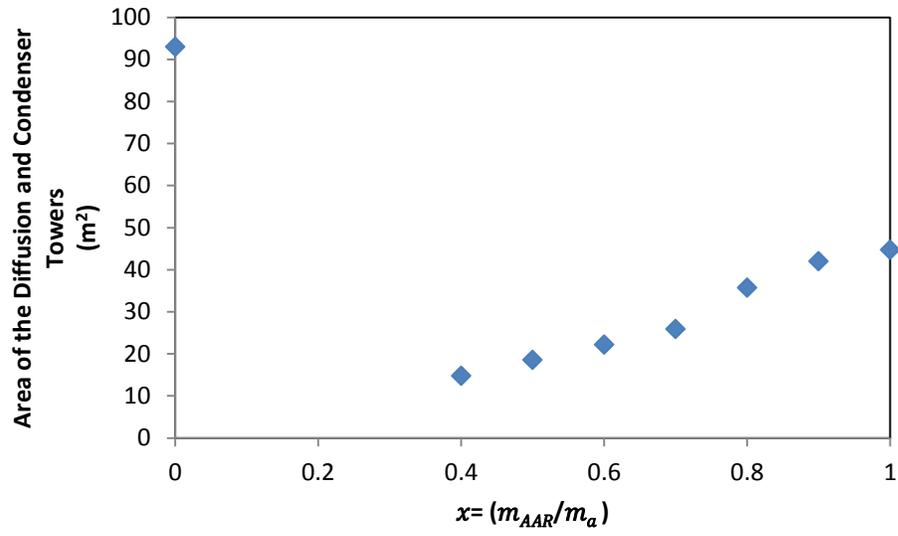


Figure 6-2. Variation of required area for diffusion and condenser towers with x .

CHAPTER 7 CONCLUSIONS

In this work, the improvement in the performance of the DDD plant due to the reduction of condenser inlet fresh water temperature from 25°C to 1°C has been presented. A literature study is conducted to understand the DDD process and the ammonia absorption refrigeration system in detail, which is included in this report. Simulations are run utilizing Yi Li's code, written in FORTRAN 77 for the heated air / heated water case, for the condenser fresh water inlet temperatures of 25°C and 1°C, for the same diffusion tower outlet conditions. The results of the simulations have been analyzed, and it is found that there is an increase in the fresh water production from the DDD plant due to reduction in the condenser fresh water inlet temperature. An ammonia absorption refrigeration system is chosen for cooling the condenser inlet fresh water from 25°C to 1°C, as it can run on waste heat. Waste heat from a solar combined cycle power plant is distributed between the DDD plant and an ammonia absorption refrigeration system and a theoretical analysis is carried out. The analysis yielded that the highest fresh water production occurs when all heat is diverted to the DDD system.. The important outcomes of this work are listed below:

1. For the heated air / heated water case, reducing the condenser fresh water inlet temperature from 25°C to 1°C results in an increase in the fresh water production.
2. Utilizing the entire available waste heat for the DDD plant yields a higher fresh water production instead of distributing the energy between the DDD plant and a low *COP* ammonia absorption refrigeration system.
3. In order to obtain higher fresh water production by reducing the condenser fresh water inlet temperature, the *COP* of the refrigeration system must be greater than 4.0 in the present application.

It is recommended that no further studies for a joint DDD/AAR system are warranted.

Economic feasibility does not appear to be achievable.

APPENDIX A
ONDA'S CORRELATION

$$*a_w = a \left\{ 1 - \exp \left[-2.2 \left(\frac{\sigma_c}{\sigma_L} \right)^{3/4} Re_{LA}^{1/2} Fr_L^{-0.05} We_L^{1/5} \right] \right\}$$

$$Re_{LA} = \frac{L}{a\mu_L}$$

$$Fr_L = \frac{L^2 a}{\rho_L^2 g}$$

$$We_L = \frac{L^2}{\rho_L \sigma_L a}$$

$$k_L = 0.0051 Re_{LW}^{2/3} Sc_L^{-0.5} (ad_p)^{0.4} \left[\frac{\mu_L g}{\rho_L} \right]^{1/3}$$

$$k_G = C Re_{GA}^{0.7} Sc_G^{1/3} (ad_p)^{-2} a D_G, \quad C = 5.23 \text{ if } d_p > 15 \text{ mm and } C = 2 \text{ if } d_p \leq 15 \text{ mm.}$$

$$Re_{LW} = \frac{L}{a_w \mu_L}$$

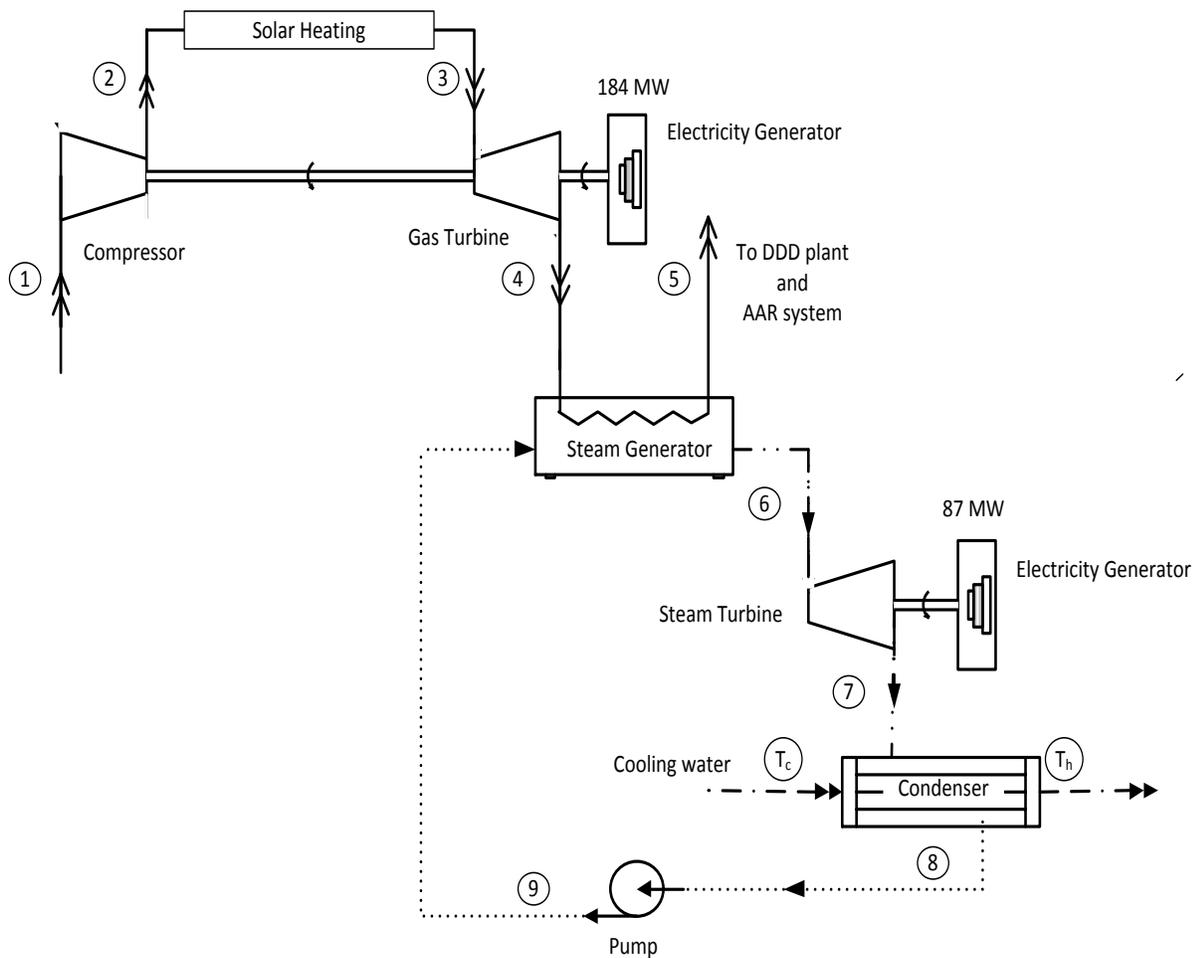
$$Re_{GA} = \frac{G}{a\mu_G}$$

$$Sc_L = \frac{\mu_L}{\rho_L D_L}$$

$$Sc_G = \frac{\mu_G}{\rho_G D_G}$$

*This equation has been modified from original Onda's correlation.

APPENDIX B CO-GENERATION PLANT DETAILS



Air / Vapor
 Steam
 Condensate of Steam
 Cooling Water

DDD- Diffusion Driven Desalination
 AAR- Ammonia Absorption Refrigeration

Figure B-1. Co-generation plant

State Point 1: Here, the air at ambient conditions enters the compressor.

Mass flow rate of air = $m_{ea} = 634 \text{ kg/s}$

Inlet temperature of air = $T_1 = 25^\circ\text{C}$

Inlet pressure of air = $P_1 = 1.01 \text{ bar}$

State Point 2: Here, the air has been compressed and enters into the solar heating chamber.

$$\text{Pressure ratio} = \frac{P_2}{P_1} = 12.4$$

$$\text{Pressure of air} = P_2 = 12.524 \text{ bar}$$

$$\text{Temperature of air} = T_2 = 338.8^\circ\text{C}$$

State Point 3: Here, the compressed air has been heated using solar energy and enters the gas turbine.

$$\text{Pressure of air} = P_3 = 12.524 \text{ bar}$$

$$\text{Temperature of air} = T_3 = 990^\circ\text{C}$$

State Point 4: Here, the hot compressed air is expanded in the gas turbine and the exhaust from it enters into the boiler or steam generator. The pressure is calculated assuming an isentropic expansion in the turbine.

$$\text{Temperature of air} = T_4 = 500^\circ\text{C}$$

$$\text{Pressure of air} = P_4 = 2.25 \text{ bar}$$

State Point 5: Here, part of the energy available in the exhaust stream of the gas turbine is utilized to heat water and produce steam. Assuming no heat loss in the steam generator and performing an energy balance on the steam generator as the control volume, we obtain the temperature of air exiting the steam generator.

$$\text{Temperature of air} = T_5 = 93^\circ\text{C}$$

State Point 6: Here, steam is generated in the steam generator and it enters the steam turbine.

$$\text{Mass flow rate of steam} = m_s = 80.28 \text{ kg/s}$$

$$\text{Inlet pressure of steam} = P_6 = 36 \text{ bar}$$

Inlet temperature of steam = $T_6 = 473^\circ\text{C}$

State Point 7: Here, the steam exits the steam turbine after expanding in it and enters the condenser.

Pressure of steam = $P_7 = 0.065$ bar

Temperature of steam = $T_7 = 37.63^\circ\text{C}$

State Point 8: Here, the exhaust steam from the turbine has been condensed in the condenser and the condensate is drawn by the pump. Assuming ideal conditions in the condenser.

Pressure of steam = $P_8 = 0.065$ bar

Temperature of condensate = $T_8 = 37.63^\circ\text{C}$

State Point 9: Here, the condensate which is drawn from the condenser has been pressurized by the pump and this high pressure condensate, which is water, enters into the steam generator for conversion into steam.

Pressure of condensate = $P_9 = 0.065$ bar

Note: T_c and T_h are the inlet and exit temperatures of the cooling water, which in this case is the saline water that is sent into the DDD plant.

Cooling water inlet temperature = $T_c = 25^\circ\text{C}$

Cooling water exit temperature = $T_h = 35^\circ\text{C}$

APPENDIX C
NUMERICAL ANALYSIS RESULTS

Table C-1. List of results from diffusion tower analysis

L_{sw}	G/L_{sw}	Exit Air Temp.	Exit Water Temp.	Exit ω
0.5	0.4	70.0848	50.4327	0.1572
0.5	0.6	66.6292	47.8207	0.1232
0.5	0.8	64.0220	47.2108	0.1024
0.5	1.0	61.9745	47.5116	0.0889
0.5	1.2	60.2114	48.1898	0.0791
0.5	1.4	58.7795	48.9668	0.0724
1.0	0.4	67.7308	49.5627	0.1620
1.0	0.6	63.7433	46.8440	0.1273
1.0	0.8	60.8751	46.1214	0.1061
1.0	1.0	58.7279	46.2363	0.0926
1.0	1.2	57.0560	46.6460	0.0835
1.0	1.4	55.5665	47.1018	0.0757
1.5	0.4	66.4673	49.2259	0.1642
1.5	0.6	62.2439	46.4168	0.1293
1.5	0.8	59.3095	45.5847	0.1084
1.5	1.0	57.1497	45.5467	0.0949
1.5	1.2	55.4478	45.7726	0.0850
1.5	1.4	54.2679	46.0424	0.0797
2.0	0.4	65.6556	49.0584	0.1654
2.0	0.6	61.3137	46.1739	0.1306
2.0	0.8	58.3221	45.2459	0.1092
2.0	1.0	56.1960	45.0900	0.0963
2.0	1.2	54.5095	45.1866	0.0859
2.0	1.4	53.1752	45.3338	0.0781
2.5	0.4	65.0900	48.9687	0.1659
2.5	0.6	60.6759	46.0190	0.1312
2.5	0.8	57.6792	45.0110	0.1100
2.5	1.0	55.5550	44.7609	0.0969
2.5	1.2	53.9666	44.7629	0.0876
2.5	1.4	52.7715	44.8238	0.0812
3.0	0.4	64.6782	48.9225	0.1664
3.0	0.6	60.2212	45.9155	0.1316
3.0	0.8	57.2418	44.8398	0.1110
3.0	1.0	55.1187	44.5130	0.0976
3.0	1.2	53.5444	44.4418	0.0880

Table C-2. List of results from condenser tower analysis (without AAR system)

L_{fw}	L_{fw}/G	Inlet Air	Inlet Water	Inlet ω	Exit Air	Exit Water	Exit ω	m_f
		Temp.	Temp.		Temp.	Temp.		
0.7	1.0	58.780	25.0	0.0724	37.2904	58.7517	0.0225	0.0349
1.4	2.0	58.780	25.0	0.0724	25.1181	46.8352	0.0110	0.0430
2.8	4.0	58.780	25.0	0.0724	25.0236	36.1076	0.0109	0.0430
4.2	6.0	58.780	25.0	0.0724	25.0145	32.4457	0.0109	0.0430
5.6	8.0	58.780	25.0	0.0724	25.0109	30.5990	0.0109	0.0430
7.0	10.0	58.780	25.0	0.0724	25.0096	29.4866	0.0109	0.0430
1.4	1.0	55.567	25.0	0.0757	38.3514	55.5117	0.0297	0.0645
2.8	2.0	55.567	25.0	0.0757	25.5727	46.4552	0.0140	0.0863
5.6	4.0	55.567	25.0	0.0757	25.1231	35.9795	0.0136	0.0869
8.4	6.0	55.567	25.0	0.0757	25.0771	32.3644	0.0136	0.0869
11.2	8.0	55.567	25.0	0.0757	25.0607	30.5392	0.0136	0.0870
14.0	10.0	55.567	25.0	0.0757	25.0530	29.4392	0.0136	0.0870
2.1	1.0	54.268	25.0	0.0797	39.3027	54.1979	0.0353	0.0933
4.2	2.0	54.268	25.0	0.0797	26.2439	46.6485	0.0165	0.1327
8.4	4.0	54.268	25.0	0.0797	25.2817	36.1914	0.0156	0.1347
12.6	6.0	54.268	25.0	0.0797	25.1798	32.5154	0.0155	0.1349
16.8	8.0	54.268	25.0	0.0797	25.1433	30.6551	0.0154	0.1350
21.0	10.0	54.268	25.0	0.0797	25.1262	29.5337	0.0154	0.1350
2.8	1.0	53.175	25.0	0.0781	38.7108	53.0598	0.0355	0.1194
5.6	2.0	53.175	25.0	0.0781	26.6554	45.7156	0.0176	0.1695
11.2	4.0	53.175	25.0	0.0781	25.4241	35.7716	0.0163	0.1730
16.8	6.0	53.175	25.0	0.0781	25.2794	32.2389	0.0162	0.1734
22.4	8.0	53.175	25.0	0.0781	25.2269	30.4488	0.0161	0.1736
28.0	10.0	53.175	25.0	0.0781	25.2000	29.3678	0.0161	0.1737
3.5	1.0	52.772	25.0	0.0812	39.2925	52.6446	0.0389	0.1479
7.0	2.0	52.772	25.0	0.0812	27.3109	45.9445	0.0194	0.2162
14.0	4.0	52.772	25.0	0.0812	25.6206	36.0065	0.0175	0.2229
21.0	6.0	52.772	25.0	0.0812	25.4131	32.4068	0.0173	0.2236
28.0	8.0	52.772	25.0	0.0812	25.3383	30.5787	0.0172	0.2239
35.0	10.0	52.772	25.0	0.0812	25.3001	29.4737	0.0172	0.2241
4.2	1.0	52.409	25.0	0.0824	39.3703	52.2576	0.0405	0.1762
8.4	2.0	52.409	25.0	0.0824	27.8048	45.8220	0.0207	0.2592
16.8	4.0	52.409	25.0	0.0824	25.8033	36.0294	0.0183	0.2691
25.2	6.0	52.409	25.0	0.0824	25.5426	32.4308	0.0180	0.2703
33.6	8.0	52.409	25.0	0.0824	25.4480	30.5997	0.0179	0.2707
42.0	10.0	52.409	25.0	0.0824	25.3980	29.4907	0.0179	0.2710

Table C-3. List of results from condenser tower analysis (with AAR system)

L_{fw}	L_{fw}/G	Inlet Air	Inlet Water	Inlet ω	Exit Air	Exit Water	Exit ω	m_f
		Temp.	Temp.		Temp.	Temp.		
0.7	1.0	58.780	1.0	0.0724	1.3221	54.3816	0.0023	0.0491
1.4	2.0	58.780	1.0	0.0724	1.0268	28.7030	0.0022	0.0491
2.8	4.0	58.780	1.0	0.0724	1.0113	15.0891	0.0022	0.0491
4.2	6.0	58.780	1.0	0.0724	1.0084	10.4432	0.0022	0.0491
5.6	8.0	58.780	1.0	0.0724	1.0074	8.1010	0.0022	0.0491
7.0	10.0	58.780	1.0	0.0724	1.0068	6.6897	0.0022	0.0491
1.4	1.0	55.567	1.0	0.0757	3.8618	54.2327	0.0034	0.1012
2.8	2.0	55.567	1.0	0.0757	1.1378	29.0659	0.0028	0.1021
5.6	4.0	55.567	1.0	0.0757	1.0610	15.2865	0.0028	0.1021
8.4	6.0	55.567	1.0	0.0757	1.0470	10.5782	0.0028	0.1021
11.2	8.0	55.567	1.0	0.0757	1.0406	8.2029	0.0031	0.1021
14.0	10.0	55.567	1.0	0.0757	1.0376	6.7719	0.0028	0.1021
2.1	1.0	54.268	1.0	0.0797	9.4508	53.6830	0.0056	0.1555
4.2	2.0	54.268	1.0	0.0797	1.3213	29.9074	0.0032	0.1607
8.4	4.0	54.268	1.0	0.0797	1.1444	15.7355	0.0031	0.1608
12.6	6.0	54.268	1.0	0.0797	1.1113	10.8824	0.0031	0.1608
16.8	8.0	54.268	1.0	0.0797	1.0975	8.4332	0.0031	0.1608
21.0	10.0	54.268	1.0	0.0797	1.0903	6.9569	0.0031	0.1608
2.8	1.0	53.175	1.0	0.0781	9.8455	52.2957	0.0060	0.2018
5.6	2.0	53.175	1.0	0.0781	1.5096	29.2423	0.0034	0.2093
11.2	4.0	53.175	1.0	0.0781	1.2392	15.3986	0.0033	0.2095
16.8	6.0	53.175	1.0	0.0781	1.1884	10.6573	0.0033	0.2095
22.4	8.0	53.175	1.0	0.0781	1.1656	8.2631	0.0033	0.2095
28.0	10.0	53.175	1.0	0.0781	1.1542	6.8210	0.0033	0.2095
3.5	1.0	52.772	1.0	0.0812	13.2588	52.0631	0.0080	0.2561
7.0	2.0	52.772	1.0	0.0812	1.7606	29.9685	0.0036	0.2715
14.0	4.0	52.772	1.0	0.0812	1.3600	15.7913	0.0035	0.2718
21.0	6.0	52.772	1.0	0.0812	1.2832	10.9234	0.0035	0.2719
28.0	8.0	52.772	1.0	0.0812	1.2508	8.4652	0.0035	0.2719
35.0	10.0	52.772	1.0	0.0812	1.2331	6.9830	0.0035	0.2720
4.2	1.0	52.409	1.0	0.0824	14.8097	51.6654	0.0092	0.3075
8.4	2.0	52.409	1.0	0.0824	2.0095	30.1804	0.0038	0.3300
16.8	4.0	52.409	1.0	0.0824	1.4839	15.9144	0.0037	0.3306
25.2	6.0	52.409	1.0	0.0824	1.3834	11.0084	0.0037	0.3307
33.6	8.0	52.409	1.0	0.0824	1.3411	8.5302	0.0036	0.3308
42.0	10.0	52.409	1.0	0.0824	1.3183	7.0360	0.0036	0.3308

Table C-4. List of results from condenser tower analysis

L_{fw}	L_{fw}/G	Increase in m_f	% Increase in m_f
0.7	1.0	0.0142	40.52
1.4	2.0	0.0061	14.27
2.8	4.0	0.0061	14.14
4.2	6.0	0.0061	14.13
5.6	8.0	0.0061	14.12
7.0	10.0	0.0061	14.12
1.4	1.0	0.0368	57.03
2.8	2.0	0.0157	18.23
5.6	4.0	0.0152	17.52
8.4	6.0	0.0152	17.45
11.2	8.0	0.0152	17.43
14.0	10.0	0.0151	17.42
2.1	1.0	0.0623	66.79
4.2	2.0	0.0279	21.05
8.4	4.0	0.0260	19.34
12.6	6.0	0.0259	19.17
16.8	8.0	0.0258	19.11
21.0	10.0	0.0258	19.08
2.8	1.0	0.0825	69.10
5.6	2.0	0.0398	23.48
11.2	4.0	0.0364	21.05
16.8	6.0	0.0361	20.79
22.4	8.0	0.0359	20.69
28.0	10.0	0.0359	20.65
3.5	1.0	0.1082	73.11
7.0	2.0	0.0552	25.55
14.0	4.0	0.0490	21.98
21.0	6.0	0.0483	21.59
28.0	8.0	0.0480	21.45
35.0	10.0	0.0479	21.38
4.2	1.0	0.1313	74.52
8.4	2.0	0.0708	27.34
16.8	4.0	0.0615	22.87
25.2	6.0	0.0604	22.36
33.6	8.0	0.0600	22.18
42.0	10.0	0.0598	22.09

APPENDIX D
INDUSTRIAL APPLICATION NUMERICAL ANALYSIS RESULTS

Table D-1. List of results from diffusion tower analysis

x	L_{SW}	G	G/L_{SW}	Inlet Air Temp.	Inlet Water Temp.	Exit Air Temp.	Exit Water Temp.	Inlet ω	Exit ω
0	1.5	1.5	1.0	93.0	80.0	62.4	53.4	0.0201	0.0797
0.1	43.0	1.5	28.7	93.0	80.0	-----	-----	-----	-----
0.2	18.9	1.5	12.6	93.0	80.0	-----	-----	-----	-----
0.3	10.9	1.5	7.2	93.0	80.0	-----	-----	-----	-----
0.4	6.8	1.5	4.6	93.0	80.0	62.8	66.1	0.0201	0.1472
0.5	4.4	1.5	2.9	93.0	80.0	62.4	61.7	0.0201	0.1293
0.6	2.8	1.5	1.9	93.0	80.0	62.3	58.0	0.0201	0.1111
0.7	1.7	1.5	1.1	93.0	80.0	62.4	53.8	0.0201	0.0836
0.8	0.6	1.5	0.4	93.0	80.0	63.1	53.5	0.0201	0.0500
0.9	1.5	1.5	1.0	93.0	35.0	48.5	42.1	0.0201	0.0271
1	1.5	1.5	1.0	40.0	35.0	35.7	34.4	0.0201	0.0229

Table D-2. List of results from condenser tower analysis

x	Inlet Air Temp.	Inlet Water Temp.	Exit Air Temp.	Exit Water Temp.	Exit ω	Area of Towers m^2	US gallons of fresh water produced
0	62.4	25.0	25.4	49.6	0.01006	92.93000	221530.92612
0.1	-----	-----	-----	-----	-----	3.70000	0.00000
0.2	-----	-----	-----	-----	-----	7.40000	0.00000
0.3	-----	-----	-----	-----	-----	11.09000	0.00000
0.4	62.8	1.0	1.1	46.3	0.00080	14.79000	68923.03823
0.5	62.3	1.0	1.1	44.6	0.00069	18.50000	81357.57666
0.6	62.3	1.0	1.1	39.8	0.00079	22.18000	83713.06224
0.7	62.4	1.0	1.2	32.4	0.00209	25.88000	72211.85920
0.8	63.1	1.0	1.1	23.0	0.00122	35.74000	59621.31179
0.9	48.5	1.0	1.1	14.4	0.00145	41.95000	36822.42561
1.0	35.7	1.0	1.1	11.3	0.00249	44.67000	31199.52009

Note: Higher air to sea water feed mass flow ratios were not considered.

LIST OF REFERENCES

1. Water: The Power, Promise, and Turmoil of North America's Fresh Water, National Geographic Special Edition, National Geographic Society, Washington D.C., November 1993.
2. A.A. Alawadhi, Regional Report on Desalination-GCC Countries, in: Proceedings of the IDA World Congress on Desalination and Water Reuse, Manama, Bahrain, (2002) 8–13.
3. Introduction to Desalination Technologies in Australia. Retrieved on May 20, 2011 from <http://www.affa.gov.au/content/publication.cfm>
4. K. Bourouni, M.T. Chaibi and L. Tadrist, Water desalination by humidification and dehumidification of air: State of the art, *Desalination*, 137 (2001) 167-176.
5. J.F. Klausner, Y. Li, M. Darwish and R. Mei, Innovative diffusion driven desalination process, *J. of Energy Resources Technology*, 126 (2004) 219-225.
6. Y. Li, Heat and Mass Transfer for the diffusion driven desalination process, PhD dissertation. Department of Mechanical & Aerospace Engineering, University of Florida, Gainesville, Florida, 32611, USA, 2006.
7. J.F. Klausner, Y. Li and R. Mei, Evaporative heat and mass transfer for the diffusion driven desalination process, *J. of Heat and Mass Transfer*, 42(6) (2006) 528-536.
8. Y. Li, J.F. Klausner, R. Mei and J. Knight, Direct condensation in packed beds, *International J. of Heat and Mass Transfer*, 49 (2006) 4751-4761.
9. Y. Li, J.F. Klausner and R. Mei, Performance characteristics of the diffusion driven desalination process", *Desalination*, 196 (2006) 188-209.
10. D. Bharathan, B.K. Parsons and J.A. Althof, Direct-Contact Condensers for Open-Cycle OTEC Applications. National Renewable Energy Laboratory Report SERI/TP-252-3108 for DOE Contract No. DEAC02-83CH10093, 1988.
11. I. Horuz, A comparison between ammonia-water and water-lithium bromide solutions in vapor absorption refrigeration systems, *Int. Comm. Heat Mass Transfer*, Vol. 25, No.5 (1998) 711-721.
12. Pongsid Sriksirin, Satha Aphornratana and Supachart Chungpaibulpatana, A review of absorption refrigeration technologies, *Renewable and Sustainable Energy Reviews*, 5 (2001) 343–372.
13. G. Lorentzen, The use of natural refrigerants: a complete solution to the CFC/HCFC predicament, *Int. J. Refri*, 9. Vol.18, No.3 (1995) 190-197.

14. J. Knight, Heat and mass transfer within the diffusion driven desalination process with heated air, Master's Thesis. Department of Mechanical & Aerospace Engineering, University of Florida, Gainesville, Florida, 32611, USA, 2006.
15. K. Onda, H. Takechi and Y. Okumoto, Mass transfer coefficients between gas and liquid phases in packed columns, J. Chem. Eng. Jpn., 1 (1968) 56-62.

BIOGRAPHICAL SKETCH

Uday Kiran Mahakali was born in Nellore, India. Uday completed his Bachelor of Technology in mechanical engineering from Jawaharlal Nehru Technological University, Hyderabad, Andhra Pradesh, India, in May 2009 after which he joined University of Florida to pursue his Master of Science degree in mechanical engineering.

Uday started working towards his master's at the University of Florida from the fall of 2009. Later, he got the opportunity to be a part of the diffusion driven desalination process research team under the guidance of Dr. James F Klausner. Upon completion of his master's in August 2011, Uday plans to continue contributing to the mechanical engineering industry and build on his knowledge and experience.