Atmospheric water extraction with deep ocean water: a techno-economic assessment

Delft University of Technology

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Atmospheric water extraction with deep ocean water: a techno-economic assessment

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ABSTRACT

The temperature difference ocean layers make seawater a tremendous thermal energy source; the cold deep ocean water can serve as a heat sink and the surface seawater as a heating source. An Ocean Ecopark is an industrial park driven by this ocean thermal energy. It consists of different technologies which make use of warm surface seawater, cold deep ocean water or both.

Fresh water production is one of the potential applications of ocean thermal energy. Seawater can be turned into fresh water with desalination processes, but these technologies are often energy intensive and produce brine as a by-product. Another water source is atmospheric air. An Ocean Ecopark is especially suitable in a tropic climate; the warm temperature leads to heating of surface seawater which creates a large ocean thermal gradient. The high temperature and humidity of tropical air also makes that the air contains a significant amount of water vapour. Cooling the air below its dew point makes it possible to extract this water. This occurs in a similar fashion as when water droplets accumulate on a cold soda can on hot summer days; when the air contacts the can, its temperature drops which makes that the air can hold less vapour. At the point where the air is saturated, condensation will occur. In this case the soda acts as a heat sink, but in an Ocean Ecopark deep ocean water could be the cooling source.

Research on atmospheric water extraction with cold deep seawater is limited and only includes general analyses of condensers used before 1990. After this period fossil fuel prices decreased which lead to less attention for ocean thermal energy. Research did continue on methods for water vapour extraction using surface seawater. This is possible through a humidification-dehumidification cycle; air is first heated and wetted in a humidifier and water vapour is then extracted from this humid air in a dehumidifier. Due to the high temperature of the air in the dehumidifier, surface seawater can be used as a cooling source. Innovative dehumidifiers have been developed as part of humidification-dehumidification research. These dehumidifiers could possibly also be used when deep ocean water is used as the cooling source. In this case, the temperature between the seawater and atmospheric air is already sufficiently large, which might make a humidifier unnecessary.

Attention for ocean thermal energy in recent years has increased and with that the attention for its application in fresh water production technology. This master thesis research investigates the feasibility of atmospheric water extraction with deep ocean water. Previous studies have described the potential for this technology, but the dehumidifiers analysed are likely to be uncompetitive with state of the art desalination technologies. This research therefore investigates whether the use of innovative dehumidifiers can further improve the energy efficiency and cost-effectiveness of atmospheric water extraction with deep seawater.

This research report is subdivided in two parts. The first part of the report discusses dehumidification methods for atmospheric water extraction with deep ocean water and the second part investigates the feasibility of an ocean thermal dehumidifier in an Ocean Ecopark in Curacao. First, the most feasible method of heat transfer for an ocean thermal dehumidifier is determined. Three dehumidifiers are chosen for comparison of which one is commercially available and two are presented in research literature. The performance of the dehumidifiers is determined through a computational analysis. This analysis is based on heat and mass transfer processes during dehumidification and specifications of a commercial heat exchanger. The three technologies are then compared in a multi-criteria analysis to determine which ocean thermal dehumidifier is most suitable in a tropical climate.

The most promising dehumidifier is further analysed in a technical feasibility study. In this study, the influence of operating conditions on the performance of the dehumidifier is investigated. A feasible system configuration is found by optimizing the operating conditions for minimum auxiliary energy requirement and purchased component cost. This system configuration is tested for its financial

feasibility in an economic analysis. The present value method is used to determine its fixed capital cost, operational expenditures and the cost price of water.

The multi-criteria analysis of the three dehumidifiers showed that direct contact dehumidification is more feasible compared to indirect contact dehumidification when non-saturated air and deep ocean water are used. The direct contact dehumidifier requires little maintenance since seawater is only used in the external heat exchanger, it is reliable due to the use of commercial components and the high specific heat transfer area of the packed bed makes it possible to construct a low cost and compact system. Indirect contact plastic heat exchangers also show potential for atmospheric water extraction, but the cost of such a system is currently too high. In addition, significant research and development investments are needed to optimize the system configuration.

The performance of the direct contact dehumidifier is strongly affected by the operating conditions in the packed bed tower and the plate spacing of the external heat exchanger. Simultaneous optimization of the tower and the external heat exchanger is required to minimize the cost and auxiliary energy requirement of the dehumidifier. The influence of the operating conditions in the tower is further investigated. This includes the air to fresh water ratio, inlet fresh water temperature and fresh water mass flux. The following insights were obtained

- **Air to water ratio**
  An increase in ratio leads to a decrease in auxiliary energy requirement and purchased component cost. The decrease in energy requirement only occurs to a certain extent; the increase in ratio leads to less energy loss in the heat exchanger, but an increase in power requirements for the air fan.

- **Fresh water inlet temperature**
  Increasing the fresh water temperature from 8 to 12 degrees Celsius at the inlet of the tower increases the auxiliary energy requirement but decreases component cost.

- **Fresh water mass flux**
  A fresh water mass flux of 0.5 kg/m2*s leads to a lower energy requirement compared to a mass flux of 1.5 kg/m2*s.

A direct contact dehumidifier in Curacao is able to produce almost 5000 litres of fresh water per day with an auxiliary energy requirement of 1.68 kWh per cubic meter of water produced and a purchased component cost of 23,600 euro. The cost price of water produced with this dehumidifier is 3.55 euro per cubic meter of water. The required air mass flow rate is 5.9 kg/s and the required seawater flow rate is 4.4 kg/s. The energy requirement of this system is lower than conventional desalination technologies and the purchased component cost are less than a reverse osmosis system with a similar production rate. The cost price of water produced with the direct contact dehumidifier is 3.55 euro. This is lower than the current water price in Curacao and the price of water produced with a reverse osmosis system. Atmospheric water extraction with deep ocean water is therefore considered feasible; optimization of operating conditions make it possible to produce water at a cost price lower than current water production technologies and its low auxiliary energy requirement make it less sensitive to fluctuating energy prices. Synergies with other ocean thermal energy solutions could further increase the technical and economic viability of the system and should therefore be investigated at the project site. To further commercialize this system, physical testing is required to validate the results of the computational analysis and further investigate the influence of operating conditions and component sizing. In addition, a transient analysis could provide more insight in the variability of the system to changing air temperatures and humidity.

PREFACE

The ocean is one of the largest unexploited energy resources in the world. Innovative technologies are needed to convert this ocean thermal energy into food, water and electricity. Bluerise aims to develop these technologies in an Ocean Ecopark; an industrial park in which ocean energy is the driving force for various thermal applications. Atmospheric water extraction might be one of these technologies. Warm tropical air contains a large amount of water vapour which could be extracted with deep ocean water. This process shows potential for low cost and energy efficient fresh water production with optimal use of available resources.

This report is the result of a master thesis research on atmospheric water extraction with deep ocean water. This graduation project is performed for the Master of Science in Sustainable Energy Technology at Delft University of Technology. The results of this research aim to increase scientific knowledge on atmospheric water extraction with deep ocean water and provide Bluerise with recommendations on the integration of a fresh water production technology in an Ocean Ecopark.

It has both been exciting and challenging to work on such an innovative concept; exciting due to its practical application and promising results, but challenging since little knowledge and data is available. I would therefore like to thank all the people that provided me with advice and took the time to brainstorm with me on how to solve difficulties. In specific, I would like to thank my supervising committee for their guidance during this research project and everyone at Bluerise for their insights and enthusiasm. My gratitude also goes out to my friends and family; Doom for his lessons in transport phenomena and MATLAB, but mostly for always listening to my stories; Marlot for her high fives and encouragement during programming; Laura for reading my thesis and the supportive phone calls; Jeske, Guy and Jan for the endless coffee sessions in the library and physical distractions on the tennis court; and of course my parents who have always supported me throughout my educational career.

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A	area	m^2
a	specific area	$\frac{m^2}{m^3}$
bh	plate spacing	m
C	coefficient	-
Ce	cost	euro
Cn	concentration	$\frac{kg}{m^3}$
Cp	specific heat	$\frac{J}{kg*K}$
Cs	humid heat	$\frac{J}{kg*K}$
D	diffusion coefficient	$\frac{m^2}{s}$
d	diameter	m
dp	effective diameter packing	m
E	effectiveness	-
e	power	W
f	friction factor	-
ff	fanning friction factor	-
Fpd	dry packing factor	$\frac{1}{m}$
Fs	superficial factor	$\frac{m}{s}(\frac{kg}{m^3})^{0.5}$
G	humid air mass flux	$\frac{kg}{m^2*s}$
g	gravity	$\frac{m}{s^2}$
GLF	gas loading factor	$\frac{kg}{m^2*s}$
H	height	m
h	enthalpy	$\frac{kJ}{kg}$
hfg	enthalpy of vaporization	$\frac{kJ}{kg}$

k	thermal conductivity	$\frac{W}{m*K}$
kf	frictional coefficient	-
km	mass transfer coefficient	$\frac{m}{s}$
L	fresh water mass flux	$\frac{kg}{m^2*s}$
LLF	liquid loading factor	$\frac{kg}{m^2*s}$
n	number of tubes	-
P	pressure	Pa
Q	heat flow	W
r	row	-
RH	relative humidity	%
T	temperature	°C
t	thickness	m
U	heat transfer coefficient	$\frac{W}{m^2*K}$
v	velocity	$\frac{m}{s}$
w	absolute humidity	$\frac{\text{kg water vapor}}{\text{kg dry air}}$
Φm	mass flow rate	$\frac{kg}{s}$
Φv	volumetric flow rate	$\frac{m^3}{s}$
ρ	density	$\frac{kg}{m^3}$
η	efficiency	-
μ	dynamic viscosity	$\frac{kg}{m*s}$

Subscripts

A	dry air
av	air vapour mixture
avg	average
c	column
cond	condensate
cross	cross sectional
db	dry bulb temperature
ec	achieved through evaporative cooling
evap	evaporation
f	friction
fw	fresh water
G	humid air
h	heat exchanger
I	interface
i	inside
in	inlet
lat	latent
o	outside
out	outlet

p	heat exchanger plate
s	specific
sat	saturation
sens	sensible
st	static
surf	surface
sw	seawater
Tub	tube
V	vapour
vert	corrected
w	wall

Dimensionless numbers

Pr	Prandtl number
Nu	Nusselt number
Sh	Sherwood number
Sc	Schmidt number
Re	Reynolds number

Punctuation marks

Point	Decimal
Comma	Thousand separator

1. Introduction

1.1 Ocean Ecopark

The ocean consists of three thermal layers: the surface layer, the thermocline and the deep ocean. The surface layer is heated up by the sun and this heat is transferred downwards through wind and waves. Figure 1 shows a temperature-depth ocean water profile. The heat is transferred to the thermocline, but is unable to reach the deeper layers of the ocean. The thermocline therefore acts as a barrier between the warm surface layer and the cold deep ocean. Ocean temperatures vary globally, but in tropical regions the temperature difference between the surface layer and the deep ocean can reach up to 20 degrees Celsius (1).

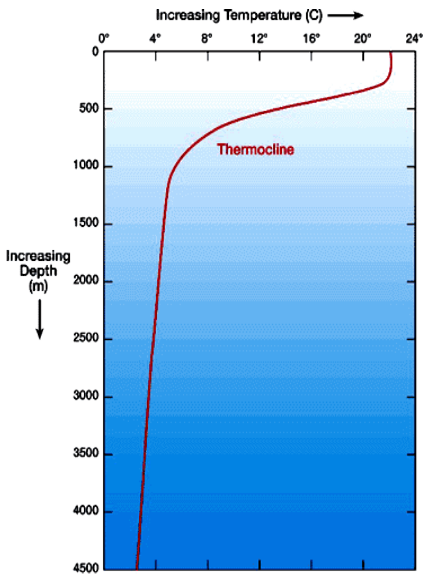

Figure 1 Ocean water temperature-depth profile

Thermal stratification of ocean layers makes the ocean a tremendous energy source. Bluerise is a Dutch company which specializes in using this energy in ocean thermal energy solutions. Bluerise has expert knowledge on Ocean Thermal Energy Conversion (OTEC), Seawater Air Conditioning (SWAC), and the development of Ocean Ecoparks. Figure 2 gives a schematic overview of an OTEC cycle. OTEC technology is based on the temperature difference between the warm ocean surface layer and the deep ocean. A cooling fluid is first heated by the warm seawater and then cooled by the deep ocean water. This establishes a thermodynamic cycle: the cooling fluid evaporates when it is heated and the vapour is used in a turbine to produce electricity. The vapour is condensed when it is cooled by the cold seawater such that it can be reused in the cycle. SWAC technology only makes use of deep ocean water. Since the thermocline acts as a heat barrier, the deep ocean water remains at a constant temperature near 4 degrees Celsius. This cold seawater is used directly as a heat sink in SWAC applications to extract heat from buildings, data centres or other spaces which require cooling.

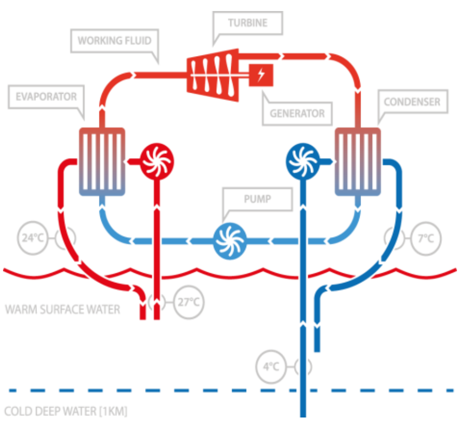

Figure 2: Schematic representationof an OTEC cycle

Both OTEC and SWAC require deep ocean water from depths near 800 to 1000 meter. The size and financial investment of the deep seawater pipeline is significant and can make up to 70 percent of the total project costs (2). Combining OTEC and SWAC systems with other ocean thermal technologies can be financially attractive, since the investment for the deep seawater pipe and the total overhead costs can be spread over the different technologies. An Ocean Ecopark is an industrial park in which these different ocean thermal technologies are combined. The technologies can be placed in a series configuration, in this case the outlet seawater from OTEC or SWAC is reused in other applications or they can be placed in a parallel configuration, when ocean water is directly used in the application. There are a variety of different ocean thermal technologies which could be part of an Ocean Ecopark.

Bluerise is currently researching the feasibility of these technologies and investigating possible configurations of different ocean thermal systems. Figure 3 gives a schematic overview of a possible lay out for an Ocean Ecopark. As can be seen in Figure 3, examples of potential technologies for an Ocean Ecopark are aquaculture, agriculture and desalination.

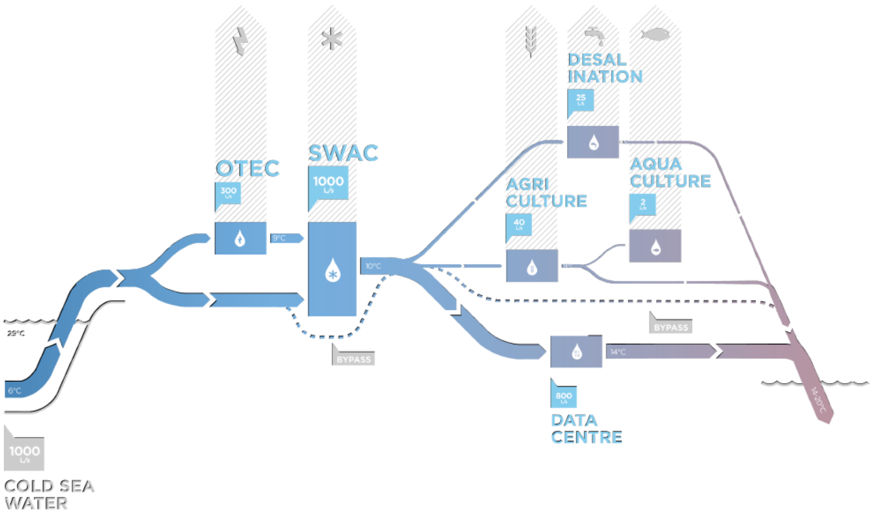

FIGURE 3: CONCEPTUAL LAY OUT FOR AN OCEAN ECOPARK

The Ocean Ecopark is currently under development. The Ocean Ecopark will not only be a means to produce food, water and energy in a cost-effective manner, but also serve as a research and education facility. It provides an opportunity to research new ocean thermal technologies, use this knowledge to construct prototypes and possibly scale up technologies which have proven to work effectively.

1.2 FRESH WATER PRODUCTION IN AN OCEAN ECOPARK

Integration of a fresh water production system in an Ocean Ecopark is currently one of the research areas within Bluerise. Water scarcity is increasing globally (the International Water Management Instituted stated that *"around 1.2 billion people or almost one-fifth of the world's population, live in areas of physical water scarcity, and 500 million people are approaching this situation"*(3)) and there is a need for new technologies that can provide clean water to regions which are lacking fresh water resources. The resources available in an Ocean Ecopark are deep ocean water, surface ocean water and atmospheric water vapour. These resources can be used to produce fresh water through desalination, atmospheric water extraction or humidification-dehumidification cycles. This section will shortly discuss these technologies and their applicability in an Ocean Ecopark.

1.2.1 DESALINATION

Only 2.5 percent of water on earth is fresh water, the remaining 97.5 percent is salt water (4). Desalination of seawater is therefore a commonly used method to obtain fresh water and has a global capacity of 80.9 million cubic meter of fresh water per day (5). Desalination takes place through a thermal process (evaporation of seawater) or a membrane process (separation of water ions from salty water). The thermal process requires a heat source, such as a fossil fuel, solar energy or waste heat. Membrane processes are driven by high pressure pumps or electric fields and therefore require electricity (4). Table 1 gives an overview of the most common desalination technologies (6).

Table 1: Global desalination technologies: capacity by process

	Percentage of global desalination capacity
Reverse Osmosis	53 %
Multi-Stage Flash	25 %
Multiple Effect Distillation	8 %
Other	14%

Desalination technologies are characterized by high energy requirements (electricity or heat) and produce brine as a by-product. Bluerise aims to use ocean thermal energy and find synergies to reduce these energy requirements. For example, one of their current research areas is the combination of an OTEC plant with low temperature thermal desalination. In this process outlet seawater from an OTEC cycle is evaporated in a vacuum chamber and cooled in a condenser.

1.2.2 Atmospheric water extraction

Another significant water resource available in an Ocean Ecopark is the atmospheric air. Globally the air contains about 12,900 km^3 of fresh water, which is similar to the amount fresh water available on inhabited land (4). Figure 4 shows the global atmospheric water content in July (7). This map shows the amount of water that could be produced if all the water vapour is condensed. It can be seen that high humidity ratios are found in regions near the equator. These regions generally experience a tropical climate, which also makes them very suitable for an Ocean Ecopark (2). The warm air heats the surface layer of the ocean and creates a large ocean thermal gradient. Not only does this support the efficiency of OTEC systems but it also leads to high atmospheric water contents: solar irradiation causes evaporation of ocean water and the high air temperatures increase the ability of the air to hold moisture.

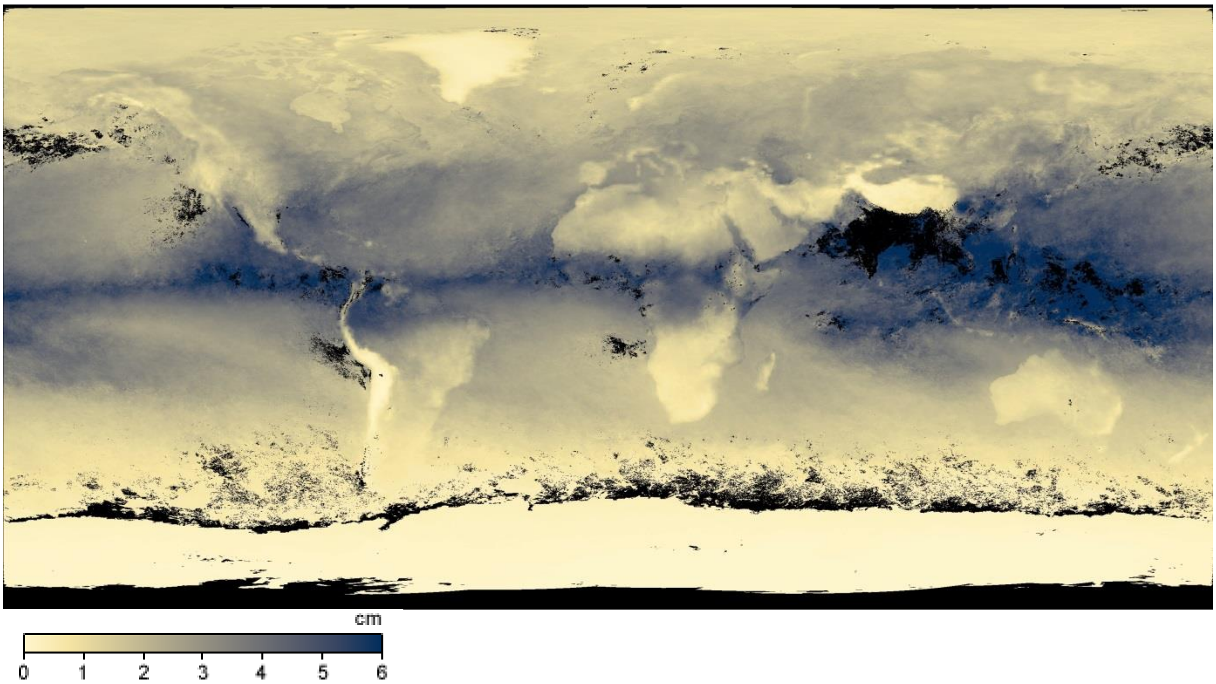

Figure 4: Global atmospheric water content in July 2013

The properties of humid air are clarified on a psychometric chart. A psychometric chart is a graphical representation of temperature, humidity and vapour pressure of the air. Appendix A1 shows a psychometric chart for atmospheric pressures at sea level. The dry bulb temperature on the chart is the temperature of air measured with a thermometer. The relative humidity of the air indicates the amount of water vapour the air is currently holding in comparison to the maximum amount of water vapour the air can hold. A low relative humidity indicates that the air is able to hold more water while a high relative humidity indicates that the air is almost saturated. The actual amount of water vapour in the air is indicated by the humidity ratio which is the ratio between the mass of water vapour in the air to the mass of dry air. For water vapour to condensate, the air needs to reach saturation. At this point the air is not able to hold any more water and therefore the vapour will condensate. The air reaches saturation when it is sufficiently cooled or when the vapour pressure decreases.

An atmospheric water vapour processor (AWVP) is a system which extracts and liquefies water vapour from humid air. AWVP's either use a convection or pressure induced technology, cooling technology, or desiccants for water vapour extraction (4). A decrease in vapour pressure can be achieved through mechanical compression (which requires energy) or by natural convection. This last method makes use of large structures to guide the air to higher altitudes and hence lower atmospheric air pressures. Desiccants extract water vapour from the air by establishing a vapour pressure gradient between the air and the desiccant. After the desiccant is saturated, the water is desorbed by placing the desiccant in a cold air stream. Cooling based systems are driven by thermal energy. In this case a heat sink cools the air below its dew point, which makes the water vapour condense.

Deep seawater could act as a heat sink for cooling based atmospheric water extraction. Gerald and Worzel (1967) already had this conceptual idea in 1967 when they described the use of a 200 meter long and 10 meter high condenser to produce near 4000 m^3 of fresh water per day (8). Costa (1980) continues on this idea and explores different condenser options (9). The model experiments performed as part of this research showed that atmospheric water extraction using deep seawater could be a low technology alternative to conventional desalination plants at that time. In 1984 Seymour and Bothman relate this type of water production to an OTEC plant and conclude that 1.5 MGD of fresh water could be produced with a power requirement of 1.4 MW (10). After 1980 the price of fossil fuels decreased and with that the attention for OTEC technology. Construction of a deep seawater pipeline for solely fresh water production is financially unattractive and is most likely the reason for limited research on atmospheric water extraction using deep seawater.

1.2.3 Humidification-dehumidification cycle

A humidification-dehumidification cycle (HDH) combines the processes of thermal desalination and cooling-based atmospheric water extraction. Air is first heated and wetted with surface seawater in a humidifier. The seawater evaporates and the temperature and the humidity of the air increase. The air is then cooled in a dehumidifier such that the water vapour condenses. There are two types of HDH-cycles: open and closed air cycles (11). An open cycle indicates that new air enters the system for each cycle, while in a closed cycle the exit air at the condenser is reused in the next cycle. Figure 5 and Figure 6 give a schematic representation of the open and closed HDH cycles.

The humidifier requires a heating source and seawater or brackish water for wetting. The heating source can be driven by fossil fuels or a renewable energy source and depends on the resources available at the project location. Orfi, Galanis, & Laplante (2007), for example, describe the potential of a HDH process using solar energy as a heating source (12). Waste heat could be another feasible heating source, as was shown by research at the University of Florida (13). The energy requirement of this HDH system driven by waste heat is 2.2 kWh per cubic meter of water produced, which is comparable to current desalination processes.

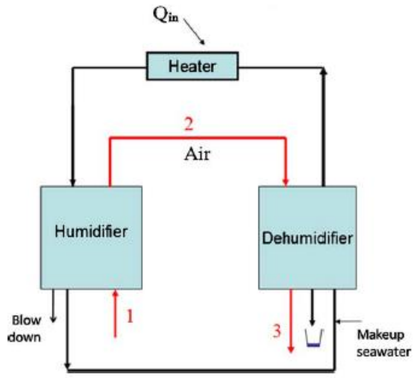

FIGURE 5: OPEN AIR CLOSED WATER HDH-CYCLE

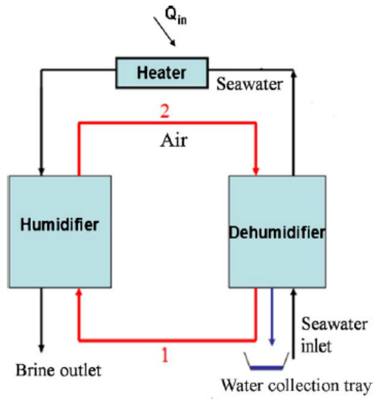

FIGURE 6: CLOSED AIR OPEN WATER HDH-CYCLE

Different types of dehumidifiers are used for the condensation of water vapour in an HDH-cycle. In general, these dehumidifiers can be classified as direct contact dehumidifiers or indirect contact dehumidifiers. In a direct contact dehumidifier heat transfer takes place through direct contact between the air and the cooling fluid. When seawater is used as a coolant, an external fresh water loop needs to be established. In this loop fresh water is cooled by the seawater in an external heat exchanger and used as cooling fluid in the condenser. In an indirect contact dehumidifier, a heat transfer medium is used between the cooling fluid and the air to prevent mixture of the two substances.

Surface seawater is most often used as the cooling fluid in the dehumidifier of HDH-cycle. Since the air is heated before it enters the condenser, the temperature difference with the seawater will still be sufficiently high for condensation. In the Ocean Ecopark deep ocean water is available, which has a temperature of 4 to 6 degrees Celsius. Using this seawater in the condenser would significantly increase the temperature difference and heat transfer between air and water.

1.3 PROBLEM STATEMENT

An Ocean Ecopark aims to provide food, water and electricity using ocean thermal energy. Not only does the Ocean Ecopark serve as an industrial park, it is also a research facility for innovative ocean thermal energy solutions. One of these solutions is fresh water production. Resources for fresh water production in an Ocean Ecopark include deep seawater, surface seawater and atmospheric air. These resources can be used in desalinations systems, AWVP's or HDH-cycles to produce fresh water. Current desalinations technologies have high energy requirements (heat or electricity) and produce brine as a by-product. Bluerise is therefore currently investigating ocean thermal desalination and its synergies with other components in an Ocean Ecopark. AWVP's make use of the water vapour in atmospheric air to produce fresh water. AWVP's show significant potential in an Ocean Ecopark; the favourable climate conditions lead to high atmospheric water contents and air dehumidification is possible with deep seawater as a heat sink. Research on AWVP's using deep seawater showed its potential for low energy and low technology water production, but literature on detailed cost and performance results is limited due to the high initial investments for a deep seawater pipeline. The HDH cycle shows promising results as well, however, a suitable heating source needs to be available. Since deep seawater is at a lower temperature than surface seawater, an opportunity exists to omit the humidifier. Deep seawater is roughly 15 to 20 degrees colder than surface water in tropical regions. A significant temperature difference can therefore still be established in the dehumidifier when atmospheric air is used instead of heated air. A larger air flow would be required to obtain a similar amount of production, but using atmospheric air directly in the dehumidifier would also reduce the need for a humidifier and heating source, which simplifies the technology and could possibly reduce energy and capital cost. In addition, since the air is not wetted with surface seawater, there is no brine production.

This research aims to determine the feasibility of integrating an atmospheric water extraction system in an Ocean Ecopark. This means an AWVP consisting of solely a dehumidifier in which deep seawater is used as a cooling source. In the remainder of this report this will be classified as an ocean thermal dehumidifier. There are various dehumidifiers which could serve as an ocean thermal dehumidifier. The main distinction between dehumidifiers is their method of heat transfer; direct contact and indirect contact between water and air. First, it needs to be determined what method of heat transfer is most suitable for an ocean thermal dehumidifier. The second part of the research will focus on determining the feasibility of the ocean thermal dehumidifier in an Ocean Ecopark in Curacao.

1.4 READING GUIDE

This report is subdivided into two parts; selection of a suitable dehumidification method and a case study to determine the feasibility of an ocean thermal dehumidifier in an Ocean Ecopark in Curacao.

Dehumidification method
To determine which dehumidification method is more suitable for atmospheric water extraction, different dehumidifier types are compared.

Chapter 2 gives a description of the dehumidifiers which are compared in this study and includes a numerical analysis of the heat and mass transfer processes during dehumidification.

Chapter 3 presents the computational analysis to determine the performance of the dehumidifiers in a tropical climate.

Chapter 4 describes the selection of the most suitable dehumidification method in a multi-criteria analysis.

Feasibility study
The computational analysis of the dehumidifiers showed that the performance of the selected dehumidifier could be significantly improved when component sizes and operating conditions are optimized for local conditions. Therefore, a case study for Curacao is performed.

Chapter 5 presents the technical feasibility study for the ocean thermal dehumidifier; this includes an improved computational analysis in which operating conditions are optimized and the selection of a feasible system configuration.

Chapter 6 describes the method and results of the economic analysis. In this study the financial viability of the ocean thermal dehumidifier is determined.

The results of this research are further discussed in chapter 7 and recommendations for future work are presented in chapter 9.

2. Heat and mass transfer in ocean thermal dehumidifiers

Cooling atmospheric air below its dew point shows potential to be an energy efficient method for fresh water production in an Ocean Ecopark due to the availability of warm and humid air and deep ocean water. To do so, heat needs to be extracted from the air to the cool ocean water. This can be done through direct and indirect contact between seawater and air. Direct contact heat exchange systems are known for their high heat and mass transfer coefficients and therefore seem suitable to use in this context. However, direct contact between the seawater and the air is not possible since the condensate would then be contaminated with salt. A cooling loop is therefore required that circulates part of the condensate in the tower and which is cooled by seawater in an external heat exchanger. An indirect contact heat exchanger makes use of a heat transfer medium. A heat transfer medium could, for example, be a tube wall or a plate which separates the air from the seawater. In order to determine which type of heat transfer is more efficient and suitable for application in an Ocean Ecopark, three dehumidifiers were selected for further investigation; a direct contact dehumidifier, a tube condenser and a compact plastic heat exchanger. These systems were selected since they have shown compatibility with seawater and atmospheric air and promising results regarding energy and capital costs have been presented. In addition, expressions for heat and mass transfer processes and experimental data are available in literature or from commercial suppliers.

The tube condenser and compact plastic heat exchanger are both indirect contact dehumidifiers. The tube condenser is a low technology solution and is constructed out of plastic tubes which are individually connected to a framework. The compact plastic heat exchanger is a commercially available system, currently used for heat extraction from high temperature flue gases, and consists of small tubes packed together in a tube bundle. The specific heat transfer area of the compact plastic heat exchanger is significantly higher compared to the tube condenser. Both systems are therefore analysed to determine the effect of the specific heat area on the performance of the dehumidifier. The direct contact dehumidifier investigated in this analysis consists of a packed bed tower and fresh water loop which is cooled in an external heat exchanger.

This chapter describes the dehumidification process of the three dehumidifiers. It presents a numerical analysis of heat and mass transfer processes in the direct contact dehumidifier and tube condenser and commercial data of the compact plastic heat exchanger.

2.1 DIRECT CONTACT DEHUMIDIFIER

The direct contact dehumidifier considered in this analysis is based on the diffusion driven desalination process developed by the University of Florida (13). The heat and mass transfer analysis is derived from their research on condensation in packed bed towers (14). However, this analysis is only suitable when the air is fully saturated and heated. Therefore, calculations are extended for the use of non-saturated air and deep ocean water.

2.1.1 PROCESS DESCRIPTION DIRECT CONTACT DEHUMIDIFIER

This HDH-cycle investigated by the University of Florida consists of a direct contact humidifier in which warm seawater is evaporated (diffusion of water into dry air) and a direct contact dehumidifier in which water vapour is condensed by a cool fresh water loop. Two packed bed towers are used for this process; one for humidification and one for dehumidification. Figure 7 shows a Diffusion Driven Desalination process with an open air loop and feed water heating by waste heat from a power plant. Research results showed that this system could achieve fresh water production with an energy requirement 2.2 kWh per cubic meter of fresh water produced (13).

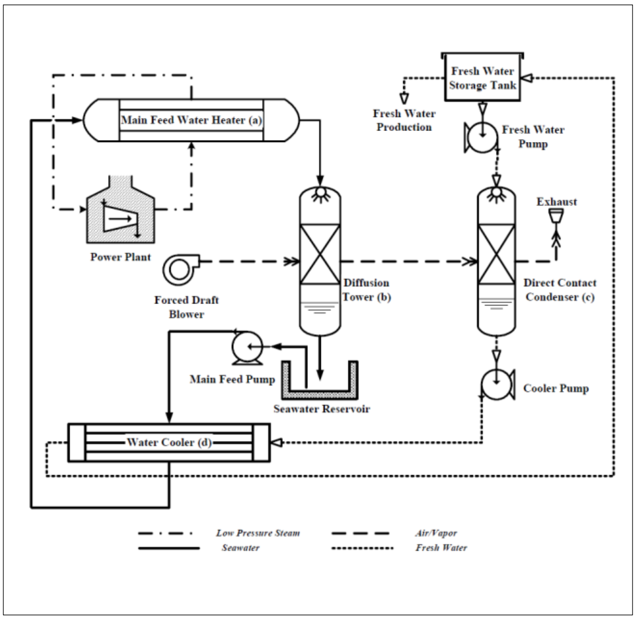

FIGURE 7: DIFFUSION DRIVEN DESALINATION PROCESS DRIVEN BY WASTE HEAT

In the diffusion driven desalination process, an evaporator and heating equipment are used to increase the humidity of the air flowing into the dehumidifier. Since the Ocean Ecopark will be located in tropical regions, atmospheric air humidity ratios will generally be high. Previous studies (13) used air temperatures at the inlet of the dehumidifier of approximately 40 degrees Celsius, while the ambient air temperature will be closer to 27 degrees Celsius. A smaller temperature difference decreases heat transfer rates and thus the efficiency of the system. However, no studies have been done on using atmospheric air in a direct contact dehumidifier with a heat sink of 6 degrees C. This is significantly lower than previous studies (15) and thus could make up for the lower air temperature.

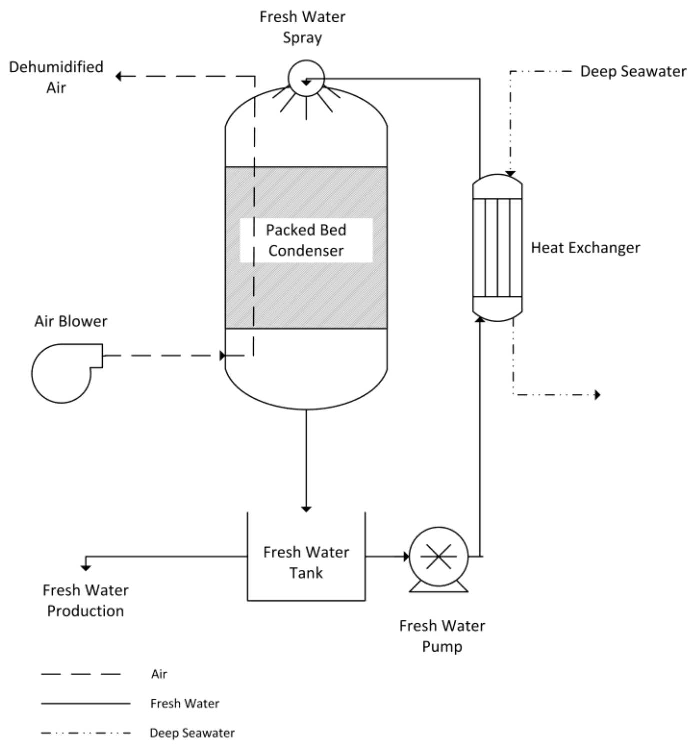

FIGURE 8: DIRECT CONTACT DEHUMIDIFIER

The direct contact dehumidifier used in this analysis is shown in Figure 8. Heat transfer between air and fresh water takes place in a packed bed tower. Fresh water is sprayed over the packed such that a liquid layer is formed. Humid atmospheric air enters at the bottom of the tower and is cooled when it contacts the layer of water on the packed bed. When the air is cooled below its saturation temperature, the water vapour condenses and the liquid condensate is added to the fresh water loop. Part of the fresh water is collected and the remaining fresh water is cooled by the seawater in an external heat exchanger and reused in the packed bed tower.

2.1.2 DIRECT CONTACT HEAT AND MASS TRANSFER ANALYSIS

The numerical analysis presented in this section is based on literature of the diffusion driven desalination process (14). The theoretical basis for direct contact condensation is a differential control volume in which mass and energy are conserved. Figure 9 shows this control volume. Fresh water is sprayed over the packed bed and forms a liquid film. When the air contacts this film, heat transfer takes place due to thermal differences between water and air. If the air is fully saturated, this will lead to condensation and thereby latent heat production.

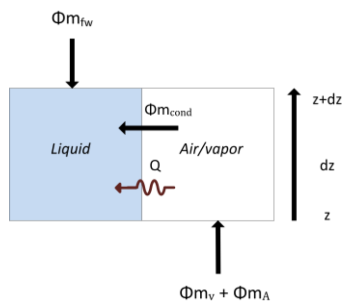

FIGURE 9: DIFFERENTIAL CONTROL VOLUME REPRESENTING DIRECT CONTACT HEAT TRANSFER BETWEEN WATER FILM AND HUMID AIR

Heat and mass transfer coefficients

Heat and mass transfer coefficients indicate the ease of heat and mass transport. These values are required to predict temperatures and humidity throughout the packed bed. The mass transfer coefficient for the liquid and gas side are calculated by Onda's correlations (16). Equation 1 gives the mass transfer coefficient of the fresh water and Equation 2 of the humid air.

$$km_{fw} = 0.0051 * Re_{fw}^{\frac{2}{3}} * Sc_{fw}^{-\frac{1}{2}} * (a * dp)^{0.4} * (\frac{\mu_{fw} * g}{\rho_{fw}})^{\frac{1}{3}} \tag{1}$$

$$km_G = 2 * Re_G^{0.7} * Sc_G^{\frac{1}{3}} * (a * dp)^{-2} * a * D_G \tag{2}$$

These coefficients are linked to the heat transfer coefficient by heat and mass transfer analogy (17). The heat and mass transfer analogy is shown by Equations 3 and 4. The heat transfer coefficients for the fresh water and humid air is expressed according to Equation 5 and 6. The overall heat transfer coefficient is found with Equation 7.

$$\frac{Nu_{fw}}{Pr_{fw}^{1/2}} = \frac{Sh_{fw}}{Sc_{fw}^{1/2}} \tag{3}$$

$$\frac{Nu_G}{Pr_G^{1/3}} = \frac{Sh_G}{Sc_G^{1/3}} \tag{4}$$

$$U_{fw} = km_{fw} * (\rho_{fw} * Cp_{fw} * \frac{k_{fw}}{D_{fw}})^{\frac{1}{2}} \tag{5}$$

$$U_G = km_G * (\rho_G * Cp_G)^{\frac{1}{3}} * (\frac{k_G}{D_G})^{\frac{2}{3}} \tag{6}$$

$$\frac{1}{U} = \frac{1}{U_{fw}} + \frac{1}{U_G} \tag{7}$$

Temperature and humidity gradients

Applying energy and mass conservation equations to the liquid and vapour side of the differential control volume make it possible to define three differential equations which relate air and water temperatures with the absolute humidity of the air. These equations are shown by Equations 8, 9 and 10. A detailed description of the method to obtain this equations can be found in research papers from the University of Florida (18).

$$\frac{dw}{dz} = \frac{dT_G}{dz} \frac{P}{P - P_{sat}} w(C_1 - 2C_2T_G + 3C_3T_G{}^2) \tag{8}$$

$$\frac{dT_{fw}}{dz} = \frac{G}{L} \frac{dw}{dz} \frac{(h_{fg} - h_{fw})}{Cp_{fw}} + \frac{Ua(T_{fw} - T_G)}{Cp_{fw}L} \tag{9}$$

$$\frac{dT_G}{dz} = -\frac{1}{1+w} \frac{dw}{dz} \frac{h_L}{Cp_{av}} + \frac{Ua(T_{fw} - T_G)}{Cp_{av}\,G(1+w)} \tag{10}$$

The equations are still interdependent and cannot be solved separately. The analysis presented by the University of Florida solves this problem by applying a 4th Runge Kutta calculation method. In consultation with researchers at the chemical engineering department of Delft University of Technology, a different approach was chosen in order to simplify the calculations. Equation 8 is first combined with Equation 10 such that all three equations can be linearized. The Euler method is then applied which leads to three expressions that can be solved for each height level in the packed bed. An explanation of the Euler method and the obtained linearized equations can be found in Appendix A2.

2.1.3 Evaporative cooling in a packed bed

Equation 8, which expresses the humidity gradient in the packed bed, is only valid when the air is at 100 percent relative humidity (18). Unsaturated atmospheric air is used in the direct contact dehumidifier and the numerical analysis described in the previous paragraph therefore only applies after the air has reached saturation.

Saturation of air can be achieved through increase in water vapour content or decrease in dry bulb temperature. Evaporative cooling combines these two methods; evaporation of water in the air increases the moisture content of the air but at the same time extracts heat. Figure 10 shows a schematic representation of evaporative cooling on a psychometric chart. Line AB indicates adiabatic evaporative cooling. This means evaporative cooling without heat loss to the environment. Line AC indicates a cooling process in which only the dry bulb temperature of the air is decreased and line AD indicate a process in which the air dry bulb temperature stays equal but the moisture content is increased.

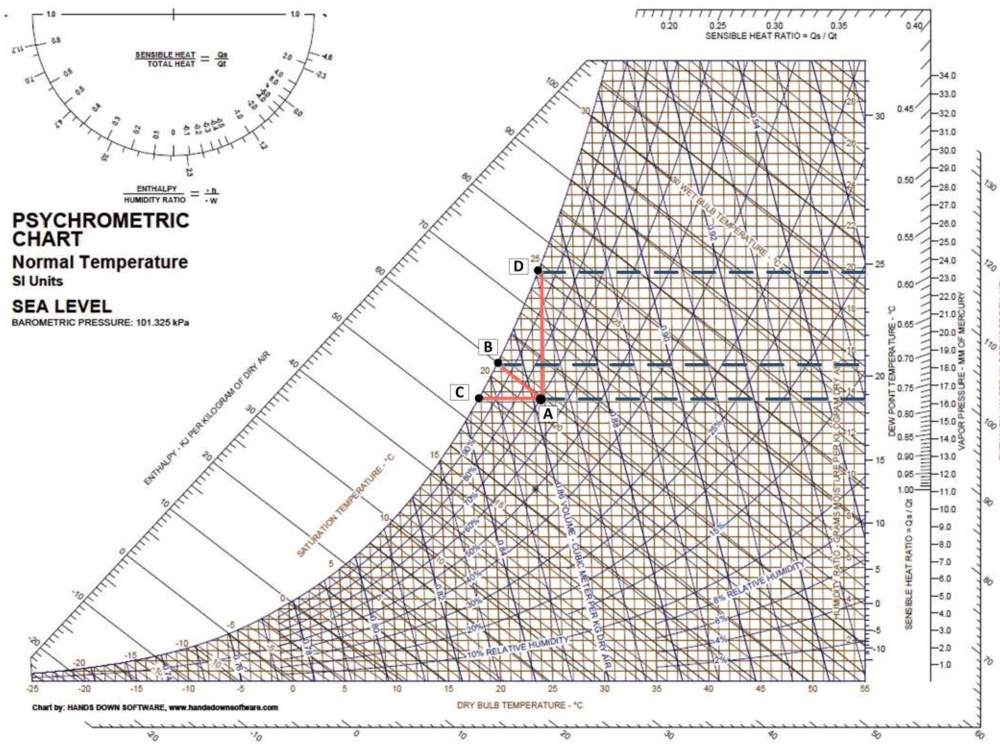

Figure 10: Methods of saturation represented on a psychometric chart

Evaporative cooling takes place when the unsaturated air contacts the liquid film in the packed bed tower. Additional packed bed height is therefore required compared to when saturated air is used in the direct contact dehumidifier. Figure 11 shows a schematic representation of the packed bed tower. In the bottom part of the packed bed tower the air is cooled and humidified through evaporative cooling. The saturated air then enters the top part of the tower where the water vapour condenses.

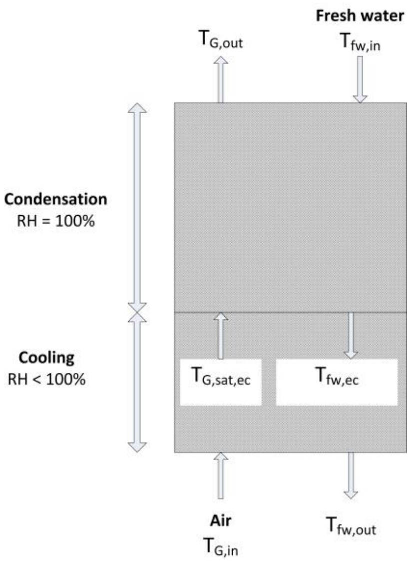

FIGURE 11: EVAPORATIVE COOLING AND CONDENSATION IN THE PACKED BED OF THE DIRECT CONTACT DEHUMIDIFIER

In this analysis it is assumed that evaporative cooling takes places through an adiabatic process and that the temperature of the fresh water remains constant. This indicates that there is no heat gain or loss with respect to the surroundings and heat transfer only takes place within the air stream. The amount of sensible heat removed from the air is equal to the latent heat gained from water evaporation. Applying an energy balance on the air stream gives Equation 11. The first part indicates the energy increase due to an increase in temperature and the second part indicates the latent heat production due to evaporation. The mass of evaporation is determined by the difference between the inlet humidity ratio and humidity ratio at saturation. An iterative approach is then required to find the temperature and humidity ratio of the air at saturation. Appendix A3 shows a detailed description of these calculations.

$$\Phi\mathrm{M}_G * Cp_G(T_{G,in} - T_{G,sat}) = \Phi\mathrm{M}_{evap} * h_{fg} \tag{11}$$

2.2 TUBE CONDENSER

The first indirect contact condenser chosen for further investigation is a plastic tube condenser. This type of condenser has shown promising results regarding energy requirements when surface seawater was used to cool down saturated atmospheric air (19). Material costs are generally high when seawater is used directly in a heat exchanger since the material needs to be corrosion resistant. Since non-saturated air is used in the indirect contact dehumidifier, the required air volume and system size are likely to be large. The material costs could therefore be a limiting factor in the development. The use of plastic shows potential to minimize these costs and therefore seem feasible to use in an indirect contact dehumidifier.

The tube condenser has been constructed and tested as part of a seawater greenhouse (20). In a seawater greenhouse the air enters the condenser after being heated and humidified in an evaporator. Tahri, Bettahar & Douani (2009) present an analysis for the heat and mass transfer in a tube condenser in a seawater greenhouse (19). This analysis is also used in this study, however, an adjustment for the presence of non-condensable gases is made. The high air mass fraction in non-saturated air, makes that the amount of non-condensable gases significantly impacts heat transfer and hence condensation. To account for this effect a different relation for the heat transfer coefficient is used compared to the analysis on a seawater greenhouse.

2.2.1 PROCESS DESCRIPTION TUBE CONDENSER

The tube condenser consists of small plastic tubes in which seawater circulates. Figure 12 shows a top view of the tube condenser. The tubes in the line of air flow are connected and form a row. The number of tubes in a row is indicated by n on Figure 12. Seawater enters at the last tube of this row and leaves at the first tube such that a counter current flow in the horizontal direction between seawater and air is achieved. Several of these tube rows are placed next to each other and together form a matrix configuration. The number of rows is indicated by r on Figure 12. Figure 13 shows a side view of one of the tubes; seawater cools the tube and when air contacts the outside of the tube, water vapour condensates and a liquid film is formed. Due to gravitation the condensate can be collected at the bottom of the tube. Figures 12 and 13 are based on a plastic tube condenser of a seawater greenhouse in Oman (20).

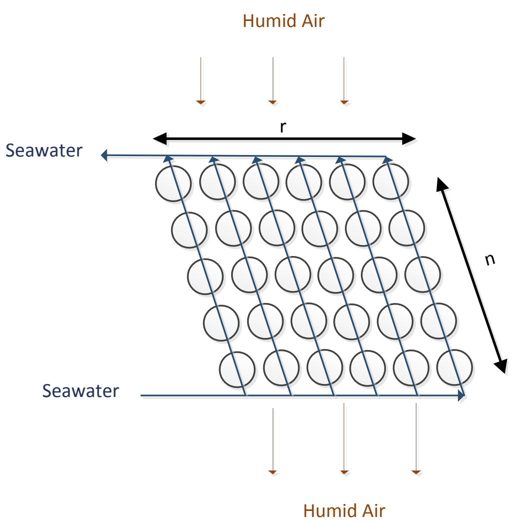

FIGURE 12: TOP VIEW OF TUBE CONDENSER

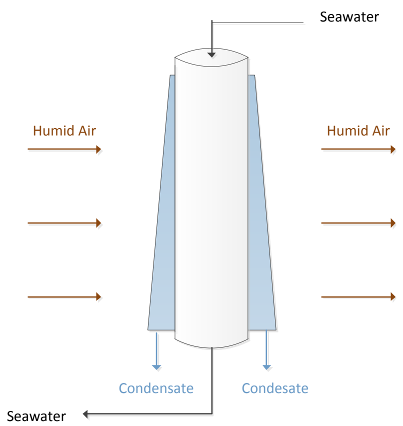

FIGURE 13: SIDE VIEW OF ONE TUBE FROM TUBE CONDENSER

2.2.2 Indirect contact heat and mass transfer analysis

The heat and mass transfer equations used by Tahri, Bettahar & Douani (2009) are used for the numerical analysis of the tube condenser. This section will shortly describe this approach.

Each row of the tube condenser experiences the same heat and mass transfer processes and the approach is therefore to first analyse one row and then reproduce the results for multiple rows. The tubes are analysed separately such that temperatures and humidity ratios can be determined along the length of the condenser. Seawater flows inside the tubes at a temperature below the saturation temperature of the incoming air. Unsaturated air contacts the liquid water film which leads to mixing of the two phases. Part of the air reaches saturation and further cooling leads to condensation.

Heat needs to be transferred from the air through the liquid film and the tube wall before it reaches the cold seawater. Each of these mediums indicates a heat transfer resistance. The total resistance is given by Equation 12. The first term in Equation 12 indicates the heat transfer resistance from the air to the outside tube wall, the second term the resistance in the tube wall and the third term the resistance inside the tube.

$$\frac{1}{U*A_{surf}} = \frac{1}{U_{vert}*A_i} + \frac{1}{2*\pi*k_{tub}*H_{tub}} \mathrm{LN}\left(\frac{D_o}{D_i}\right) + \frac{1}{U_i*A_o} \tag{12}$$

This heat transfer coefficient is then used to calculate the heat transferred from the air to the seawater given by Equation 13.

$$Q = U * A_{surf} * \frac{(T_{G,sat}-T_{sw,in})-(T_{G,sat}-T_{sw,out})}{\mathrm{LN}\left(\frac{T_{G,sat}-T_{sw,in}}{T_{G,sat}-T_{sw,out}}\right)} \tag{13}$$

The heat transferred from the air to the water-air interface is the sum of latent heat due to condensation and sensible heat transfer. The sensible heat transfer is much smaller than the latent heat production and can therefore be ignored. The amount of latent heat production is given by Equation 14. The latent heat is transferred to the outside tube wall, which is expressed by Equation 15. Combining Equations 14 and 15 gives an expression for the mass condensate rate (Equation 16).

$$Q = m_{cond} * h_{fg} \tag{14}$$

$$Q = U_{vert} * A_o * (T_{G,sat} - T_{w,o}) \tag{15}$$

$$m_{cond} = \frac{U_{vert}*A_o*(T_{G,sat}-T_{w,o})}{h_{fg}} \tag{16}$$

2.2.3 Effect of non-condensable gases on heat transfer coefficient

Atmospheric air contains a high amount of non-condensable gases (NCG): atmospheric air at 28 degrees Celsius and 70 percent relative humidity has a NCG mass fraction of 0.98. NCG's can build up near the interface between the liquid water film and water vapour which can lead to a significant reduction in heat transfer during condensation. Previous analyses of the tube condenser (19) accounted for this effect by correcting the average film heat transfer coefficient, which is indicated as U_{vert} in Equation 16, for the presence of NCG's. This value is based on a graph presented in Sacadura (21). However, this graph only suffices for NCG mass fractions of up to 0.1 which is much lower than atmospheric conditions. It is therefore inaccurate to assume a linear relation with the values presented in this graph.

A number of different correlations exist for predicting the effect of NCG's on the heat transfer coefficient. The Tagami correlation, shown in Equation 17, is most suitable to use in this context since it is valid for atmospheric operating conditions and it is based on experiments with forced convection condensation on the outside wall of a vertical tube (22). This equation will therefore be used for calculation of the heat transfer from the air to the liquid water film.

$$U_{vert} = 11.4 + 284 * w_{db} \tag{17}$$

2.3 Compact Plastic Heat Exchanger

The third dehumidifier considered for analysis is a compact plastic heat exchanger. This is also an indirect contact heat exchanger, but it differs from the tube condenser since it consists of smaller tubes. The tubes are made of plastic and have a diameter of approximately 5-8 cm. This makes it possible to construct a compact system with a high specific heat transfer area.

2.3.1 Process description compact plastic heat exchanger

Figure 14 shows a compact plastic heat exchanger developed by Heat Matrix B.V.. In this configuration air enters at the bottom of the heat exchanger and is blown through the tubes with an air fan. The seawater enters at the top of the bundle and flows downwards due to the gravity. The tubes are configured in such a way that only the tube wall separates the air from the seawater.

Figure 14: Compact plastic heat exchanger from Heat Matrix B.V.

2.3.2 Specifications of a commercial compact plastic heat exchanger

The heat exchangers developed by Heat Matrix B.V. are mainly used for the extraction of waste heat from high temperature flue gases. Heat Matrix B.V. provided data on the performance of these systems when cold seawater and atmospheric air would be used. This data is provided in Table 2. Multiple tube bundles are placed in a parallel configuration to increase production. The height of the bundle is constant in this analysis, such that the provided data can be used to calculate the system performance for different sizes.

Table 2: Data of the compact plastic heat exchanger

Air data	Case 1	Case 2
Air mass flow rate (kg/hr)	800	100
Inlet air temperature (C)	28	28
Inlet relative humidity (%)	70	70
Outlet air temperature (C)	18	10
Seawater data		
Seawater mass flow rate (kg/hr)	1080	1080
Inlet seawater temperature (C)	6	6
Outlet seawater temperature (C)	21	10
Water production rate		
Fresh water production (kg/hr)	3.1	0.9

2.4 Conclusion

There are different types of dehumidifiers which could serve as an atmospheric water extraction system in an Ocean Ecopark. The main distinction between these dehumidifiers is the type of heat transfer. A direct contact dehumidifier, tube condenser and compact plastic heat exchanger were chosen for comparison. The numerical analysis for the direct contact dehumidifier and the tube condenser presented in literature is sufficient to establish a theoretical basis but needs adjustment for the use of non-saturated air and deep ocean water. A commercial compact plastic heat exchanger is also included in this study since the smaller tube diameter, compared to the tube condenser, could possibly lead to a smaller size system. Heat Matrix B.V. provided heat and mass transfer data on this type of heat exchanger which includes production rates and air and water outlet temperatures.

3. Performance of ocean thermal dehumidifiers in a tropical climate

Chapter two described three technologies which show potential for atmospheric water extraction in an Ocean Ecopark. In this chapter the performance of these ocean thermal dehumidifiers is determined. The optimal operating conditions and component sizes are unknown at this point, since experiments found in literature and in commercial applications focus on the use of high temperature humid air and surface seawater for cooling. The number of unknown variables for the design of these dehumidifiers is extensive, which makes an optimization study complex and time consuming. For this research it was therefore chosen to compare the systems based on their predicted performance in an Ocean Ecopark. Section 3.1, describes this approach in more detail. Due to the large number variables, which include both operating conditions and sizing of components, only a limited number of system configurations are tested. A computational analysis is performed to determine the performance of these configurations. The computational models used for this analysis are described in section 3.2. The results of the computational analysis are presented in section 3.3.

3.1 APPROACH

The operating conditions and component sizes for optimal performance of the dehumidifiers is currently unknown. The systems described in literature provide data on sizing and operating conditions, however, these designs are based on the use of surface seawater and heated air. The number of unknown variables for both the operating conditions and component sizing of the dehumidifiers is extensive and an optimization study for all of these variables requires a complex computational program. Since this section is aimed at finding the most feasible type of heat transfer and not an optimal system configuration, a different approach is taken.

3.1.1 SELECTION OF DESIGN PARAMETERS FOR OCEAN THERMAL DEHUMIDIFIER

First, a basic system configuration is set up including values for the operating conditions and component sizes. These values are based on the research data found in literature and the expected deviation from this data due to the use of atmospheric air and deep seawater. The critical elements of the design are then determined; these are the operating conditions and system sizes indicated in literature which have the most impact on the performance of the dehumidifier. In the computational analysis these design elements are varied individually to explore if they have the same impact when non-saturated air and cold seawater are used. This gives different system configurations for each dehumidifier; a basic configuration based on values from literature and configurations in which one of the critical design elements is varied. Table 3 gives an overview of the different system configurations used in the computational analysis. Appendix A5 shows the supporting literature and the analysis that was used for creating the different configurations.

TABLE 3: SYSTEM CONFIGURATIONS TESTED FOR PERFORMANCE IN A TROPICAL CLIMATE

Dehumidifier	Configuration name	Design parameter	Value of design parameter (basic – adjusted)
Direct Contact Dehumidifier	**DCD- Basic**	-	-
	DCD-Height	Tower height (m)	0.5 – 1
	DCD-Air rate	Air mass flux (kg/m2*s)	0.5 – 1
	DCD-Water rate	Fresh water mass flux (kg/m2*s)	0.8 – 0.4
Tube Condenser	**Tube-Basic**	-	-
	Tube-Tubes	Number of tubes in a row	16 – 65
	Tube-Velocity	Air velocity (m/s)	5 – 1
Compact Plastic Heat Exchanger	**Compact-Basic**	-	
	Compact-Velocity	Air velocity (m/s)	9 – 1.1

3.1.2 SIMULATION METHOD

The ocean thermal dehumidifiers are simulated in MATLAB using the computational models described in section 3.2. MATLAB was used instead of a commercial design program due to large amount of unknown variables. Design programs are suitable when the operating conditions are specified and the sizing of the system is unknown. However, in this case the optimal operating conditions and sizing for each dehumidifier still need to be determined.

The inlet conditions of seawater and air and the required water production is equal for all simulations in order to make the systems comparable. The inlet air temperature and relative humidity are taken as 28 degrees Celsius and 70% respectively. These values are based on data from Bluerise for typical temperature and humidity in a tropical climate. It is a steady state analysis and variations in temperature and relative humidity are not taken into account. The temperature of the available seawater temperature is taken as 6 degrees Celsius. There is no restriction on the required mass flow rate of the seawater. However, minimization of seawater use is preferred. The required water production is kept at a constant value for all three systems at 7200 L per day. This value is chosen based on a sizing recommendation from Bluerise and available data on a similar sized reverse osmosis system which makes it possible to accurately benchmark results.

TABLE 4: BOUNDARY CONDITIONS USED IN SIMULATIONS

Air	
Inlet air temperature (C)	28
Inlet relative humidity (%)	70
Seawater	
Inlet seawater temperature (C)	6
Water Production	
Production rate (L/d)	7200

3.2 COMPUTATIONAL MODELS

Three computational models are constructed which are able to simulate the ocean thermal dehumidifiers. The computational models for the direct contact dehumidifier and the tube condenser are based on the numerical analysis on heat and mass transfer processes presented in chapter 2. These calculations are excluded from the computational model for the compact plastic heat exchanger since data was provided by Heat Matrix B.V.

All three computational models include integrated calculations for the auxiliary energy requirement and purchased component costs of the dehumidifiers. The calculation method for the auxiliary energy requirement for each of the dehumidifiers is presented in this section. The component costs are also directly integrated in the computational model. The cost equations need to be variable for different operating conditions. Therefore, equations were formed by adding a trend line to commercial cost data for different component sizes.

3.2.1 DIRECT CONTACT COMPUTATIONAL MODEL

Calculation approach for the auxiliary energy requirement

An air fan is required to blow the air through the packed bed, a fresh water pump is needed for the fresh water loop and a seawater pump is required to pump the seawater through the external heat exchanger. In this case it is assumed that the seawater is delivered to the site and therefore only the pressure drop in the heat exchanger is taken into account.

Energy requirement air fan

The auxiliary energy required for the air fan is determined by the pressure drop of the air throughout the packed bed. A number of methods exist to calculate the pressure drops on the gas and liquid side of the packed bed. In this analysis, Robbins method (23) was chosen since it is especially suitable for packed beds with a low liquid loading. The presence of a liquid in a packed bed increases the pressure drop since it decreases the available cross section for gas flow. Robbins method calculates the total pressure drop by adding the dry pressure drop and the pressure drop caused by the liquid. First, the liquid loading factor is determined. This is based on the type of packing used and its dry packing coefficient. For a packing coefficient higher than 200 Equation 18 is used to calculate the liquid loading factor, for a packing coefficient lower than 15 Equation 19 is used and for a factor in between these values Equation 20 is used. These equations are based on English units. The fresh water mass flux is therefore given in lb/hr*ft^2, the density in lb/ft^3 and the dynamic viscosity in Cp.

$$LLF = L * \frac{62.4}{\mathrm{P}_{fw}} * \left(\frac{Fpd}{20}\right)^{0.5} * \mu_{fw}{}^{0.2} \tag{18}$$

$$LLF = L * \frac{62.4}{\mathrm{P}_{fw}} * \left(\frac{20}{Fpd}\right)^{0.5} * \mu_{fw}{}^{0.1} \tag{19}$$

$$LLF = L * \frac{62.4}{\mathrm{P}_{fw}} * \left(\frac{Fpd}{20}\right)^{0.5} * \mu_{fw}{}^{0.1} \tag{20}$$

The gas loading factor is determined by the superficial F-factor (Equation 21) and the dry packing factor.

$$Fs = v_G * \mathrm{P}_G{}^{0.5} \tag{21}$$

$$GLF = 986 * Fs * (\frac{Fpd}{20})^{0.5} \tag{22}$$

Equation 23 gives the dry pressure drop and Equation 24 the pressure drop due to the liquid loading. In these equations the pressure drop is indicated in inches of H_2O per feet of packing. This value is converted to metric units and multiplied by the height of the tower to obtain the total pressure drop through the packed bed.

$$\Delta P_{dry} = 7.4 * 10^{-8} * GLF^2 * 10^{2.7*10^{-5}*LLF} \tag{23}$$

$$\Delta P_{liquid} = 0.4 * \frac{LLF}{20000}^{0.1} * (7.4 * 10^{-8} * GLF^2 * 10^{2.7*10^{-5}*LLF})^4 \tag{24}$$

The difference between the pressure drop calculated with Robbins method and with a formula from a packing manufacturer (14) is 33 Pa. This is insignificant compared to the total pressure drop and the Robbins method therefore seems suitable for prediction of the pressure drop in the packed bed of the direct contact dehumidifier. The total power requirement of the air fan is calculated using Equation 25 which takes into account an air fan efficiency of 80 percent.

$$e_{air\,fan} = \frac{\Phi \mathrm{v}_G * \Delta P_G}{\mathrm{H}_{air\,fan}} \tag{25}$$

Fresh water pump and seawater pump

The total height of the column consists of the bottom (evaporative cooling) and upper (condensing) part of the packed bed and additional space for spray droplets and air to disperse. The additional height required for dispersion depends on the liquid and gas loading rates and the type of blowers and spray nozzles used. These are unknown at this point and therefore the additional height needed for air and water dispersion is kept the same as in the experiments performed at the University of Florida (14) at 1.5 meter. The pressure drop required to pump the fresh water to the top of the tower is calculated by Equation 26. The pressure drop in the nozzles is based on commercially available nozzles (20 kPa). Nozzles with a spray angle of 120 degrees are chosen, such that only one nozzle is needed for a cross sectional area of one square meter (24).

$$\Delta P_{st,fw} = \rho_{fw} * g * H_c \tag{26}$$

An external heat exchanger is required for heat transfer from the fresh water to the seawater. The required capacity and the design of the heat exchanger depend on the mass flow rate and inlet and outlet temperatures of the fresh water. Since these values are subject to further system optimization, the pressure drop throughout the heat exchanger is assumed to be similar to a commercial heat exchanger. Table 4 shows the design data of a commercial plate heat exchanger. This data is provided by Kapp B.V., a company specializing in the design of heat exchangers. Since the heat exchanger needs to process seawater, material costs will likely be high. A plate heat exchanger was therefore chosen for this analysis since it can achieve high heat transfer coefficients which minimize the required heat exchange surface area and material costs (25).

TABLE 5: SPECIFICATION COMMERCIAL PLATE HEAT EXCHANGER

Operating conditions	
Seawater to fresh water mass flow ratio	*0.82*
Heat capacity (kW)	*567*
Log mean temperature difference (K)	*2.47*
Pressure drop fresh water (kPa)	*49*
Pressure drop seawater (kPa)	*34*

The pressure drop can only be considered constant for the heat exchanger in the dehumidifier, when the dimensions of the flow channel are similar to the commercial heat exchanger. An increase in the total fresh water mass flow rate leads to more plates and flow channels but the mass flow rate and velocity through each channel remains equal. The pressure drop and heat transfer coefficients are can therefore be considered as a constant value. The deviation in fresh water temperature could have an effect on the heat transfer and the required heat exchange area. To account for this effect, the log mean temperature difference is kept in a similar range as the commercial heat exchanger design.

The total power requirement for the seawater pump and the fresh water pump are then found with Equation 27 and 28 respectively.

$$e_{seawater\ pump} = \frac{\Phi V_{sw} * \Delta P_{sw}}{\mathrm{H}_{seawater\ pump}} \tag{27}$$

$$e_{fresh\ water\ pump} = \frac{\Phi V_{fw} * \Delta P_{fw}}{\mathrm{H}_{fresh\ water\ pump}} \tag{28}$$

Computational model description

Predicting the fresh water temperature, air temperature and humidity ratios at different heights of the packed bed is complex, since these values are interdependent. Li et al (2006) presents guidelines for setting up an iterative computational code which makes it possible to predict water and air properties at different heights of the packed bed (14). This code is reconstructed in this research and modified for the use of atmospheric air and deep ocean water. Figure 15 is a flow diagram of the constructed direct contact computational model to predict temperatures and humidity throughout the packed bed. The processes in green indicate the parts which have been modified from the computational analysis presented by Klausner. These modifications include calculations of the air saturation temperature and humidity, the height of the packed bed required to reach air saturation and the use of the Euler method to calculate the air and water properties throughout the packed bed. These calculation methods are described in the numerical analysis section of the direct contact dehumidifier. Another addition to the model presented by Klausner is the integration of a calculation process for the energy requirement and component costs of the dehumidifier.

This section will shortly describe the computational programming code based on Figure 12. The numbers in brackets relate to the blocks shown in the flow diagram. The computational model starts with an estimate for the saturation temperature of the air [1]. A new value for the saturation temperature is then calculated based on the formulas presented in Appendix A3 [2]. This value is compared to the initial estimated value; if these are not equal a new estimation is made [3].This iterative process continues until a matching value is found. The calculated saturation temperature and humidity are then used to find the required height of the packed bed for saturation [4]. The values for air saturation temperature, humidity and height are starting values for the calculation of air and water properties during condensation [5]. First the temperature of the fresh water at the saturation height is estimated [6]. The temperature of the fresh water, air and humidity ratio are then calculated for each step height based on the formulas derived from Equation 8, 9 and 10 [7]. At the end of the packed bed the temperature of the fresh water is compared to the inlet fresh water temperature, defined at the start of the computation [11]. If these values are equal, the iteration will stop, if not a new value for the fresh water temperature is estimated.

The calculated air and water properties at the inlet and outlet of the packed bed tower are then used to calculate the auxiliary energy requirement and component cost of the dehumidifier. First the required cross sectional area is determined based on the production rate and total required water production [13]. This value is then used to calculate to the total mass flow rate of fresh water and air [14]. Based on these values the energy and cost calculations per component are made [15].

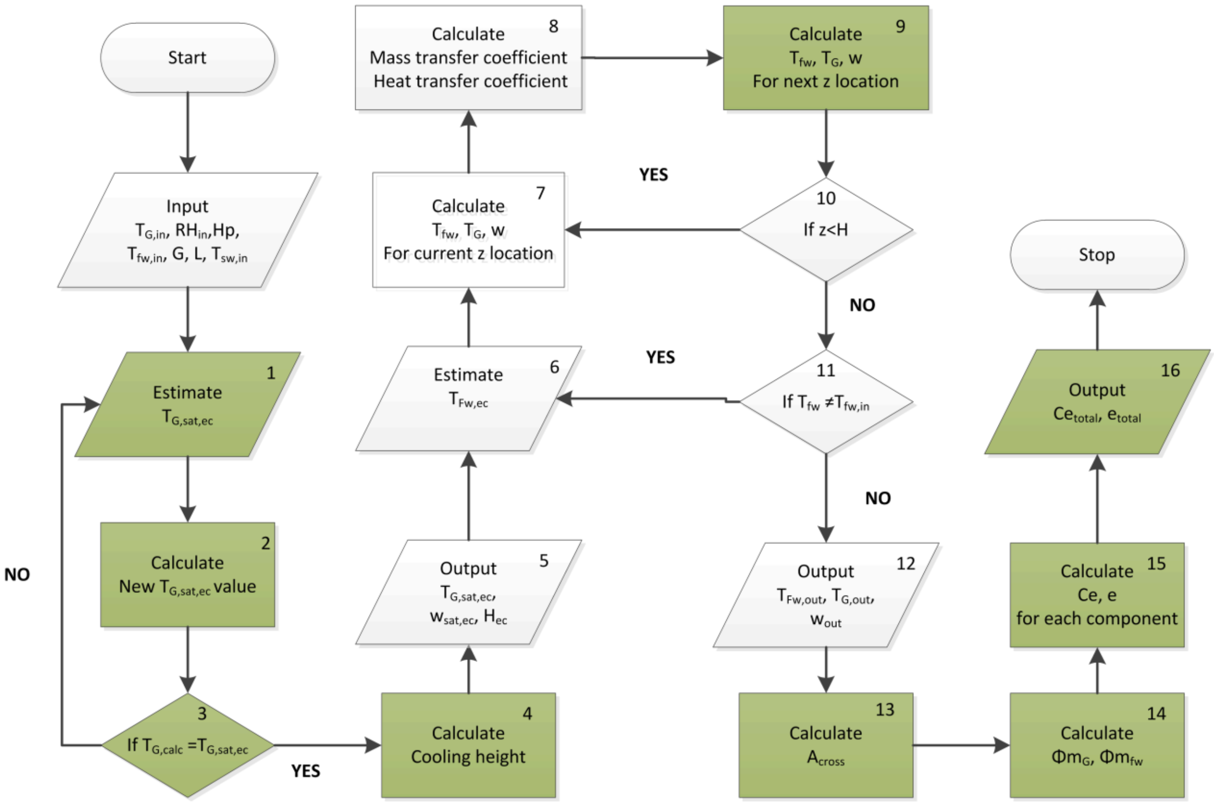

FIGURE 15: FLOW DIAGRAM OF THE DIRECT CONTACT COMPUTATIONAL MODEL

Validation of the direct contact computational model

The calculation approach for condensation of saturated air in a packed bed deviates from the method found in literature since the Euler method was used to calculate temperatures and humidity instead of 4th Runge Kutta. To validate this method, results are compared with data from experiments done at the University of Florida (14). Figure 16 shows the data points for the physical experiments together with the results obtained from the direct contact computational model (dc-computational model). The results for water and air temperature obtained with the computational model are close to the measured results (difference of tenths of degrees) and the maximum difference between the model results and the measured data for the humidity ratio is 0.0035 gram of water per gram of dry air. For an air mass flow of 300 kg/s (assuming a water to air ratio of 1 and 25 percent water vapour extraction) this relates to a difference of 6 percent on the total production. This can be considered as a small difference and therefore the direct contact computational model will be used for analysis of the direct contact dehumidifier.

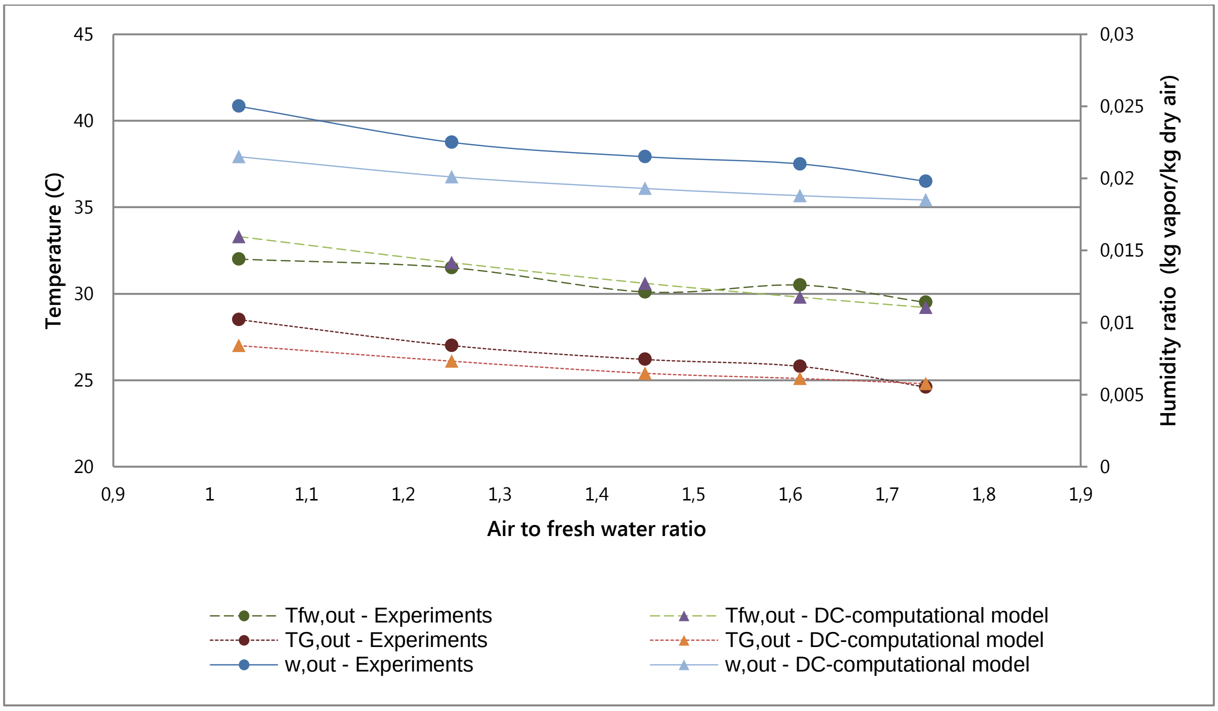

FIGURE 16: AIR AND WATER PROPERTIES IN THE DIRECT CONTACT DEHUMIDIFIER: COMPARISON OF RESULTS FROM DIRECT CONTACT COMPUTATIONAL MODEL AND EXPERIMENTAL DATA FROM THE UNIVERSITY OF FLORIDA

3.3.1 TUBE CONDENSER COMPUTATIONAL MODEL

The numerical analysis presented by Tarhi, Bettahar & Douani (2009) is applicable for the tube condenser in an Ocean Ecopark, but it makes use of measurement data of air and seawater properties at the inlet and outlet of the condenser. This data cannot be used for the analysis of a condenser in an Ocean Ecopark, since the properties of air and seawater deviate from the seawater greenhouse. A new calculation method is therefore added to the numerical analysis of the tube condenser to account for the lacking data. In addition, the analysis is extended with calculations for the auxiliary energy requirement and the purchased component costs of the tube condenser.

Calculation approach auxiliary energy requirement

A seawater pump is required for pumping the seawater through the tubes and an air fan is needed to blow the air through the condenser. The auxiliary energy requirement of the seawater pump is derived from the energy losses due to the static pressure drop and a frictional pressure drop. The pressure drop due to an increase in potential energy is given by Equation 29 and is based on the height of the tube. The frictional pressure drop is given by Equation 30. This equation takes into account frictional losses related to the tube length and the connections between the tubes. The total power requirement of the seawater pump is found using Equation 31.

$$\Delta P_{sw,st} = \rho_{sw} * g * H_{tub} \tag{29}$$

$$\Delta P_{sw,f} = \rho_{sw} * \left(\left(4 * f * n * \frac{H_{tub}}{D_i} \right) + (kf_{in} + kf_{out} + (n-1) * kf_{bend}) \right) * 0.5 * v_{sw}{}^2 \tag{30}$$

$$e_{seawater\ pump} = \frac{\Phi\mathrm{v}_{sw} * (\Delta P_{sw,p} + \Delta P_{sw,f})}{\mathrm{H}_{seawater\ pump}} \tag{31}$$

The pressure drop the air needs to overcome to flow through the condenser is calculated using Equation 32 (26). In this equation, Bo represents a coefficient for the use of unbaffled tubes and f a friction factor for flow through staggered tubes. The pressure drop is based on the friction which the air experiences when it flows on the outside of the tubes. The power requirement for the air fan is then calculated by multiplying the pressure drop by the required air volumetric flow rate and accounting for 80 percent fan efficiency (Equation 33).

$$\Delta P_{G,f} = \frac{2 * Bo * f * n * G^2}{\rho_G} \tag{32}$$

$$e_{air\ fan} = \frac{\Phi\mathrm{v}_G * \Delta P_{G,f}}{\mathrm{H}_{air\ fan}} \tag{33}$$

Calculation approach for seawater temperature and relative humidity

Based on the numerical analysis presented in the previous section, a computational model for the tube condenser is constructed. The computational analysis presented by Tahri, Bettahar, & Douani (2009) (20) is based on measurements of relative humidity and seawater temperature at the inlet and outlet of the tube condenser. These values are used to calculate the humidity and seawater temperature at each tube by assuming a linear increase between inlet and outlet conditions. However, the air used in a seawater greenhouse first flows through an evaporator before it reaches the condenser and is therefore close to saturation and at a higher temperature than atmospheric air. When deep seawater is used instead of surface seawater the temperature difference between the air and seawater will be different than described in literature. It is therefore not possible to directly use the measured data in the computational analysis for the indirect contact dehumidifier.

The numerical analysis presented by Tahri, Bettahar, & Douani (2009) makes it possible to calculate the air temperature and humidity ratio after each tube. This then makes it possible to calculate the dry bulb and saturated vapour pressure and the relative humidity. This is shown by Equation 34.

$$RH_{out} = \frac{P_{v,db}}{P_{v,sat}} * 100 \tag{34}$$

The exit air temperature and humidity at each tube are calculated based on the heat transfer to the seawater. The amount of heat transfer is related to the increase in seawater temperature per tube. Since no measured data is available, an estimate is made for the increase in seawater temperature per tube. The relative humidity values are then analysed in order to check if the initial value for seawater temperature increase is correct; when the relative humidity value is higher than 100 percent, it indicates that the partial pressure of the vapour is higher than the saturation pressure. This is not possible and hence the computational model either overestimates the value for outlet humidity or underestimates the outlet air temperature. These values are based on the value for increase in seawater temperature and it therefore indicates that this assumption is incorrect. The value is therefore adjusted until the relative humidity is below 100 percent. To validate the results, an energy balance of the total system is used to calculate the total increase in seawater temperature.

$$\Phi M_{sw} * Cp_{sw} \Delta T_{sw} * n = \Phi M_A * Cp_A * (T_{A,in} - T_{A,out}) + \Phi M_{V,out} * Cp_V * (T_{A,in} - T_{A,out}) + \Phi M_{cond} * Cp_V * (T_{A,in} - T_{cond}) + \Phi M_{cond} * hfg \tag{35}$$

Description of computational model

Figure 17 is a flow diagram of the tube condenser computational programming code. The processes in green indicate the parts which have been adjusted from the numerical analysis presented in literature. The adjusted processes are the calculation of seawater temperature and relative humidity and the use of the Tagami relation for calculation of the heat transfer between the air and the liquid water film. The first step in the computational program, is the estimation of the increase in seawater temperature per tube [1]. Following, the saturation temperature is estimated for the current tube [2]. A new saturation temperature is calculated according to equations presented in Appendix A3 [3]. The calculated and estimated saturation temperatures are then compared [4]. If the values are not equal, a new estimation is made. When an equal saturation temperature is found, the inside wall temperature and the outside wall temperature are estimated [5]. A new value for these temperatures is found based on an iterative process. In this process the Tagami relation is used to calculated the heat transfer coefficient due to non-condensable gases [6]. Once the calculated temperatures are equal to the estimated temperatures, the condensate per tube is calculated and the air properties leaving the tube [8]. These values are then used for the heat transfer calculations for the next tube in this row. Once this calculation process has been performed for all tubes, the relative humidity at the outlet of the condenser is determined [10]. If this value is higher than 100 percent, a new estimation is made for the increase in seawater temperature. This process continues until a relative humidity of less than 100 percent is found. The required number of rows are then determined by comparing the production rate per row to the total required production rate [13]. Based on this configuration the total required seawater flow rate and air flow rate are calculated [14]. The auxiliary energy requirement and component cost are then calculated in the last step of the computational model [15].

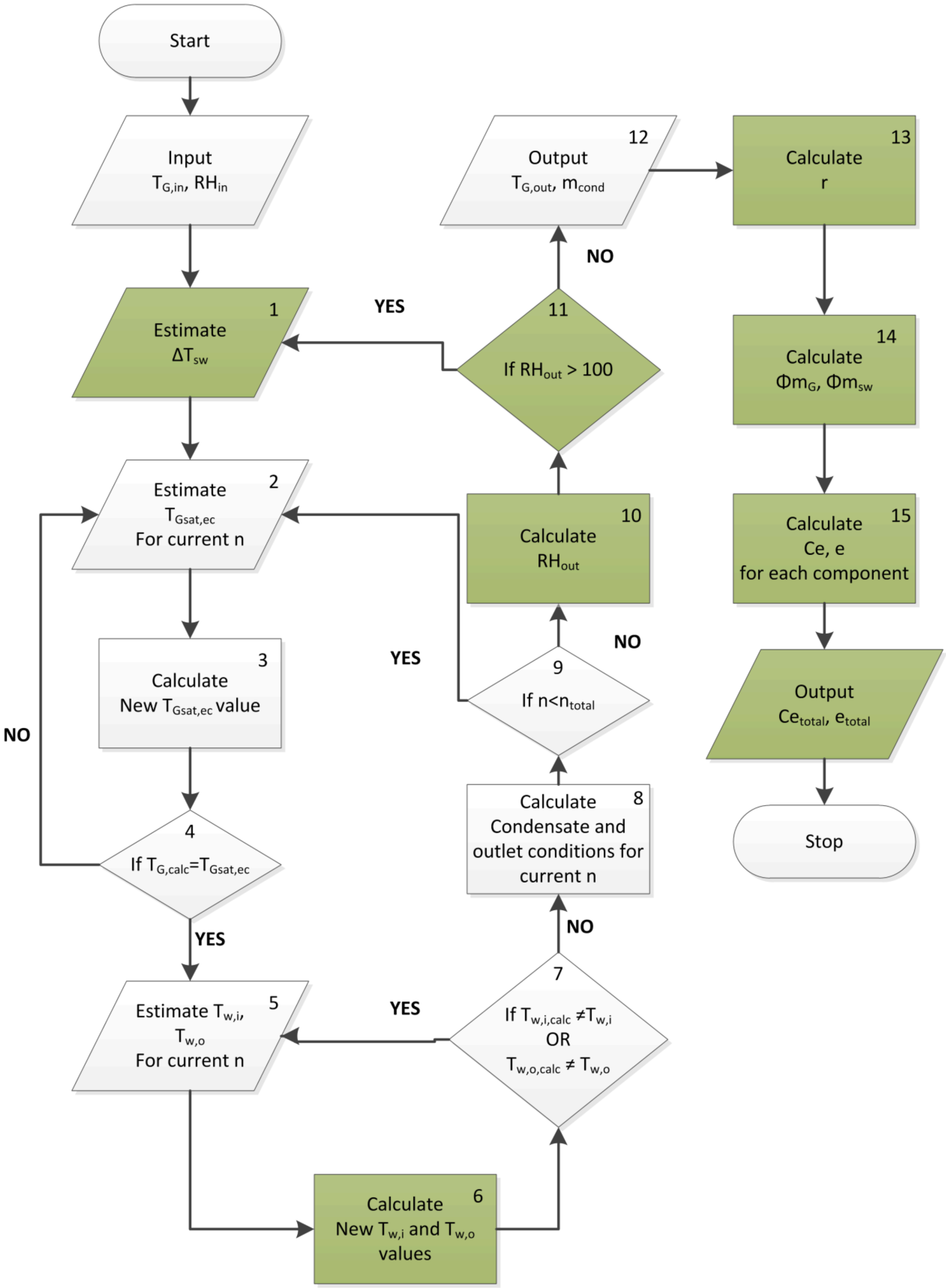

FIGURE 17: FLOW DIAGRAM TUBE CONDENSER COMPUTATIONAL MODEL

Validation of the tube condenser computational model

The University of Wageningen provided measurement data from experiments with the tube condenser in a seawater greenhouse in Oman (20). This data was used for validation of the tube condenser computational model including the Tagami relation for NCG's and calculations for relative humidity and seawater temperature per tube. In this validation values for heated air and surface seawater are used in order to accurately compare it to available data.

Since it is a steady state model, measurements at noon are chosen as a reference, since this the time at which maximum water production occurs. The seawater temperature, relative humidity and related water production are presented in the data from the University of Wageningen. However, the velocity of the seawater is not specified. The values from the dataset are therefore used in the tube condenser computational model to a point where the same production rate is achieved as in the given dataset. Table 5 summarizes the measured data from the seawater greenhouse and its fresh water production. Using this data in the tube condenser computational model with a seawater velocity of 0.07 m/s gave a production rate of 0.78 L/min, which is a 6 percent difference with the model results.

TABLE 6: MEASUREMENT DATA FROM SEAWATER GREENHOUSE IN OMAN

Air measurements	
Air velocity (m/s)	5
Inlet relative humidity (%)	89
Air inlet temperature (C)	29.25
Seawater measurements	
Seawater inlet temperature (C)	21.3
Seawater outlet temperature (C)	25.5
Seawater temperature increase per tube (C)	0.263
Production rate	
Production (L/min)	0.83

(36)

3.4.1 COMPACT PLASTIC HEAT EXCHANGER COMPUTATIONAL MODEL

Calculation approach auxiliary energy requirement
The air fan and seawater pump both have an auxiliary power requirement. A seawater pump is needed to pump the seawater to the top of the plastic heat exchanger and an air fan is required to blow the air through the tubes. The power requirement for the seawater pump is calculated using Equation 37. $\Delta P_{st,sw}$ indicates the static pressure drop which is required for the seawater to reach the top of the plastic heat exchanger. This pressure drop is calculated using Equation 38. In this case a frictionless pipe is assumed, since the frictional losses are small compared to the required potential energy increase.

$$e_{seawater\ pump} = \frac{\Phi v_{sw} * \Delta P_{st,sw}}{\eta_{seawater\,pump}} \tag{37}$$

$$\Delta P_{st,sw} = \rho_{sw} * g * H \tag{38}$$

This pressure drop on the air side is determined by analysing one of the tubes. Since the condensate is less than 1 percent of the air flow, a single phase flow is considered. The pressure drop in this tube consists of a static pressure drop, needed to push the air upwards, and a frictional pressure drop due to the contact with the plastic walls.

The data presented by Heat Matrix B.V. did not include information on the diameter of the tubes. This value is found by first determining the total cross sectional area of the compact plastic heat exchanger and then, based on the volume flow rate and velocity of the air and seawater, the fraction of this area which is used for each of these flows. It is assumed that the number of air and seawater tubes are similar, which makes it possible to calculate the diameter for each of the tubes. The obtained diameter for the tubes through which the air flows is 8mm, which is comparable to values found in literature for other plastic heat tube heat exchangers. Based on the tube diameter and the velocity of the air, the Reynolds number is calculated. For the first data set, the Reynolds number showed that the flow is turbulent and for the second data set the flow is laminar. This is due to the difference in velocity; the air velocity in the first data set is 9 m/s and in the second data set 1.1 m/s. For the turbulent flow the Swamee-Jain equation is used to find the friction factor (Equation 39) and for the laminar flow Equation 40 is used.

$$f_{turbulent} = \frac{0.25}{\left(\mathrm{LOG}\left(\frac{eabs}{3.7*d}+\frac{5.74}{Re^{0.9}}\right)\right)^2} \tag{39}$$

$$f_{laminar} = \frac{16}{Re} \tag{40}$$

The calculated diameter and friction factor are used in Equation 41 to find the frictional pressure drop.

$$\Delta P_{f,G} = \rho_G * 2 * f * \frac{H}{D_{o,G}} * v_{sw}{}^2 \tag{41}$$

Equation 42 is used to find the static pressure drop. The total power requirement for the air fan is then found by adding the frictional pressure drop with the static pressure drop and taking a fan efficiency of 80% into account.

$$\Delta P_{st,G} = \rho_G * g * H \tag{42}$$

$$e_{air\,fan} = \frac{\Phi v_G * (\Delta P_{f,G} + \Delta P_{st,G})}{\eta_{air\,fan}} \tag{43}$$

3.3 Performance of ocean thermal dehumidifiers in a tropical climate

In this section the results of the computational analysis are shown. In addition, the improvement potential of each dehumidifier is discussed.

3.6.1 Auxiliary energy requirement

Figure 18 shows the auxiliary energy requirement for the three dehumidifiers. "DCD" indicates the direct contact dehumidifier, "Tube" the tube condenser and "Compact" the compact plastic heat exchanger. The energy requirement is indicated as kWh per cubic meter of fresh water produced. In the remainder of the report this will be indicated by kWh/m^3. Three of the chosen system configurations (tube-basic, tube-tubes and compact-basic) have energy requirements higher than 10 kWh/m^3 and are therefore not shown in this figure.

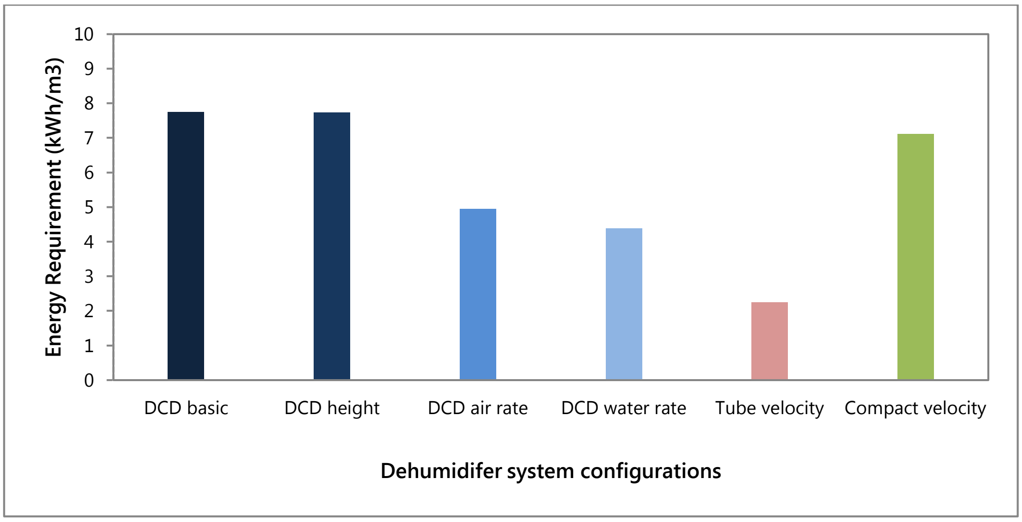

Figure 18: Energy requirements for selected dehumidifier configurations

Improvement potential of the energy requirement: direct contact dehumidifier

Figure 18 shows that the DCD air rate and DCD water rate configurations have a lower energy requirement than the basic configuration. In the DCD air rate configuration a higher air mass flux is used and in the DCD water rate configuration a lower fresh water mass flux. Optimization of air and water rates and their ratio could therefore possibly lead to further reduction in energy requirements. The majority (59%) of the energy loss in the DCD air rate configuration is due to the pressure drop in the external heat exchanger. The heat exchanger design used in these configurations is not optimized for current operating conditions. Since the heat exchanger has the largest energy requirement and considering that this part of the design is not yet optimized, it seems that there is significant potential for further reduction in the final energy requirement of the direct contact dehumidifier.

Improvement potential of the energy requirement: tube condenser

Reducing the air velocity from 5 m/s to 1 m/s reduces the energy requirement of the tube condenser significantly (from 78 to 2.24 kWh/m^3). The size of the system also reduces which indicates that more water is being produced per tube at lower air velocities. At a lower air velocity, the increase in seawater temperature per tube also decreases. Therefore, the temperature of the outlet seawater is sufficiently cold which could lead to possibly further cooling of the outlet air. Simultaneously decreasing the air velocity and increasing the number of tubes therefore shows potential for further energy reduction.

Improvement potential of the energy requirement: compact plastic heat exchanger
Decreasing the air velocity in the plastic heat exchanger also leads to lower energy requirements. However, the energy requirement is still above 6 kWh/m^3. The small tube diameter leads to more friction and hence energy loss. Increasing the tube diameter would decrease the specific heat transfer area and decrease heat transfer. Reduction of the air velocity also leads to less production per tube bundle and therefore larger system requirements and increased capital cost. Further reduction of the air velocity to decrease the energy requirements therefore does not seem feasible.

3.6.2 PURCHASED COMPONENT COSTS

Figure 19 shows the total purchased component costs for the different dehumidifier configurations.

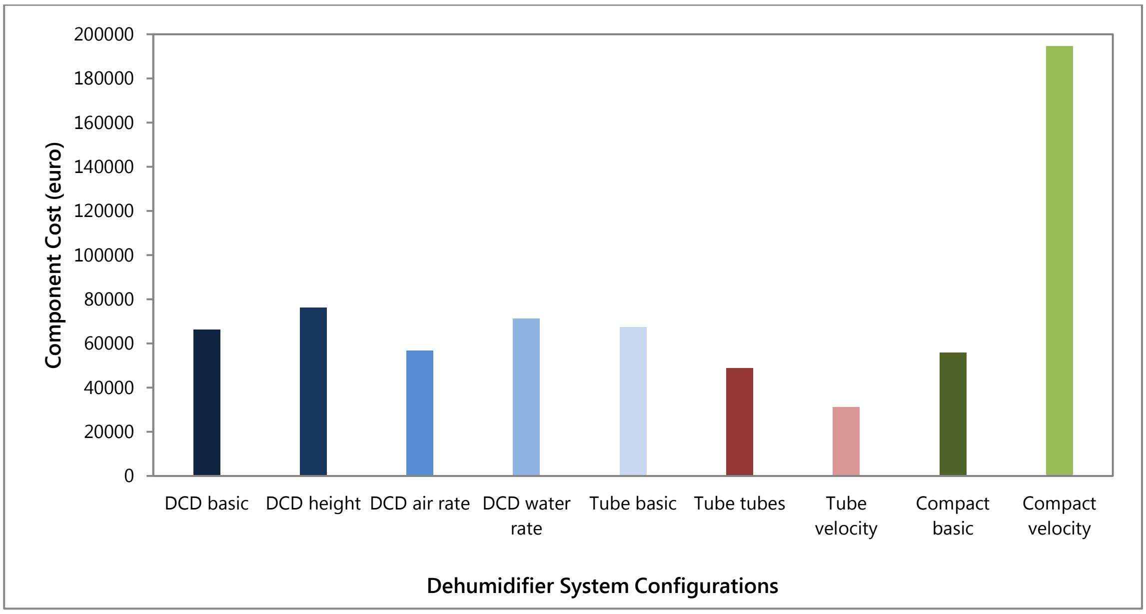

FIGURE 19: ENERGY REQUIREMENTS FOR SELECTED DEHUMIDIFIER CONFIGURATIONS

Improvement potential of the component cost: direct contact dehumidifier
The component cost for the different direct contact system configurations vary from 56,000 euro to 76,000 euro. Increasing the air rate not only gave lower energy requirements compared to the basic scenario, but also a decrease in component cost. The main cost component in the design is the tower (more than 60 percent of the total cost). The cost estimations are based on the use of a commercial tower used in chemical processing. These towers are designed for high pressures, however, the direct contact dehumidifier operates close to atmospheric pressure and therefore the material requirements are less stringent. In addition, since the diameter is larger than the height of the column, cost could possibly be reduced by making the system modular.

Improvement potential of the component cost: tube condenser
The main cost component in the tube condenser are the tubes. Low cost tube material is already used in the design, however, the large amount of tubes required (almost 6500) makes that the cost are high. Reducing the component cost therefore requires an overall more efficient design.

Improvement potential of the component cost: compact plastic heat exchanger
The component cost of the compact plastic heat exchanger increase when the air velocity is decreased from 9 m/s to 1.1 m/s. The production rate per tube bundle is lower and therefore more tube bundles are required to obtain the required amount of fresh water. Increasing the length of the tubes could possibly lead to more production, but this would lead to increased friction and therefore a higher energy requirement.

3.6.1 Intermediate conclusion

Since the performance of the compact plastic heat exchanger is already below a commercial reverse osmosis system and improvement potential seems limited, further analysis of a suitable dehumidification system will solely focus on the direct contact dehumidifier and the tube condenser.

3.6.3 Air Temperature at dehumidifier outlet

The outlet air temperature at the direct contact dehumidifier is lower than the tube condenser at all tested configurations. The air temperature is determined by the amount of heat transferred and therefore strongly depends on the operating conditions and the sizing of the dehumidifier. A high outlet air temperature also indicates that further cooling could be beneficial. Lowering the air velocity and increasing the number of tubes in the tube condenser both decrease the exit air temperature. Simultaneously changing these variables could therefore possible lead to further cooling of the air and improved heat transfer.

3.6.4 Resource efficiency

From Figure 21A it can be seen that the direct contact dehumidifier is more efficient with its resources; for the same production, less water and air are used. This is reflected in the outlet temperatures of the system, shown in Figure 21B. The outlet air temperature of the direct contact dehumidifier is lower and the outlet seawater temperature is higher since more heat has transferred. The heat transfer coefficients for both systems are similar, since air dominates the total heat transfer coefficient, but the direct contact dehumidifier has a higher specific heat area (267 m^2/m^3) compared to the tube condenser (55 m^2/m^3). To produce more water from the seawater and air, heat transfer should be improved. This can be done by increasing the heat transfer area or the heat transfer coefficient. Both of these parameters cannot be increased significantly for both the direct contact dehumidifier and the tube condenser without increasing the energy and component costs. The condensation efficiency of the tube condenser (18.2 %) is much lower than the direct contact dehumidifier (44%). Increasing the number of tubes with low air velocity could possibly increase this efficiency.

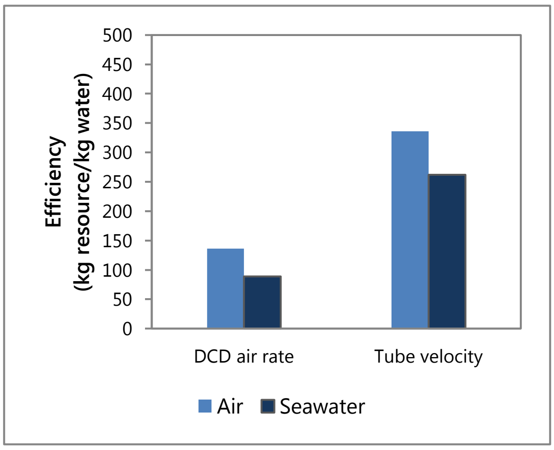

Figure 21A: Air and seawater efficiency for selected dehumidifier configurations

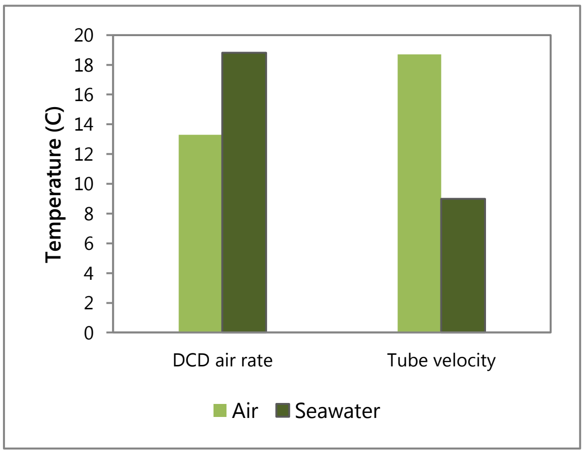

Figure 21B: Outlet temperatures of air and seawater for selected dehumidifier configurations

3.6.5 System size

Both the direct contact dehumidifier and tube condenser are significantly larger than a reverse osmosis system. The tube condenser is twice as large as the direct contact dehumidifier. The specific heat transfer area is significantly lower and hence the total volume of the system needs to be larger. The direct contact dehumidifier does not show much potential for a decrease in size since the specific heat transfer area is already large. Optimizing the design parameters could possibly increase the efficiency of the system which will reduce size, but this effect will be minimal. The same holds for the tube condenser; increasing the specific heat transfer area seems difficult since the decreasing the tube diameter will increase friction and increasing the efficiency of the system will only have a minor effect on the size.

3.4 Conclusion

The performance results of the ocean thermal dehumidifiers are promising. Only a limited amount of system configurations were tested, but energy requirements of less than 6 kWh/m^3 were achieved, which is comparable to commercial desalination technologies. The component cost is relatively high, but optimization of the operating conditions and component sizes of the direct contact dehumidifier and tube condenser could possibly reduce this cost. The performance of the compact plastic heat exchanger is less feasible due to the high cost of the tube bundle. An increase in air velocity reduces this cost, but increases the energy requirement significantly. It was therefore decided to omit the compact plastic heat exchanger from the analysis.

4. SELECTION OF DEHUMIDIFICATION METHOD

The computational analysis presented in chapter 3 showed the performance of selected system configurations of the ocean thermal dehumidifiers. In this chapter the direct contact dehumidifier and tube condenser are compared in a multi-criteria analysis to determine which dehumidifier is more suitable in an Ocean Ecopark. This analysis is done based on the expected performance of the dehumidifier, which means that both the performance results of the selected configurations and the improvement potential are taken into account. Section 4.1 describes the criteria and rating method used in the multi-criteria analysis .Section 4.2 presents the results of the analysis and in section 4.3 the most suitable dehumidification method is chosen.

4.1 Description and range of criteria in multi-criteria analysis

The three dehumidifiers are compared on different criteria. Table 7 gives an overview of the these criteria and the values used to rate the systems performance. This section will shortly describe the rating scale for each criteria.

Table 7: Criteria and range for multi-criteria analysis

Criteria	Range		
	1	**1.5**	**2**
Energy (kWh/m3)	> 6	2 - 6	< 2
Component Cost (euro)	> 37,200	24,800 - 37,200	< 24,800
Air temperature (C)	> 15 C	10 C - 15 C	< 10 C
Efficiency	< 0,33	0,33 - 0,66	> 0,66
Size	> 0.37 m3	0.37 - 0.55 m3	< 0.55 m3
Maintenance and Cleaning (hr/day)	> 1	0.5 -1.0	< 0.5
Reliability	low	medium	high

4.1.1 Description of criteria

Auxiliary Energy requirement

The energy requirement of the dehumidifier is indicated by the total kWh required per cubic meter of water produced. This does not include the energy required to pump the seawater from the deep ocean to the surface, but it does include pumping energy required for the seawater to move through the system. The energy requirement for desalination technologies vary with size, location and availability of resources. General guidelines indicate an energy requirement of 4 to 8 kWh/m^3 for reverse osmosis systems (6). For thermal desalination technologies this is 2 to 6 kWh/m^3 (27). The aim is to design a system which is competitive with desalination technologies and therefore a rating of 2 is given to dehumidifiers which have an energy requirement less than 2 kWh/ m^3 and 1 for energy requirements more than 6. Dehumidifiers with an energy requirement between 2 and 6 are given a rating of 1.5.

Purchased component cost

The component costs for the dehumidifiers are derived from data available for commercial components. Component costs are not easily comparable and strongly depend on the size and type of feed sources of the system. Cost data is available at Bluerise B.V. on a similar sized reverse osmosis system which uses ocean water as its feed source. This data is therefore taken as a guideline to rate the component costs of the different dehumidifiers.

Air temperature

The cold exit air temperature could provide additional revenue when used for air conditioning or ventilation. Since the additional revenue strongly depends on the cooling application, it is assumed that air colder than 15 degrees can be used for ventilation and air colder than 10 degrees for air conditioning. Air conditioning will likely provide more revenue and therefore dehumidifiers which produce air with a temperature less than 10 degrees receive a higher performance rating.

Resource efficiency
There are various methods to indicate the resource efficiency of the dehumidifier: the seawater efficiency (how much seawater is needed for production), the air efficiency (how much air is needed for production) and the condensation efficiency (how much water vapour is removed from the air). Air is readily available, but the seawater will have to be obtained from the deep ocean or from one of the ocean thermal technologies in the Ocean Ecopark which will induce additional pumping costs. The use of seawater should therefore be minimized. The recovery rate (unit of fresh water produce per unit of seawater used) for reverse osmosis systems is 50 to 80 percent, depending on the type of reverse osmosis system used (28). These values are used as a guideline to determine the seawater efficiency of ocean thermal dehumidification.

Size
The size of the dehumidifier is also a performance indicator; a small system will require less space which will be financially beneficial and advantageous when other ocean thermal technologies are also integrated in the Ocean Ecopark. The size of a reverse osmosis system with a similar fresh water production rate is used as a guideline.

Maintenance
The performance of the dehumidifiers on the previous criteria could be determined from the computational analysis presented in chapter 3. The maintenance and reliability of the dehumidifier are more difficult to predict. However, they could have a significant effect on the technical and economic performance of the systems and should therefore be taken into account. An estimation of the required maintenance will be made based on a comparison with a commercial reverse osmosis system.

In the direct contact dehumidifier seawater only flows through the external heat exchanger, which is specifically designed for seawater use. The packed bed will need some maintenance due to the use of atmospheric air in which contaminants might be present, however, since fresh water is used this is still low. Maintenance and cleaning could be further reduced by providing an air filter at the inlet of the condensation tower. In the tube condenser seawater is directly used in the system which makes than maintenance and cleaning will likely have to occur more often.

Reliability
The reliability will be indicated by assessing the reliability of the individual components and information of system reliability from literature. The direct contact dehumidifier solely consists of commercially available components which require little installation. Since these components are used and tested in a number of applications, the reliability of the system is high. The tube condenser requires more installation and data from the University of Wageningen showed that mechanical failure of the system was the main reason for ineffective performance. The reliability of both systems can be improved by constructing a prototype and performing mechanical tests.

4.1.2 RATING METHOD IN MULTI-CRITERIA ANALYSIS

The multi-criteria analysis used a rating scale for the performance, improvement potential and significance of the criteria. A rating of 1, 1.5 and 2 was given for each of these aspects. The total rating for each criteria was then determined by multiplying these values. To determine the effect of this rating on the outcome of the analysis, a sensitivity analysis is performed. Appendix A6 shows the results when the rating scale is changed and when the values are added instead of multiplied. In both cases the direct contact dehumidifier still performs better than the tube condenser.

4.2 Multi-criteria Analysis

Table 8 and 9 show the results from the multi-criteria analysis. The analysis consists of 7 criteria which each have received a value for importance (ranging from 1 to 5). The performance of each system is benchmarked against values for desalination technologies, and rated by 1, 1.5 or 2. The improvement potential is also rated by a value of 1, 1.5 or 2 and is based on the analysis as described in chapter 3. Multiplying these three components gives the total value for each criteria.

Table 8: Multi-criteria analysis of direct contact dehumidifier

Direct Contact Dehumidifier				
Criteria	**Importance**	**Performance**	**Improvement Potential**	**Total**
Energy	5	1,5	2	17,5
Component Cost	5	1	2	10
Air temperature	4	1,5	1,5	9
Efficiency	3	1,5	1	4,5
Size	3	1	1	3
Maintenance and Cleaning	3	2	1	6
Reliability	4	2	1,5	12
			TOTAL	**59,5**

Table 9: Multi-criteria analysis of tube condenser

Tube Condenser				
Criteria	**Importance**	**Performance**	**Improvement Potential**	**Total**
Energy	5	1,5	2	15
Component Cost	5	1,5	1	7,5
Air temperature	4	1	2	8
Efficiency	3	1	1,5	4,5
Size	3	1	1	3
Maintenance and Cleaning	3	1	1	3
Reliability	4	1	1,5	6
			TOTAL	**47**

4.3 Selection of dehumidifier for integration in an Ocean Ecopark

The multi-criteria analysis indicated that the direct contact dehumidifier shows more potential for integration in an Ocean Ecopark compared to the indirect contact dehumidifiers. Both the direct contact dehumidifier and tube condenser show significant potential for improvement, especially for further reduction in energy requirement. The direct contact dehumidifier, however, shows more potential for cost reduction when actually constructing the system. The reliability of the system is high since commercial components are used and seawater only flows through the external titanium heat exchanger (with a 30 year lifetime) which makes that maintenance and cleaning costs are low. In addition, the column costs are likely to be overestimated and further reduction in component costs is possible.

The operating conditions and component sizes of the dehumidifiers used in the multi-criteria analysis were not fully optimized and only a limited number of system configurations were tested. Minor adjustments to the operating conditions and component sizes lead to a significant impact on the performance of the dehumidifier. The simulations with different configurations not only made it possible to make an estimation for the improvement potential for each dehumidifier, but it also gave insight in the thermodynamic processes taking place in the dehumidifiers. This makes it possible to further support the choice for a direct contact dehumidifier. During the simulations it became clear that the heat transfer coefficient of both systems is similar. The resistance to heat transfer is largely determined by the high amount of non-condensable gases in atmospheric air. The effect of the heat transfer medium on the total heat transfer resistance is small. It should therefore be possible to achieve a similar heat extraction rate for both humidifiers if the heat transfer area and mass flow rates are equal. To test this hypothesis an additional simulation was run in which the air velocity in both dehumidifiers is set at 1 m/s and the number of tubes is increased such that it matches the total heat exchange area of the direct contact dehumidifier. In these configurations both dehumidifiers achieved a similar condensation efficiency (amount of water vapour removed from the air) and outlet air temperature. The main difference between the direct contact dehumidifier and the tube condenser is then the specific heat transfer area. The use of a packed bed makes that the specific heat transfer area is approximately five times as large for the direct contact dehumidifier as compared to the tube condenser. The system size is therefore significantly smaller. The direct contact dehumidifier is therefore selected for further investigation. The system shows potential for low energy and component cost, the construction is reliable, maintenance cost are low and the high specific heat transfer area of the packed bed shows potential for a compact system design.

Table 10: Performance of direct contact dehumidifier and tube condenser with comparable air velocity and heat exchange area

	Direct Contact Dehumidifier	Tube Condenser
Average heat transfer coefficient (W/m2*K)	23,4	15
Outlet air temperature (C)	13,3	12,6
Air efficiency (kg air/kg fresh water)	127	134
Size (m3)	24	40
Specific area (m2/m3)	267	55
Condensation efficiency (%)	44	46

5. Technical feasibility of a direct contact dehumidifier in Curacao

Chapter four showed that direct contact dehumidification is more feasible for application in an Ocean Ecopark compared to an indirect contact system. The system configurations which were presented in chapter three showed promising results for energy requirement, but also had high component costs. The aim is therefore to further optimize the system configuration to such an extent that a feasible energy requirement is achieved as well as a low component cost. The computational analysis also showed that the operating conditions have a significant impact on the performance of the dehumidifier. For example, when the water mass flux is doubled, a decrease in auxiliary energy requirement of almost 50 percent is achieved. This chapter will therefore further investigate the effect of the operating conditions on the auxiliary energy requirement and component cost of the dehumidifier.

The computational analysis in chapter three also indicated that optimization of the external heat exchanger could improve the performance of the dehumidifier. The operating conditions in the packed bed tower do not only influence the dehumidification process, but also the required sizing of the heat exchanger. It is therefore not possible to optimize the operating conditions of the tower and the heat exchanger separately. Section 5.1 describes the adjustments to the direct contact computational model which were required to further investigate the effect of the operating conditions on the performance of the dehumidifier. The optimal operating conditions of the dehumidifier also depend on local conditions of the project site. For example, higher air temperatures support the heat transfer in the tower and high humidity ratios increase production. This feasibility study will therefore take the shape of a case study in which these external conditions are specified. Section 5.2 describes these conditions. Section 5.3 describes the different operating conditions which are explored. The results of these simulations are shown in section 5.4. Based on specified targets for energy requirement and component costs, the technical feasibility of a direct contact dehumidifier in Curacao is determined.

5.1 EXTENSION OF COMPUTATIONAL ANALYSIS

Previous studies on direct contact dehumidification focus on the dehumidification process in the packed bed tower. The operating conditions of the tower influence the design of the heat exchanger and the heat exchanger in turn significantly impacts the energy requirement. This study will therefore also include the performance of the heat exchanger in determining the technical feasibility of the dehumidifier. To do so, the direct contact computational model is adjusted for more accurate modelling of the heat exchanger. The system configurations used in the multi-criteria analysis were based on the use of a commercial heat exchanger with a constant pressure drop. A higher inlet fresh water temperature or mass flow rate would lead to a higher seawater mass flow rate or an increase in flow channels. However, this method only makes it possible to investigate a limited amount of configurations since both the fresh water and seawater mass flow ratios and the log mean temperature difference needs to be comparable to the commercial design in order to assume a constant pressure drop. A code calculating the pressure drop of the heat exchanger is therefore developed as part of this research which makes it possible to investigate the effect of the fresh water mass flow rate and temperature on the performance of the dehumidifier. The development of this code is described in section 5.1.1.

The direct contact computational model is also based on the assumption that evaporative cooling in the packed bed tower is an adiabatic process. The temperature of the fresh water after condensation is equal to the temperature after evaporative cooling. The increase in fresh water temperature during evaporative cooling might not have a large impact on the humidification process in the tower, but it could impact the design of the heat exchanger. Evaporative cooling in a packed bed dehumidifier was researched by students from Delft University of Technology. They developed a code for evaporative cooling in a packed bed dehumidifier and integrated this code with the computational model developed as part of this research. The integrated energy and cost calculations were not part of the direct contact computational model when the students started their research. Section 5.1.2 will compare the results from the integrated model with the results obtained from the model with adiabatic evaporative cooling. Based on this comparison it will determined if the assumption of adiabatic evaporative cooling provides sufficiently accurate results.

The component costs are mainly affected by the cost of the tower. In the multi-criteria analysis it was mentioned that the tower cost could be overestimated since the design is unconventional (large diameter for low height) and included high pressure resistant materials. To investigate the effect of making the system modular, a different code was written for the cost calculation of the packed bed tower. This code is further described in section 5.1.3.

5.2.1 CALCULATION APPROACH FOR PRESSURE DROP IN HEAT EXCHANGER

In practice, heat exchanger design is an iterative process in which an optimum needs to be found for energy and material cost. Part of this iterative process is therefore added to the direct contact computational model such that the influence of the dehumidifier operating conditions on the required size and the pressure drop of the heat exchanger could be investigated. A plate heat exchanger is chosen for this analysis since it can reach high heat transfer rates which minimizes the required heat exchange area. The iterative design process described by Li et al (2006) is used to set up the calculations for the required heat exchange area and pressure drop of the heat exchanger. The length, width and thickness of the plates are typical values used in heat exchanger design and these values are kept constant for all simulations. First, the required heat extraction is calculated using a heat balance for the fresh water (Equation 44).

$$Q = \Phi M_{fw} * Cp_{fw} * (T_{fw,in} - T_{fw,out}) \quad (44)$$

The outlet seawater temperature is then calculated based on a heat exchanger effectiveness of 0.8 (Equation 45).

$$\mathrm{T}_{sw,out} = \mathrm{E} * \left(\mathrm{T}_{fw,in} - \mathrm{T}_{sw,in}\right) + \mathrm{T}_{sw,in} \tag{45}$$

This makes it possible to calculate the required mass flow rate of seawater and the log mean temperature difference. Heat transfer coefficients for plate heat exchangers can range from 3500 W/m2*K to 7500 W/m2*K and therefore an estimated heat transfer coefficient of 5000 W/m2*K is chosen to calculate the required heat transfer area in the first step of the iteration. Following, the required number of plates and the velocity through the channels is calculated which makes it possible to determine the overall heat transfer coefficient (Equation 46). This value is then compared to the first estimation and if the difference is larger than 100 W/m2*K, the number of plates is adjusted in the second step of the iteration.

$$\frac{1}{U} = \frac{1}{U_{fw}} + \frac{1}{U_{sw}} + \frac{t_p}{k_p} \tag{46}$$

When a suitable number of plates is found, and hence the total heat transfer area, the friction factors and the pressure drops are calculated (Equation 47).

$$\Delta P_{h,sw} = 8 * ff_{sw} * \rho_{sw} * \frac{H_p}{2*bh} * 0.5 * {v_{sw}}^2 \tag{47}$$

To validate this calculation method, the results are compared with a commercial heat exchanger. Table 12 shows the data of the commercial design and the results of the direct contact computational model for similar mass flow rates and temperatures. It can be seen that the spacing between the plates and the length of the plates has an influence on the performance of the system. For a spacing of 0.002 m and a plate length of 1.25 m, the results of the computational model and the commercial design are comparable.

TABLE 11: COMPARISON PERFORMANCE COMMERCIAL HEAT EXCHANGER AND RESULTS DIRECT CONTACT COMPUTATIONAL MODEL

	Specifications Commercial Heat Exchanger	Results DC-model	Results DC-model	Results DC-model
Input value DC-model	**-**			
Spacing		0.003	0.002	0.002
Plate length	-	1.5	1.5	1.25
Performance				
Outlet seawater temperature (C)	11.5	12	12	12
LTMD (K)	2.47	2.16	2.16	2.16
Heat transfer area (m^2)	46	78	46.8	55.9
Plates	99	105	56	81
Pressure drop fresh water (kPa)	49.5	18.6	103	50.8
Pressure drop seawater (kPa)	34.3	11.6	69	34.1
Heat transfer coefficient (W/m^2*K)	5000	3458	6754	5580

5.2.2 Effect of evaporative cooling on design parameters

Students from Delft University of Technology developed a computational model for the evaporative cooling in a packed bed dehumidifier. The calculation method in the direct contact computational model is based on adiabatic cooling and does not take heating of the fresh water into account. In this section the results of both models will be compared in order to determine if integration is feasible. To do so, a simulation is run for a fresh water mass flux of 0.8 kg/m2*s and air to water mass flow ratios of 0.625 and 1.25. The computational model developed by the students from Delft University of Technology will be referred to as DUT-model.

Height for saturation
Both models calculate the packed bed height needed to reach saturation. The difference in results was 2.5 cm for a ratio of 0.625 and 6 cm for a ratio of 1.25. The total packed bed height is 1 m and the difference therefore seems neglible.

Fresh water production
The calculated water production is lower for the direct contact model. The difference with the DUT model is 0.36 L/hr for the low ratio and 2.16 L/hr for the high ratio. This represents a 2 and 7 percent difference in total production.

Fresh water temperature
The difference in outlet water temperature is 2 degrees at the high water to air ratio. Even though this value seems small, on a log mean temperature difference of about 2.5 it could have a significant impact.

It is therefore decided to combine the direct contact computational model and the DUT-model. The difference in outlet fresh water temperature can significantly affect the sizing of the heat exchanger and hence the performance of the humidifier.

5.2.3 Modular packed bed column

The cost calculations performed as part of the multi-criteria analysis are based on a packed bed tower used in chemical processing. The cost includes installation and auxiliaries and the effect of the height to diameter ratio was not taken into account. This leads to an unconventional design with a high diameter to height ratio and auxiliaries which are not specific for the dehumidifier design. Therefore, the cost calculations for the packed bed column are reconsidered. The cost for the tower is first subdivided such that the tower, packed bed and auxiliaries are calculated separately. The cost for the tower is estimated based on Guthrie's modular method to preliminary design (29). Equation 48 shows the cost calculation for the column. In this equation, C_0 represents a cost parameter for a base configuration. H_0 and d_0 represent the base height and diameter of the tower, this is 4 feet and 3 feet respectively. The parameters a and b take economy of scale into account. The maximum diameter of the column is 10 ft and the maximum height 100 ft. The cost for the packed bed and the nozzles are calculated separately and scale linearly with the size of the system.

$$Ce_{column} = C_0 \left(\frac{H}{H_0}\right)^a \left(\frac{d}{d_0}\right)^b \quad (48)$$

5.2 Context of case study

Location of case study
Bluerise is currently in the process of developing OTEC and SWAC technologies in Curacao and it is investigating the potential of an Ocean Ecopark. The geographic location and climate have proven to be suitable for obtaining deep ocean water and the results of a case study in Curacao will be useful for Bluerise in developing an Ocean Ecopark.

Environmental conditions in Curacao
The air temperature and humidity values used in this case study originate from climate data provided by the Meteorological Department of Curacao (30). Average monthly temperatures and relative humidity are fairly constant throughout the year. The maximum difference between the average monthly temperature and the average yearly temperature is 1.6 degrees Celsius. Seasonal changes are therefore expected to have little effect on the performance of the direct contact dehumidifier. The hourly changes in air temperature and humidity are more significant. For example, measurements from a day in July indicate a temperature high of 31 degrees Celsius (at noon) and a low of 27 degrees Celsius (at 4 AM). The relative humidity for this days varies from 84 percent (4 AM) and 62 percent (1 PM). The hourly difference in temperature and humidity could have an effect on the performance of the system; the lower air temperature decreases heat transfer, but the higher relative humidity requires less evaporative cooling for saturation. In this feasibility study only a steady state analysis will be performed based on yearly average values.

Required fresh water production
When the produced fresh water is used for human consumption (drinking water) purification is required. Finding a suitable purification method falls out of the scope of this research and therefore this case study will focus on water production for irrigation. The direct contact dehumidifier could specifically be beneficial for a greenhouse, since it produces both water and cool air. There are different sizes of greenhouses possible, but the sizing of the seawater greenhouse in Oman (20) will be used as a guideline to determine the required water production. Appendix A7 shows the calculations for the required water production based on the sizing of a seawater greenhouse.

Required cold air production
The air leaving the direct contact dehumidifier could be used for cooling of the greenhouse. However, it is not possible to run simulations with a requirement for fresh water production and outlet air temperature. The cold air produced by the dehumidifier will therefore be considered separately and its economic value will be determined from comparing it with cost values for cooling.

Table 12: Boundary conditions for case study in Curacao

Case study	
Water production (L/d)	*4838*
Air temperature (C)	*28*
Relative humidity (%)	*76*

5.3 Explored range of operating conditions

Research data from literature showed that the packed bed height, air mass flux and fresh water mass flux could have a significant impact on the performance of the direct contact dehumidifier (14). The analysis made for the multi-criteria analysis showed that the air and water mass fluxes in specific had a large influence on the performance of the system when using atmospheric air and deep seawater and that optimizing these operating conditions could lead to lower auxiliary energy requirements and component costs compared to the performance results shown in the multi-criteria analysis. In addition, the analysis showed that the heat exchanger is a critical design component. The pressure drop and its component costs have a significant impact on the total performance of the system. The design of the heat exchanger is mainly determined by the amount of heat which needs to be extracted from the fresh water. The required outlet temperature at the heat exchanger, which is the inlet fresh water temperature at the tower, significantly impacts the required heat extraction. To investigate the effect of this temperature on the performance of the system, this value is also taken into account in the different system configurations.

Table 13 shows an overview of the explored ranges for each variable. These operating conditions are simulated and tested for energy requirement and component cost. The goal of this analysis is not to find an optimized design, but to test if a direct contact dehumidifier could be feasible in an Ocean Ecopark. Therefore, not a full optimization analysis is performed, in which an extensive amount of variables would be tested, but only the indicated range. The dehumidifier is tested for feasibility by setting a target for energy requirement and component cost. As mentioned in previous sections, the energy requirement for desalination technologies is between 2 and 6 kWh/m^3. Since the aim for this system is to be competitive with desalination technologies, the energy requirement needs to be at least 6 kWh/ m^3 and preferably at the lower end of this range. Indicating a general value for component cost is more complex since this can vary significantly per location and sizing. Therefore, the component cost for a reverse osmosis system specifically designed for an Ocean Ecopark is taken as a guideline. Since the direct contact dehumidifier system will likely generate additional revenue from cold air and does not have costs for brine disposal, the component cost can be slightly higher. Adding a 20% increase in cost, will give a maximum component cost of 25,500 euro.

Table 13: Explored operating conditions of direct contact dehumidifier in technical feasibility study

Operating condition	Explored range
Fresh Water Mass Flux (kg/m2*s)	0.5 – 1.5
Air to fresh water ratio	0.4 – 1.4
Fresh water temperature at tower inlet	8 – 12

5.4 Performance of a direct contact dehumidifier in Curacao

5.4.1 Auxiliary energy requirement

Figure 22 shows the auxiliary energy requirement of the direct contact dehumidifier with a fresh water flux of 0.5 kg/m2*s. The data points indicated by triangles represent the configurations with a fresh water temperature of 12 degrees Celsius at the inlet of the tower, the square data points a temperature of 10 degrees Celsius and the round data points a temperature of 8 degrees Celsius. The data points show that a high fresh water temperature leads to a high energy requirement. This difference decreases at high air to fresh water ratios. Also, a high air to fresh water ratio is more favourable for low energy requirements. Figure 23 shows the energy requirement of the direct contact dehumidifier with a fresh water flux of 1.5 kg/m2*s. In this case a higher air to fresh water ratio and a low fresh water temperature also leads to lower energy requirements. When comparing the two figures, it can be seen that a lower fresh water flux leads to lower energy requirements for this configuration.

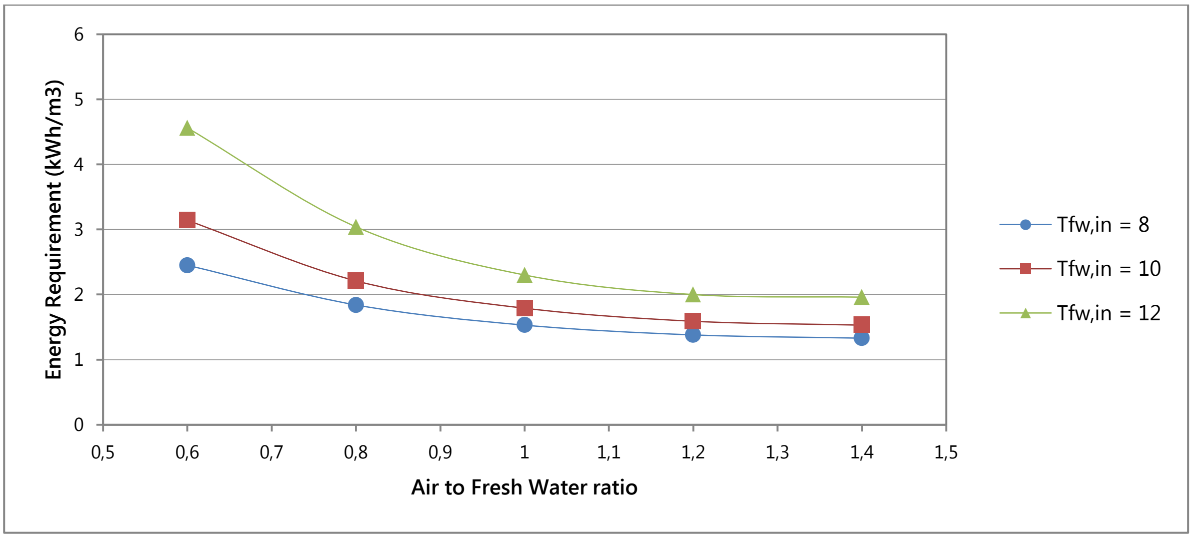

*Figure 22: Auxiliary energy requirement of direct contact dehumidifier with a fresh water flux of 0.5 kg/m2*s and a production rate of 4838 litres per day*

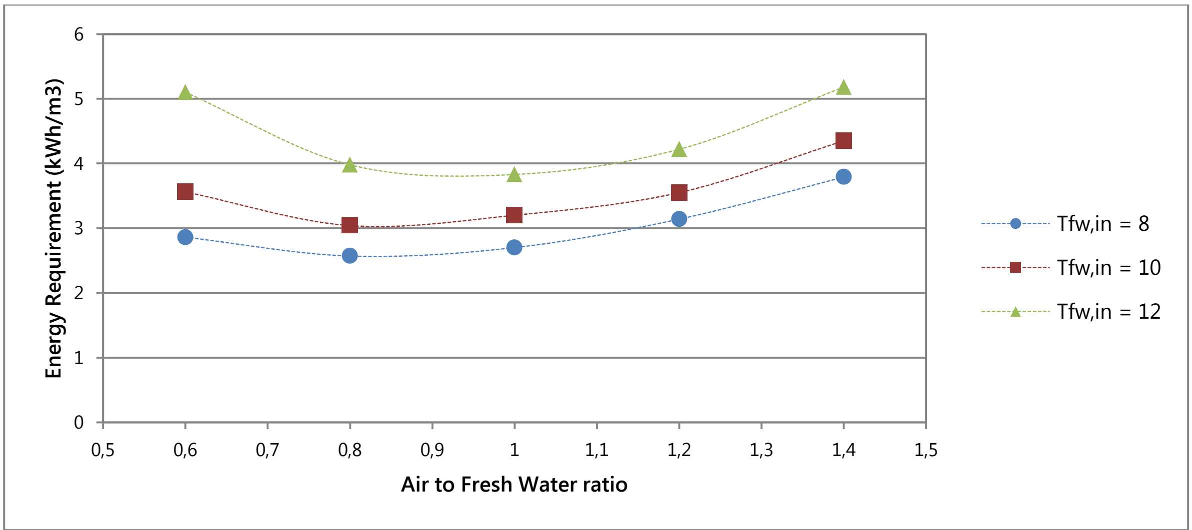

*Figure 23: Auxiliary energy requirement of direct contact dehumidifier with a fresh water flux of 1.5 kg/m2*s and a production rate of 4838 litres per day*

For the configurations with a fresh water flux of 1.5 kg/m2*s energy usage is minimized at an air to fresh water ratio near 0.8 -1 kg/m2*s. At first the additional air available in the tower, and hence water vapour, will lead to increased production and therefore the total required production can be obtained with a smaller sized system. Since the fresh water mass flux is constant, a smaller system also leads to a lower total fresh water mass flow rate and hence a smaller heat exchanger. However, increasing the ratio also increases the energy required for the air fan. At some point, the decrease in energy requirement due to the increase in production will not outweigh the additional energy requirement for the air fan. This is shown in Figure 25 where the energy requirement per component is shown for different air to fresh water ratios. This minimum is not yet reached for the configurations with an a fresh water mass flux of 0.5 kg/m2*s. However, Figure 25 shows that the energy requirement of the air fan is increasing at higher ratios. It can therefore be expected that the energy requirement at a ratio of 1.4 is already close to optimum. Also, the auxiliary power requirement of the air fan is lower at a flux of 0.5 than at a flux of 1.5; at a similar ratio, a higher water mass flux also leads to a higher air mass flux.

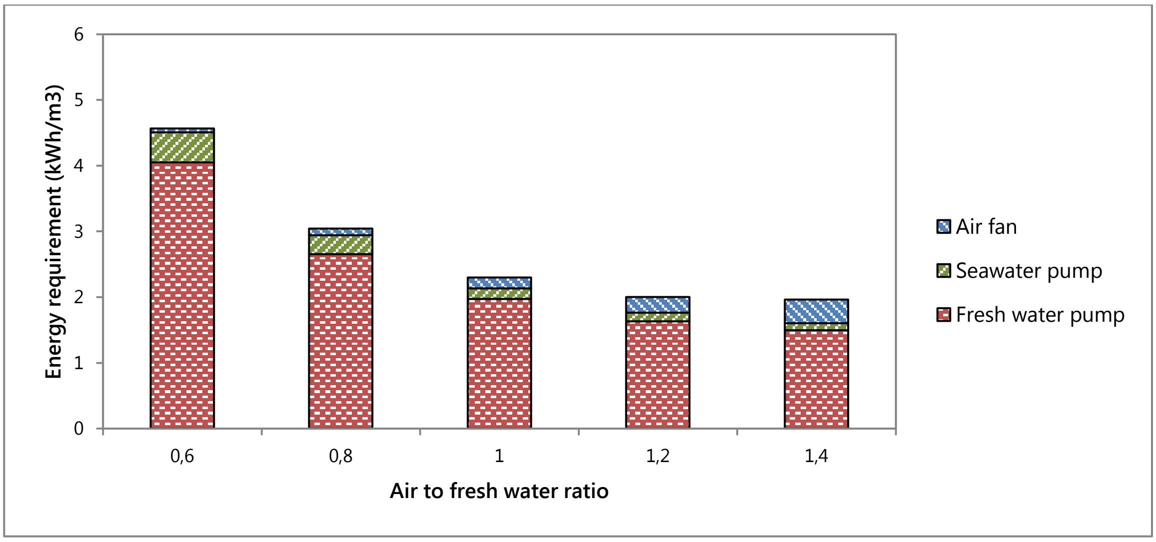

*Figure 24: Auxiliary energy requirement per component of the direct contact dehumidifier with a fresh water mass flux of 0.5 kg/m2*s and a fresh water tower inlet temperature of 12 degrees Celsius*

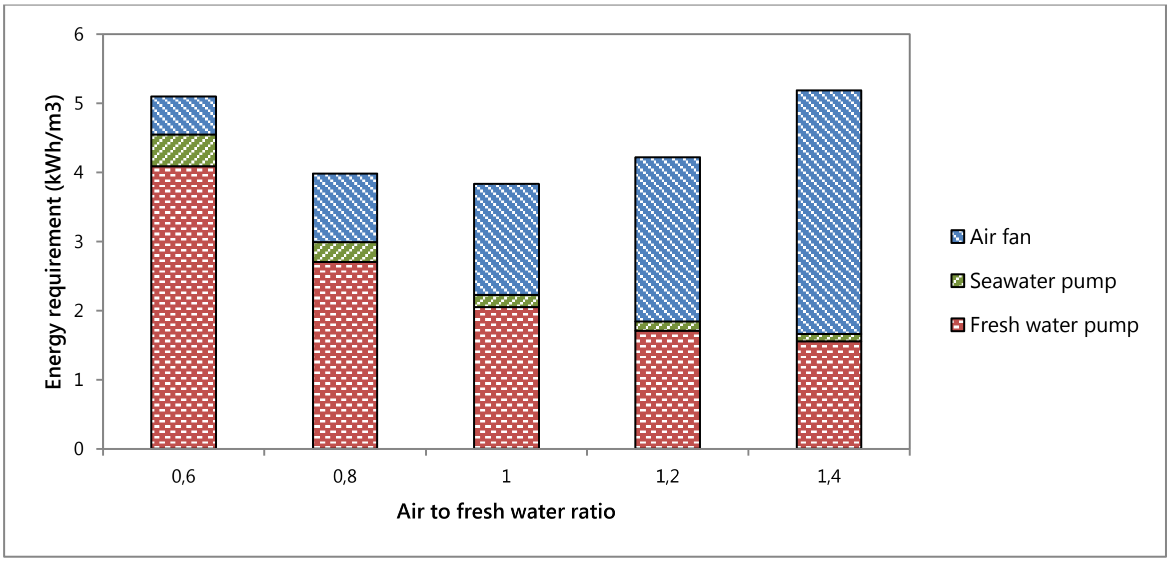

*Figure 25: Auxiliary energy requirement per component of the direct contact dehumidifier with a fresh water mass flux of 1.5 kg/m2*s and a fresh water tower inlet temperature of 12 degrees Celsius*

5.4.2 PURCHASED COMPONENT COST

Figure 26 shows the component costs for similar operating conditions as in the previous section. The component cost decrease at high air to water ratios and, contrary to the energy requirement, the costs are higher for low fresh water inlet temperatures. Looking at the individual component costs it can be seen that the increase in costs is due to the heat exchanger. Figure 27 shows the individual component costs at an air to water ratio of 1.4. The figure shows that the difference in costs between a low and high inlet temperature is made by the heat exchanger. A lower inlet fresh water temperature requires more cooling and hence a larger heat exchanger. The additional plates required for this heat exchange induce additional costs. At higher ratios the cost for the heat exchanger also increases since the fresh water heats up more when higher air mass fluxes are used. However, the cost for the tower will decrease at higher ratios, since more production is achieved per cross sectional area of the tower.

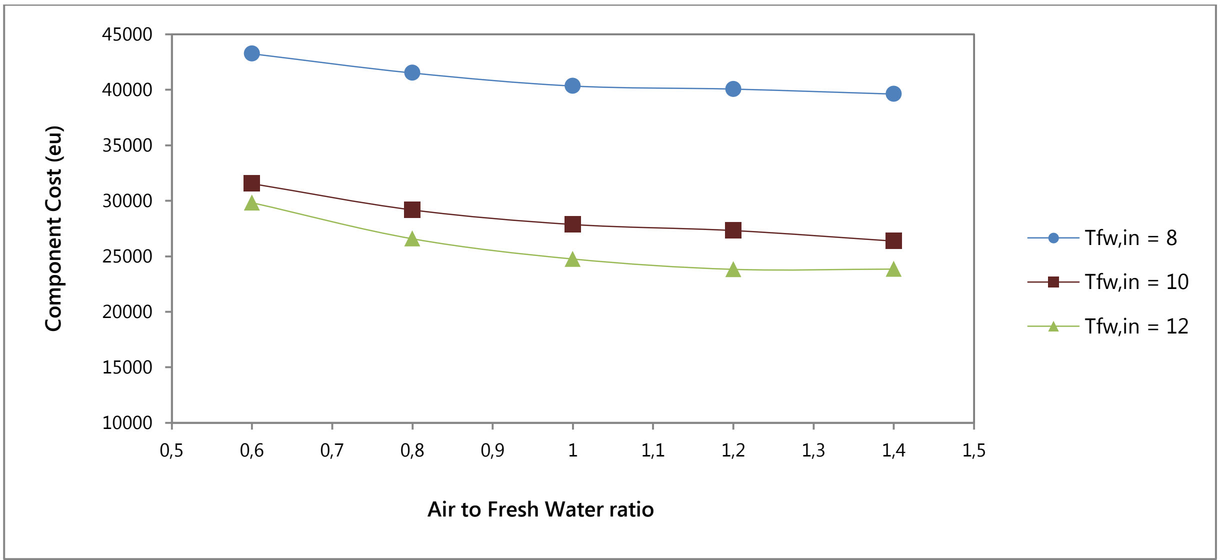

*FIGURE 26: TOTAL PURCHASED COMPONENT COST FOR A DIRECT CONTACT DEHUMIDIFIER WITH A FRESH WATER MASS FLUX OF 0.5 KG/M2*S*

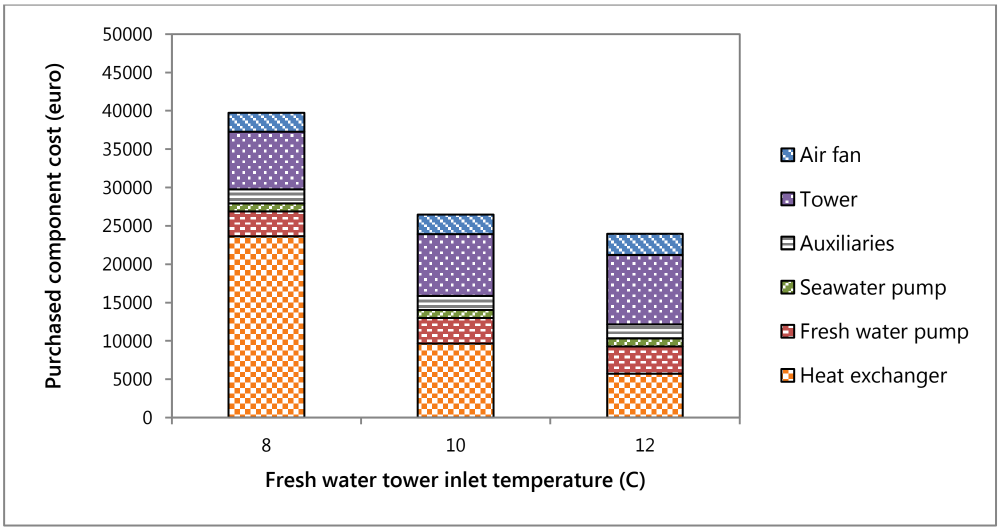

*FIGURE 27: PURCHASED COST PER COMPONENT FOR A DIRECT CONTACT DEHUMIDIFIER WITH AN AIR TO WATER RATIO OF 1.4 AND A FRESH WATER MASS FLUX OF 0.5 KG/M2*S*

5.4.3 Impact of heat exchanger design on performance

The simulations showed that for the selected ratios and inlet temperatures both the energy requirement and component costs decrease at higher air to fresh water ratios. However, there is an inverse relationship between energy and component costs when comparing fresh water temperatures at the inlet of the tower; a high temperature leads to low component costs but a high energy requirement and a low temperature leads to high component costs and a low energy requirement. Figure 26 showed that the difference in component costs between the configuration with an inlet temperature of 8 degrees Celsius and the configuration with 10 degrees Celsius is much smaller than comparing the configurations with 10 and 12 degrees Celsius. This difference is due to the design of the heat exchanger. The direct contact computational model includes an approach to find a suitable heat transfer coefficient and heat exchange area, but it assumes a constant plate size and spacing. At low fresh water temperatures it seems that the spacing is not optimized and leads to high component costs. At low temperatures more heat exchange is required in the heat exchanger. This can be achieved by increasing the heat exchange area, and hence the number of plates, or by increasing the heat transfer coefficient. The heat transfer coefficient is determined by the velocity of the fluid. Increasing this velocity increases turbulence and heat transfer, but also increases friction and hence the pressure drop. Spacing of the heat exchanger plates therefore needs to be taken into account when finding an optimal relation between energy requirement and component cost of the system.

Figure 28 shows the effect of the heat exchanger plate spacing on the energy requirement and component cost of the system for a fresh water flux of 0.5 kg/m2*s and a ratio of 1.4. The dotted lines indicate the component cost for different inlet temperatures while the solid lines indicates the energy requirement. The figure shows that the effect of heat exchanger plate spacing is especially large for a low fresh water temperature at the inlet of the tower; an increase in spacing from 2 to 3 mm increases costs by more than 10,000 euro, while the energy requirement stays roughly similar. When the spacing increases, the velocity of the fluid decreases and hence the heat transfer coefficient decreases. In this case, the calculated heat transfer coefficient drops by more than 50 percent. To make up for this decrease in heat transfer, more plates are required. For an increase in spacing of 1 mm, the required heat transfer area is more than doubled.

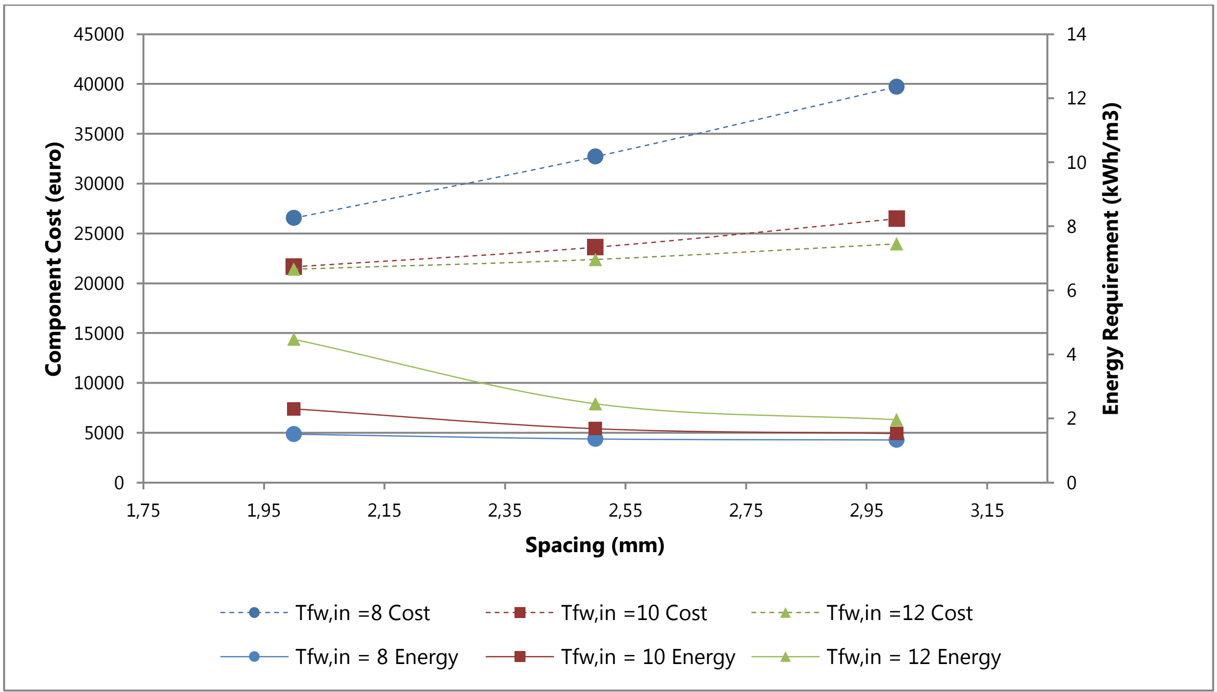

*Figure 28: Effect of heat exchanger plate spacing on performance of direct contact dehumidifier: component cost and energy requirement for a fresh water mass flux of 0.5 kg/m2*s and an air to water ratio of 1.4*

5.5 TECHNICAL FEASIBILITY OF A DIRECT CONTACT DEHUMIDIFIER IN CURACAO

The simulation results showed that energy requirements lower than 2 kWh/ m^3 could be achieved as well as component costs of less than 25,000 euro. However, the low energy requirements were achieved with a low fresh water mass flux and a low fresh water temperature, while the low component costs were achieved with a high fresh water mass flux and a high inlet temperature. When constructing a direct contact dehumidifier in an Ocean Ecopark, an optimum needs to be found between these two components. The heat exchanger is a critical design component in finding this optimum. It was shown that the plate spacing significantly impacts the performance of the direct contact dehumidifier and affects both energy requirement and component costs. Designing the heat exchanger for optimal energy and cost conditions is a complex and iterative process. In this feasibility study an iterative programming code was developed which finds a suitable heat transfer coefficient and heat exchange area, but it does not take into account the plate sizing and spacing of the heat exchanger. To optimize all variables and make a detailed design of the heat exchanger a commercial heat exchanger design program can be used. In this study it was chosen not to perform this extensive design process and solely focus on the feasibility of the direct contact dehumidifier by providing design guidelines and testing one of the suitable configurations.

5.5.1 DEFINITION OF TECHNICAL FEASIBILITY

In this case, feasible systems are indicated by energy requirements of at least 6 kWh/ m^3, in order to be competitive with desalination systems and a component cost of less than 25,500. This value is chosen based on the cost of a reverse osmosis system. The component cost of a reverse osmosis system of 5000 L/d is 21,278 euro. Since the direct contact dehumidifier will generate additional revenue from cold air and does not have costs for brine disposal, the component cost can be slightly higher. Adding a 20% increase in cost, will give a maximum component cost of 25,500 euro.

Figure 28 shows that all three system configurations are able to reach the energy target, but only with a fresh water temperature of 10 or 12 degrees Celsius the requirement for component costs is met. It can therefore be concluded that the direct contact dehumidifier is technically feasible in an Ocean Ecopark. The energy requirement and component costs for the system could possibly be further reduced when a commercial heat exchanger design program is used or when the explored scenarios are extended. However, this study only aims to indicate the feasibility of the system and therefore detailed optimization and design will not be included.

5.5.2 Feasible system configuration

Chapter 4 showed that there are several system configurations possible which can achieve an energy requirement below 6 kWh/m^3 and with a purchased component cost below 25,500 euro. Figure 28 indicated that with some optimization of the operating conditions and component sizes an auxiliary energy requirement of less than 2 kWh/m^3 can be achieved. The operating conditions used in this feasibility study were chosen since it had an energy requirement less than 2 kWh/ m^3, while also meeting the target for component costs. The energy requirement achieved with this configuration is 1.68 kWh/ m^3 and the component costs are 23,600 euro. Table 14 gives the operating conditions of this configuration and Table 15 the sizing of the components.

Table 14: Direct contact dehumidifier in Curacao: operating conditions

Fresh water	
Temperature at tower inlet (C)	10
Temperature at tower outlet (C)	24.7
mass flux (kg/m2*s)	0.5
mass flow rate (kg/s)	4.23
Air	
Inlet temperature (C)	28
Relative humidity (%)	76
mass flux (kg/m2*s)	0.7
mass flow rate (kg/s)	5.93
Temperature at tower outlet (C)	11.80
Seawater	
Mass flow rate (kg/s)	4.38
Temperature at heat exchanger outlet (C)	21.0
Fresh water	
Production rate (L/d)	4848

Table 15: Direct contact dehumidifier in Curacao: sizing of components

Heat exchanger	
Plate height (m)	1.5
Plate width (m)	0.5
Plate spacing (mm)	2.5
Number of plates	33
Tower	
Cross sectional area (m^2)	8.47
Tower height (m)	2.5
Packed bed height (m)	1
height for evaporative cooling (m)	0.19
height for condensation (m)	0.81

6. Economic analysis of a direct contact dehumidifier in Curacao

The previous chapter showed that the direct contact dehumidifier which is able to operate with an auxiliary energy requirement that is competitive with commercial desalination systems and a component cost lower than reverse osmosis system with a similar production rate. These results are promising; however, an economic analysis is necessary to determine whether a direct contact dehumidifier in an Ocean Ecopark can actually be financially competitive with other fresh water production technologies. The component costs are a good indicator for the fixed capital costs and the energy requirement for the operational costs but to accurately compare the financial viability of a direct contact dehumidifier with other fresh water production technologies a different indicator needs to be used which takes into account all of the cost factors. The price of water is one such indicator. Calculating the price of water not only makes it possible to compare the dehumidifier with current fresh water technologies in Curacao, but it also delivers some insightful economic indicators in the calculation process such as the required initial investment and operating expenses.

The economic analysis method used to determine the price of water is described in section 6.1. The present value method is chosen since it is one of the more accurate calculation methods available in literature. Section 6.2 presents the results of the economic analysis. The results are subdivided in fixed capital costs, operating costs and the price of water.

6.1 ECONOMIC ANALYSIS METHOD

6.1.1 ESTIMATION OF PRODUCTION COSTS

The production cost of the direct contact dehumidifier consists of fixed capital costs and operational costs. The method used to calculate these costs is described in Chemical Engineering Design Principles, Practice and Economics of Plant and Process Design (31). This method and the values used in the economic analysis for the direct contact dehumidifier are described in this section.

Fixed Capital costs

The fixed capital costs include the purchased component costs and the cost for design, construction and installation. Since the dehumidifier will be located in an Ocean Ecopark it is assumed that the costs for preparation of the plant site are neglible. The purchased component costs are calculated with the direct contact computational model. This simulation was already performed as part of the technical feasibility study and lead to a total purchased component cost of 23,600 euro. Detailed factorial estimates are used to calculate the installation, design and engineering costs for each component. This means that the purchased component costs are multiplied by a factor to obtain the total physical plant costs which includes installation, design and engineering. The factors used in these estimations are derived from Coulson and Richardson's Chemical Engineering book (32) and are shown in Table 16. A contingency charge of 10 percent is added to the total physical plant cost to account for variability in the project as well as a contractors fee of 5 percent (32).

TABLE 16: FACTORIAL ESTIMATES FOR INSTALLATION, DESIGN AND ENGINEERING

Factorial estimate	
Piping	0,7
Instrumentation	0,2
Electrical	0,1
Design and engineering	0,3

Operating Costs

The operating costs of the dehumidifier consist of the fixed operating costs, variable operating costs and general operating expenses. General overheads are excluded from these costs, since it assumed that these are part of the Ocean Ecopark.

Fixed operating costs

The fixed operating costs include costs for maintenance, capital charges and local taxes and insurance. Maintenance is assumed to take 1 hour per week and costs approximately 1000 euro per year. The capital charges include interest payments on the loans taken out to finance the project. In this analysis it is assumed that capital is raised through debt financing at an interest rate of 8 percent. Local taxes and insurance are both estimated at one percent of the total physical plant costs (31).

Variable operating costs

The electricity required to drive the pumps and air fan form the largest component of the variable operating costs. The price of electricity in Curacao is 0.28 euro per kWh. This value is based on data of from Auqualectra, the water and power provider in Curacao, for April 2014. The costs of cleaning and replacement of components are also part of the variable operating costs. The lifetime of the direct contact dehumidifier is estimated at 20 years. Since the packed bed experiences unpurified atmospheric air and fresh water, it will likely have to be replaced after 10 years.

General operating costs

The general overhead expenses are excluded from this analysis since these are covered by the Ocean Ecopark. The only component of the general operating expenses is therefore the required investment for research and development. The dehumidifier is currently still in the development phase. The first step in realising a large scale direct contact dehumidifier will be to construct a prototype. The cost for developing a prototype is estimated at 2500 euro.

6.1.2 ECONOMIC EVALUATION METHOD

There are various methods available to determine the financial viability of a project. In this study, the present value method is chosen. This method is suitable for calculating the price of water and it takes annual variations and the time value of money into account. This makes it one of the more accurate economic analysis methods available in literature (31). The time value of money indicates that money earned early on in the project is more valuable than money earned later in the project. Money which has been earned can be reinvested and hence earn returns. The present value method takes this into account by calculating the net present value of a future sum on money. This is done by determining the cash flow in each year of the project, which is the required future sum of money, and calculating the present value of this cash flow using Equation 49. In this equation i represents the discount rate and n the number of years. Summing the present value of all these years gives a net present value for the entire project. Equation 50 summarizes the present value method and gives the net present value (NPV) as a function of the yearly cash flow (CF_n).

$$Present\ value = \frac{Future\ worth}{(1+i)^n} \tag{49}$$

$$NPV = \frac{\sum_{t=1}^{t=n} CF_n}{(1+i)^n} \tag{50}$$

The future revenues of the project are currently unknown since these depend on the price of water. The net present value, calculated according to Equation 50, is therefore solely based on the production cost of the project. The net present value represents the total investment required to cover the cost of the project. The price of water can then be calculated by dividing this investment over the total water produced during the project lifetime. Since the water production also represent a future worth, the net present value of the water production also needs to be found. The price of water is then equal to the net present value of the production costs (NPVC) divided by the net present value of the water production (NPVW).

$$Price\ of\ water = \frac{NPVC}{NPVW} \tag{51}$$

6.2 Results of Economic Analysis

6.2.1 Fixed capital costs

The purchased component costs of the dehumidifier are shown in Table 17. It can be seen that the column and the packed bed column are the most costly components of the dehumidifier. The physical plant cost, which includes installation, design and engineering is 33 percent more than the purchased component costs. This is mainly due to the additional costs of design and engineering of the heat exchanger. When the contractor's fee and contingency costs are added to the physical plant cost, the total fixed capital cost is 27,486 euro.

Table 17: Purchased cost of direct contact dehumidifier in Curacao

Purchased Cost	
Column	€ 4,962
Packed bed	€ 3,107
Heat Exchanger	€ 6,758
Auxiliaries	€ 331
Seawater pump	€ 1.027
Fresh water pump	€ 3.371
Air fan	€ 2.551
Fresh water tank	€ 1.505

6.2.2 Operating cost

The yearly operating cost of the direct contact dehumidifier is 2,746 euro. It is assumed that the dehumidifier only start working in the second year of its project life and the yearly operating costs will therefore only be added to the cash flow in this period. The operating cost is mainly determined by the cost for maintenance (1,040 euro per year) and electricity (852 euro per year). However, due to the low energy requirement of the system these costs do not significantly affect the price of water.

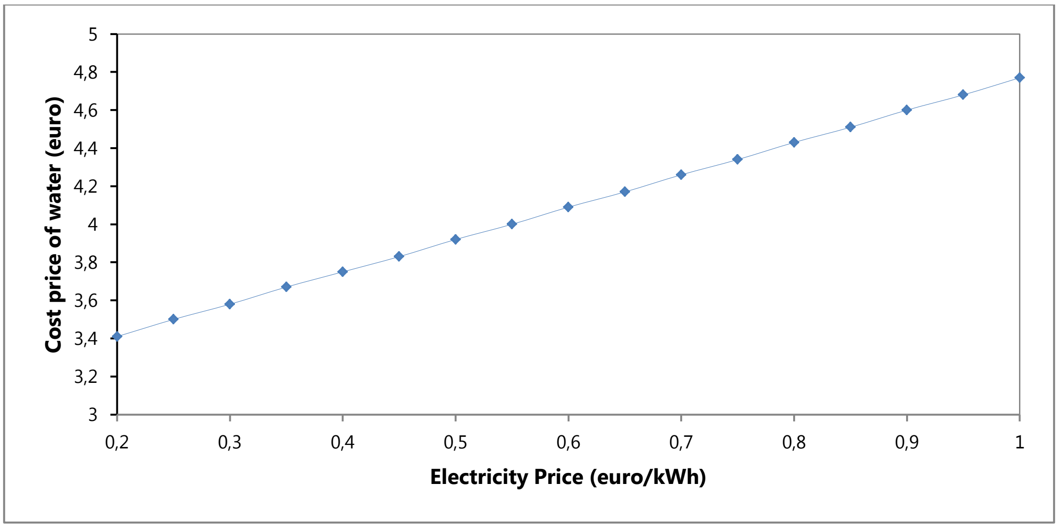

Figure 29: Effect of electricity price on cost price of water

6.2.3 Cost price of water

The calculated cost price of the fresh water produced by the direct contact dehumidifier is 3.55 euro per cubic meter of water (euro/m^3). This is the price that covers the costs of the total project and it does not include a profit margin. The price of water in April 2014 in Curacao for commercial purpose was 5.45 euro/m^3. Bluerise also provided data on the performance of reverse osmosis system in an Ocean Ecopark. This system produces 7200 litres per day at a cost price of 4.13 euro/m^3. The dehumidifier therefore seems to be economically competitive with both currently used water production technologies in Curacao and a potential reverse osmosis system. Figure 30 shows that the cost price of water from the dehumidifier remains competitive with current water prices up until a discount rate of 19 percent. Even when a profit percentage of 10 percent is added to the cost price, a discount rate up until 16 percent will still be sufficient to obtain a lower price than the current water price in Curacao.

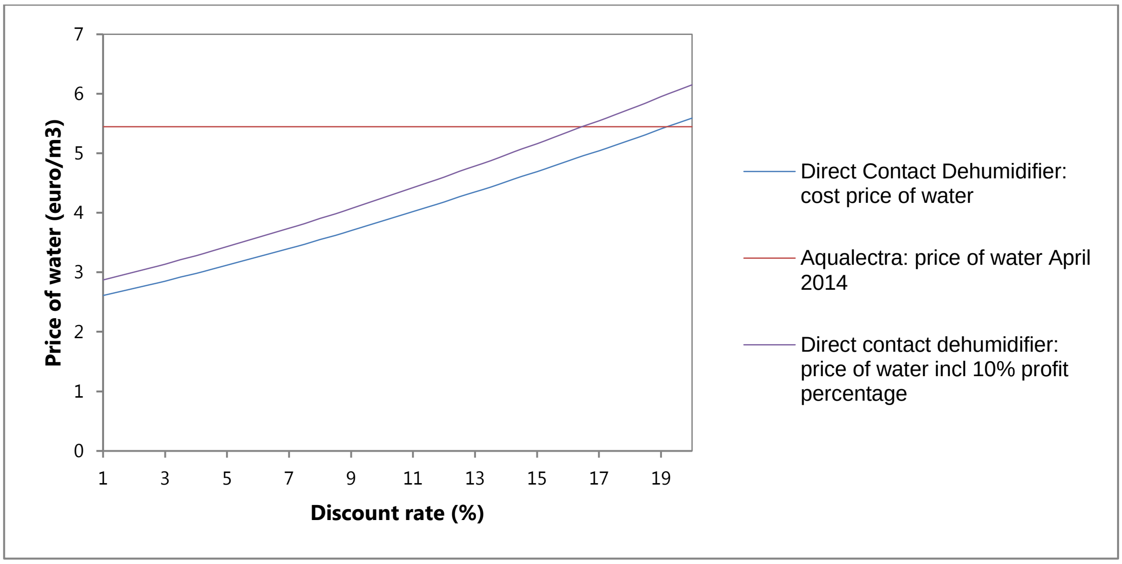

Figure 30: Effect of discount rate on price of water

6.3 Conclusion

The present value method was used to evaluate the economic performance of the direct contact dehumidifier in Curacao. The results of this analysis showed that the fixed capital costs are 27,486 euro. These costs are mainly determined by the cost of the packed bed tower and the external heat exchanger. Maintenance and the auxiliary energy requirement of the dehumidifier have the largest impact on the operational expenditures. The yearly operational expenditures for this system configuration are 2746 euro. The cost price of water is 3.55 euro per cubic meter of water. This value is competitive with the current water price in Curacao, which is 5.33 euro per cubic meter of water and with a potential reverse osmosis system. Even when a profit percent of 10 percent is added to the cost price, the price of the water produced by the direct contact dehumidifier remains lower than the current water price in Curacao for discount rates below 16 percent. Also, the low auxiliary energy requirement makes that the dehumidifier is less sensitive to the electricity price.

7. DISCUSSION

7.1 IMPROVEMENT POTENTIAL COMPACT PLASTIC HEAT EXCHANGER

The results of the multi-criteria analysis showed that the expected performance of an optimized direct contact dehumidifier would be better than for a tube condenser. The reason being that the heat transfer coefficients are similar but the specific heat transfer area of the packed bed is significantly higher. Increasing the specific heat transfer area of the tube condenser would lead to a similar system design as the compact plastic heat exchanger. This heat exchanger was omitted from the multi-criteria analysis since its component cost and energy requirements were too high and there seemed to be no significant improvement potential. The component costs are specified by Heat matrix B.V. and are based on a system used for heat extraction from high temperature flue gases. The material requirements of the compact plastic heat exchanger in this application are therefore much higher than when used for atmospheric water extraction. Further development of the plastic heat exchanger and optimizing the material for dehumidification could possibly lead to improved results and lower component costs. Research and development for this optimization would however be costly. R&D would include material testing and the fabrication of multiple tube bundles to investigate the effect of dehumidification of air in the tubes. An optimized compact plastic heat exchanger will still have a comparable heat transfer coefficient to the direct contact heat exchanger since the heat transfer resistance is determined by the non-condensable gases in the air. The performance will therefore likely be comparable to the direct contact dehumidifier. However, the direct contact dehumidifier has the additional advantage that it only uses seawater in an external loop and that the reliability of the components is high. The cost to construct an optimized system is also much lower since no significant research and development investments are required. The choice to further investigate the direct contact dehumidifier is therefore still valid.

7.2 AUXILIARY ENERGY REQUIREMENT OF DIRECT CONTACT DEHUMIDIFIER

This study showed that is possible to construct an ocean thermal dehumidifier with a production rate of 5000 litres per day and an auxiliary energy requirement of less than 2 kWh/m^3. Bluerise also obtained a quote for a similar sized reverse osmosis system for an Ocean Ecopark. The energy requirement of this system is 7.7 kWh/m^3, which is significantly higher than for the dehumidifier. Also, previous research on atmospheric water extraction with deep seawater obtained energy requirements of 5.9 kWh/m^3 for a production of near 6000 m^3 per day (10). This shows that the use of state of-the art dehumidifiers is beneficial.

The low auxiliary energy requirement also makes the dehumidifier less dependent on the electricity price. Doubling the electricity price makes it still possible produce water at a cost price below 5 euro/m^3. The dependence on the electricity price could be further reduced when electricity from an OTEC plant is used. The cost price of electricity from an OTEC plant is near 0.20 euro per kWh. Obtaining this electricity directly at its cost price would reduce the operational cost of the dehumidifier, but it would also decrease the revenues of an OTEC plant since the market price of electricity is likely higher than the cost price. Therefore, in this case study it has been assumed that electricity for needed for the dehumidifier is obtained from the grid at market price.

7.3 Seawater efficiency of direct contact dehumidifier

The inlet seawater temperature considered in this analysis has been 6 degrees Celsius. This means that water directly from the deep ocean is used in the dehumidifier. When the dehumidifier is placed in a series configuration with other ocean thermal technologies such as OTEC, the seawater temperature might be higher. This will impact the water production rate and requires different operational conditions. The addition of a low cost humidifier in this case could be beneficial and should also be further investigated.

The outlet seawater temperature for the ocean thermal dehumidifier as described in section 5.2 is almost 21 degrees Celsius. This provides limited opportunities for cooling, but it could be used in a humidifier. The temperature of the seawater would be lower than the atmospheric air and evaporative cooling would take place in the humidifier. This increases the humidity of the air, but decreases its temperature. A suitable heating source is therefore required to make the humidifier an additional advantage.

7.4 Feasibility of adding a humidifier

As mentioned in chapter 1, the humidification-dehumidification cycle consists of a humidifier and dehumidifier. Seawater is used for both humidifying and dehumidifying humid air. The HDH-cycle can take on different configurations. In an open air closed water cycle air is continuously refreshed, while the seawater remains in a closed loop. In an closed air open water cycle the air is in a closed loop and the sweater is continuously refreshed. This means that in an open air system water vapour is the fresh water resource and in open water system seawater is the fresh water resource.

The direct contact dehumidifier analysed in this research differed from direct contact HDH-cycles on two points; atmospheric air is directly used in the dehumidifier, instead of first being heated and humidified in a humidifier, and deep ocean water is used for cooling the fresh water loop instead of surface water. This section will look into the effect of these modifications on the air and fresh water temperatures throughout the packed bed and the fresh water production rate.

Comparison with dehumidifier with heated air and surface water

Figure 31 shows the humidity ratio in a packed bed tower for atmospheric air and deep seawater (AA-DSW) and Figure 32 shows the humidity ratio for humidified air and surface seawater (HA-SSW). Both conditions are simulated with a fresh water mass flux of 0.5 kg/m2*s. This value is favourable for the AA-DSW dehumidifier, as was shown in chapter 5 of this report, and experiments at the University of Florida (14) showed that this is also a suitable for range for a HA-SSW dehumidifier. The total height of the packed is 1 meter for both systems. The figures only show the condensation process in the packed bed. Part of the packed bed of the AA-DSW dehumidifier is used for saturation of the air and therefore its value on the graph is slightly less than 1 meter (0.917 m). The air has a saturation temperature of 22.66 degrees Celsius when water vapour starts to condense. The air to water ratio in these simulations is taken as 1. At the entrance of the dehumidifier the temperature of the fresh water needs to be lower than the air to initiate heat transfer and condensation. When a higher air to water ratio is used, the fresh water in the HA-SSW dehumidifier heats up too much and this temperature difference cannot be achieved.

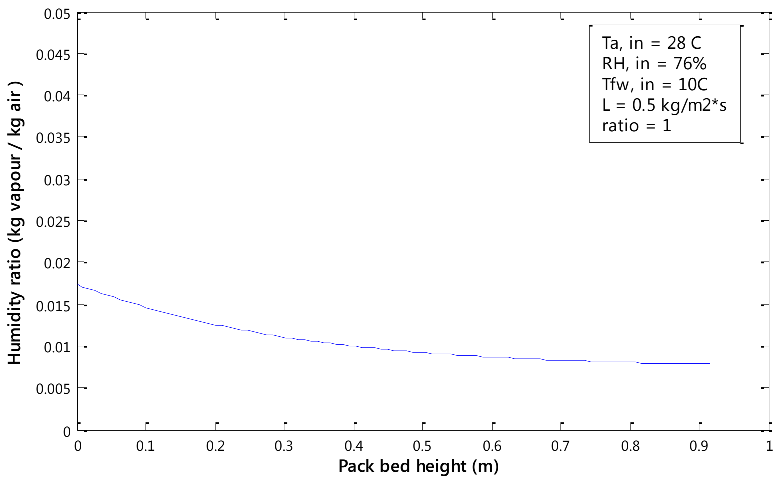

FIGURE 31: HUMIDITY PROFILE FOR CONDENSATION OF ATMOSPHERIC AIR WITH DEEP OCEAN WATER

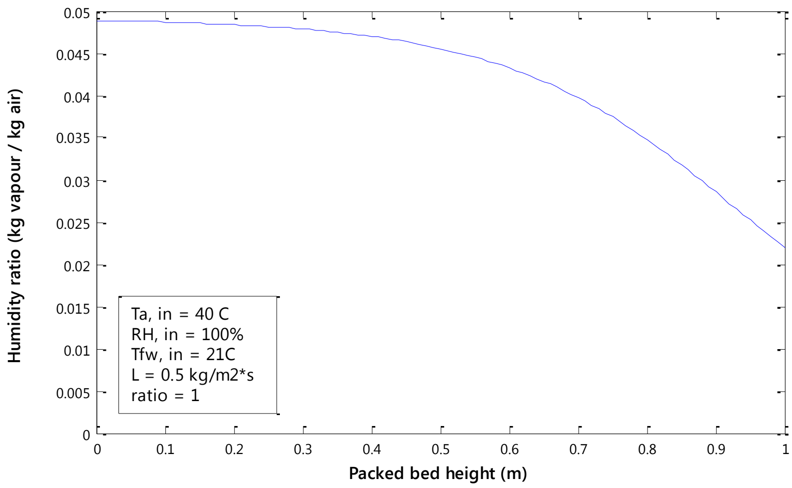

FIGURE 32: HUMIDITY PROFILE FOR CONDENSATION OF HUMID AIR WITH SURFACE SEAWATER

The figures show that the HA-SSW dehumidifier is able to produce more fresh water than the AA-DSW. The HA-SSW dehumidifier produces 0.0135 kg/s*m2 and the AA-DSW 0.0048 kg/s*m2. The difference in the shape of the graph is due to the different temperature profiles. Figure 33 and 34 show the air and water temperature profiles for both of the dehumidifiers. In the AA-DSW the temperature difference at the bottom of the tower is high, which leads to more condensation and hence a steep humidity curve. For the HA-SSW this is reversed; the temperature difference is high at the top of the tower which makes the humidity ratio decrease more quickly in this part. If production is higher, more condensation takes place and hence more heat is produced. If the fresh water enters at the top of the tower, it will heat up more quickly and therefore the temperature difference with the air at the bottom of the tower will be smaller.

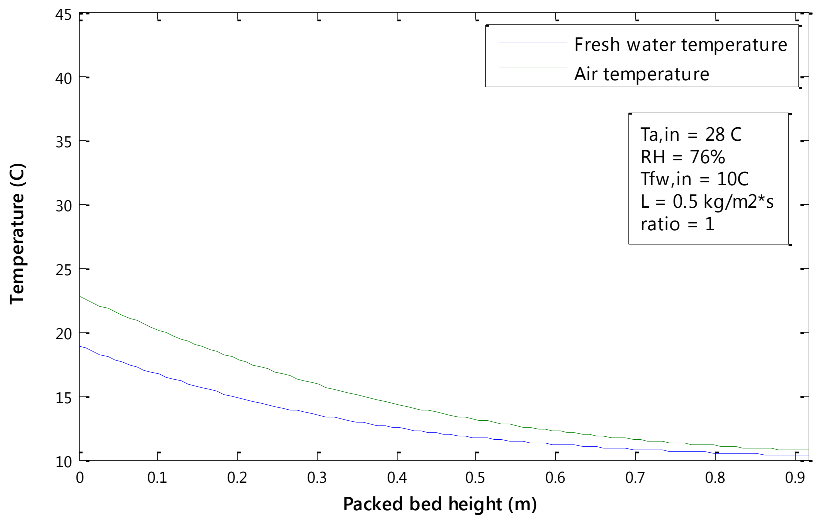

FIGURE 33: TEMPERATURE PROFILE FOR CONDENSATION OF ATMOSPHERIC AIR WITH DEEP OCEAN WATER AT AN AIR TO WATER RATIO OF 1

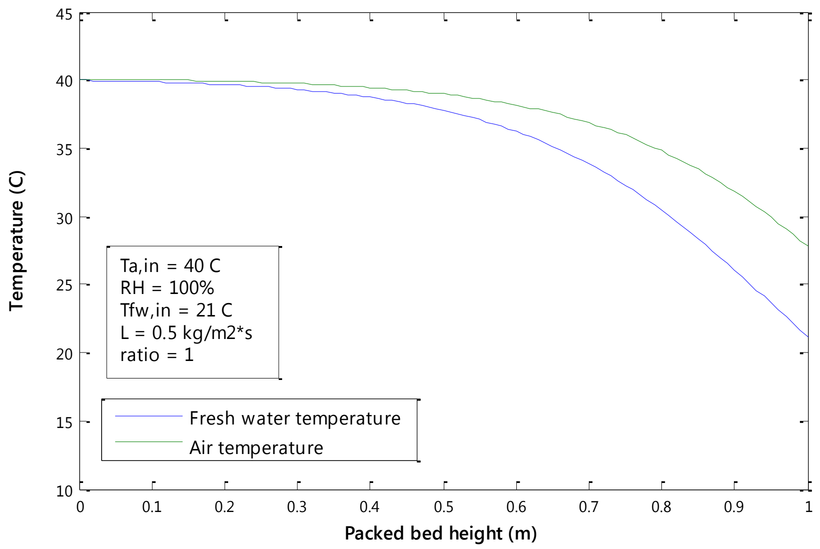

FIGURE 34: TEMPERATURE PROFILE FOR CONDENSATION OF ATMOSPHERIC AIR WITH DEEP OCEAN WATER AT AN AIR TO WATER RATIO OF 1

Comparison with dehumidifier with heated air and deep seawater

The production rate of the dehumidifier is even further increased when deep seawater is used to cool humidified air. Figure 35 and 36 show the temperature profiles of a dehumidifier with humid air and deep seawater (HA-DSW). The production for this dehumidifier is 0.02 kg/m2*s. This is about four times as high compared to the AA-DSW. The auxiliary energy requirement of the HA-DSW dehumidifier would be 0.52 kWh/m^3 and the estimated component cost 12,223. For a AA-DSW dehumdifier with similar operating conditions this would be 3.24 kWh/m^3 and 24,000 euro. These values are subject to further optimization, but the increase in production rate seems to show potential for a decrease in auxiliary energy requirement and component cost.

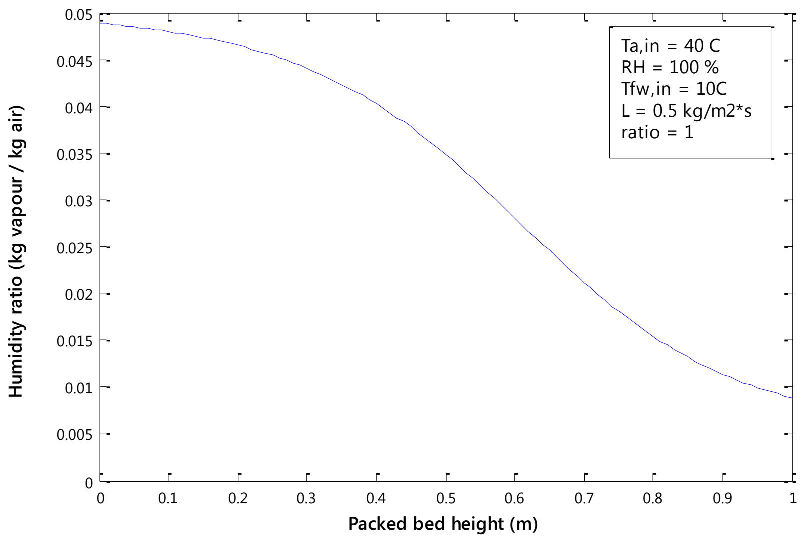

FIGURE 35: HUMIDITY PROFILE OF CONDENSATION OF HUMID AIR WITH DEEP OCEAN WATER

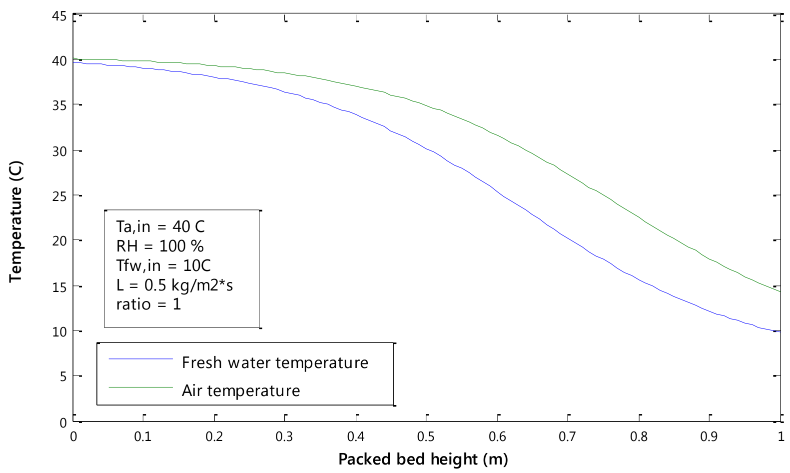

FIGURE 36: TEMPERATURE PROFILE OF CONDENSATION OF HUMID AIR WITH DEEP OCEAN WATER

7.5 Production of cold air for additional revenue

The dehumidifier processes air at a mass flow rate of 5.93 kg/s for a production of almost 5000 litres per day. The temperature of the exit air is 11.8 degrees Celsius. This is a sizeable amount of air and at a temperature which is sufficient for cooling or ventilation. The application of this air should be further considered at the project site, examples could be a greenhouse or visitor centre, but it is clear that the cold air could be a source of additional revenue.

For example, if a room needs to be cooled from 28 to 20 degrees a cooling temperature of 12 degrees should be sufficient. Depending on the rate of heat infiltration and the size of the room, a certain ventilation rate is required to keep the room at 20 degrees. The dehumidifier is able to provide a ventilation rate of 5.93 kg/s. Assuming that heat infiltration requires an air change rate of 10 times an hour, the dehumidifier is able to cool a room of approximately 1700 m^3.

$$ventilation\ rate = air\ change\ rate\ x\ room\ volume$$

The cooling load this room would normally require is calculated according to Equation … which gives a load of 16 MJ. The mass of air in the room which needs to be cooled is equal to its volume multiplied by the density of the air. At an air change rate of 10 times an hour, this means that each hour this cooling load is required. This brings the total cooling load per year 141 GJ. The cooling cost in Curacao is 21.26 euro per GJ of cooling. Assuming that the price of cooling is cost based, the yearly additional revenue could be 3000 euro. The yearly revenue of the water sold is almost 6500 euro. The additional revenue of due to cooling could therefore significantly decrease the profitability of the dehumidifier.

$$Cooling\ load = m_A * Cp_A * (T_{room} - T_{required})$$

7.6 Effect of fresh water production rate on component cost

The fresh water production rate in the feasibility study has been kept constant at 5000 litres per day. A sensitivity analysis shows that for similar operational conditions, the component cost per cubic meter of water produced decreases. The auxiliary energy requirement however remains quite similar; the energy requirement for production rates between 2500 L/d and 20,000 L/d ranged between 1.67 to 1.82 kWh/m^3.

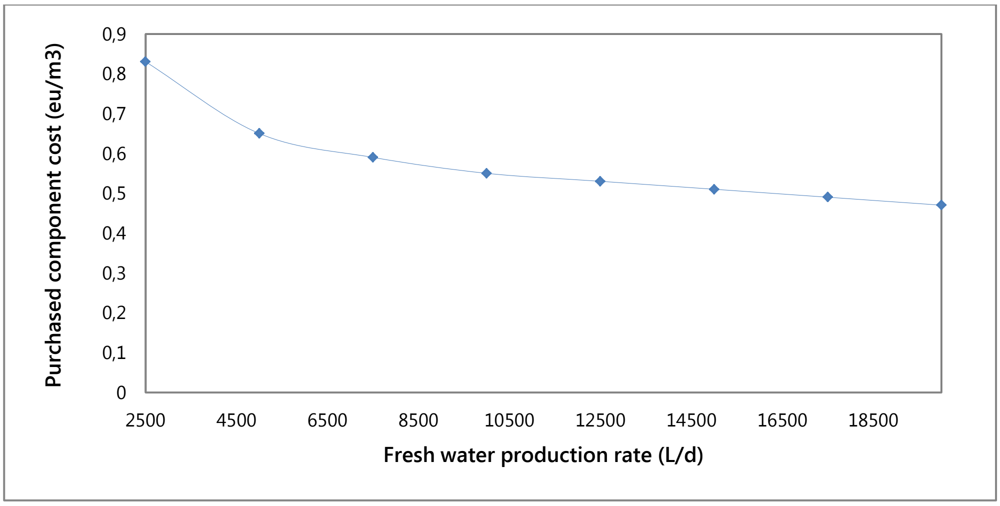

Figure 37: Effect of fresh water production rate on purchased component costs of the direct contact humidifier

8. CONCLUSION

The aim of this research has been to determine the feasibility of integrating a dehumidifier for atmospheric water extraction in an Ocean Ecopark. To determine this feasibility, three topics have been discussed: dehumidification method, technical feasibility and economic feasibility of the dehumidifier. This conclusion will therefore follow this structure.

Dehumidification method

Three ocean thermal dehumidifiers have been compared in a multi-criteria analysis; a tube condenser, compact plastic heat exchanger and a direct contact dehumidifier which consists of a packed bed tower and external heat exchanger. This analysis showed that direct contact heat transfer is a more cost effective method for dehumidification of non-saturated air with deep ocean water. The heat transfer coefficient for direct contact and indirect contact heat transfer is similar for all three systems due to the large amount of non-condensable gases. However, the large specific heat transfer area of the packed bed allows for more efficient heat transfer and a compact design. In addition, seawater is only used in the external heat exchanger which makes cleaning and maintenance costs low. The reliability of the direct contact dehumidifier is high since it consists of commercially available components. A high specific heat transfer area could be achieved with an indirect contact heat transfer method, but leads to high material costs since seawater is used directly in the system. A compact plastic heat exchanger could be a solution for the high material cost, but significant R&D investments are required for optimization of the design and material for atmospheric water extraction.

Feasibility study

The technical feasibility study showed the impact of the tower operating conditions (fresh water and air mass flux, the temperature of the fresh water at the inlet of the tower) and sizing of the heat exchanger (plate spacing) on the performance of the dehumidifier. This lead to the following insights:

- **Air to water ratio**
  An increase in ratio leads to a decrease in auxiliary energy requirement and component cost. The decrease in energy requirement only occurs to a certain extent; the increase in ratio leads to less energy loss in the heat exchanger, but an in increase power requirements for the air fan.
- **Fresh water temperature**
  Increasing the fresh water temperature from 8 to 12 degrees Celsius at the inlet of the tower increases the auxiliary energy requirement but decreases component cost.
- **Fresh water mass flux**
  A fresh water mass flux of 0.5 kg/m2*s leads to a lower energy requirement compared to a mass flux of 1.5 kg/m2*s.

The heat exchanger is a critical component in minimizing both material costs and energy losses. Optimization of plate spacing made it possible to find a technical feasible configuration with an energy requirement of less than 2 kWh/m^3 and component costs less than 25,000 euro for a fresh water production rate of almost 5000 litres per day. These values are competitive with current desalination technologies and the dehumidifier is therefore considered technically feasible.

The economic performance of the direct contact dehumidifier in Curacao was evaluated using the present value method. This analysis gave three clear economic indicators; the fixed capital costs are 27,486 euro, the yearly operational expenditures are 2746 euro and the cost price of water is 3.55 euro per cubic meter of water produced. This price is lower than the current water price in Curacao and the cost price of a potential reverse osmosis system in an Ocean Ecopark. Even when a profit percent of 10 percent is added to the cost price, it remains lower than the current water price for discount rates below 16 percent.

This research has led to development of a detailed computational model which simulates the performance of a direct contact dehumidifier. This model is constructed such that it can simulate dehumidification of both saturated and non-saturated air at various mass flow rates and temperatures. The component sizes and operating conditions are variable, such that different system configurations can be tested. Integrated energy and component cost calculations make it possible to quickly predict the feasibility of an ocean thermal dehumidifier. In this research this has been done for a direct contact dehumidifier in an Ocean Ecopark in Curacao. The integrated cost and energy calculations proved to be sufficient and only a limited amount of design parameters had to be explored in order to find a feasible system configuration. With a cost price of water of 3.55 euro per cubic meter of water produced, the ocean thermal dehumidifier has shown its potential. In addition, cold air is produced which could provide an additional source of revenue and further improve the financial viability of the system.

9. Recommendations

This research showed the feasibility of an ocean thermal dehumidifier for fresh water production. This section will describe recommendations for further development of this technology.

Physical testing

The first step is to validate the direct contact computational model with physical experiments. The computational code for condensation in a packed bed has been verified at the University of Floria, but the results for using unsaturated air have not been tested. Therefore, construction of a prototype could give more insight in the validity of the obtained results. Students from Delft University of Technology have researched the construction of a prototype, but due the extensive computational modelling required for evaporative cooling in a packed bed they were unable to actually construct it. However, information on the required components and construction of the testing facility is readily available.

Investigating the feasibility of adding a humidifier

The discussion showed that adding a humidifier could increase production significantly; the production increased 4 times when humidified and heated air was used in the dehumidifier compared to atmospheric air. Adding a humidifier therefore shows potential to further decrease the cost per cubic meter of water produced. There are various humidifiers possible which could be integrated with the ocean thermal dehumidifier. The type of humidifier used should be determined at the project site. The main energy requirement of the humidifier is the heating source. The feasibility of adding a humidifier therefore strongly depends on the availability of a suitable heating source. In addition, possible synergies with other ocean thermal technologies should be investigated. For example, the dehumidifier could be integrated in a greenhouse or a low temperature thermal desalination technology.

Optimization Study

The design of the dehumidifier includes a large amount of variables, both operating conditions and component sizes, which influence its performance. In this research only a limited amount of variables in a specified range were explored to determine the system's feasibility. Now that it is shown that a dehumidifier in an Ocean Ecopark could be feasible, a full optimization study could be beneficial. This optimization study could take place by further extending the direct contact computational code or through integration of this code with a commercial heat exchange program.

Transient Analysis

This research assumed steady state climate conditions. Hourly temperature and humidity variations could however affect the performance of the dehumidifier. A transient analysis which taken these differences into account could be next step in determining an optimal system configuration.

10. REFLECTION

This section will reflect on the methods used in this research. The reflection consists of four parts; the selection of the dehumidifier type, the technical feasibility study, the economic analysis and the discussion of results.

Dehumidification method

There are various dehumidifiers which could possibly be applied in an Ocean Ecopark; both commercial designs and innovative technologies which are only described in literature. One of the difficulties encountered in this research has been to determine how to best compare these systems. The large number of unknown variables made a commercial design program unsuitable. It was therefore chosen to model a direct contact and indirect dehumidifier using MATLAB. This turned out to be a very time consuming process. The construction of the computational models was complex, especially since they needed to be adjusted for atmospheric air and measurement data was lacking. In addition, due to the large number of unknown variables, only a limited amount of configurations were explored. However, it did give more insight in the process of atmospheric water extraction. In the end, the most important conclusion was validated with an energy balance.

Technical feasibility study

This part of the research proved to be most insightful. In the selection process of the dehumidifier it became clear how much effect the operating conditions have on the total performance of the system. By using data from literature, it was possible to time-effectively determine suitable ranges for these conditions and find a feasible system configuration. One of the difficulties in this study has been the modelling of the heat exchanger and in specific the decision to which extent to model the design. Commercial design programs are able to design heat exchangers, which make the time consuming process of developing an iterative code which resembles this process irrelevant. However, some detail of the heat exchanger needed to be modelled in order to determine the effect of the operating conditions of the tower on the total dehumidifier design. During modelling of the heat exchanger more variables were encountered that have a large impact on the total performance which made that the results continuously had to be adjusted. This was frustrating at times, especially since modelling of a heat exchanger was not part of the initial research approach. However, in the end a feasible design has been found and all the variables which have an impact are identified. This can serve as a guideline for future research, which might include an optimization study.

Economic analysis

One of the technical feasible system configurations has been analysed for its economic feasibility. Due to the integrated calculations for energy and component costs in the computational models, the technical feasibility already included indicators for economic feasibility. The economic feasibility study therefore showed promising results. This analysis also indicated that the approach to include cost and energy calculations in the models aid in quickly determining the feasibility of a dehumidifier.

Discussion of results

Due to the unexpected additional modelling of the direct contact dehumidifier, limited time remained for investigation of further improvement of the system and possible synergies. However, the constructed model can easily be applied for different situations and applications and could therefore be used for future research.

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APPENDIX

A.1 PSYCHOMETRIC CHART

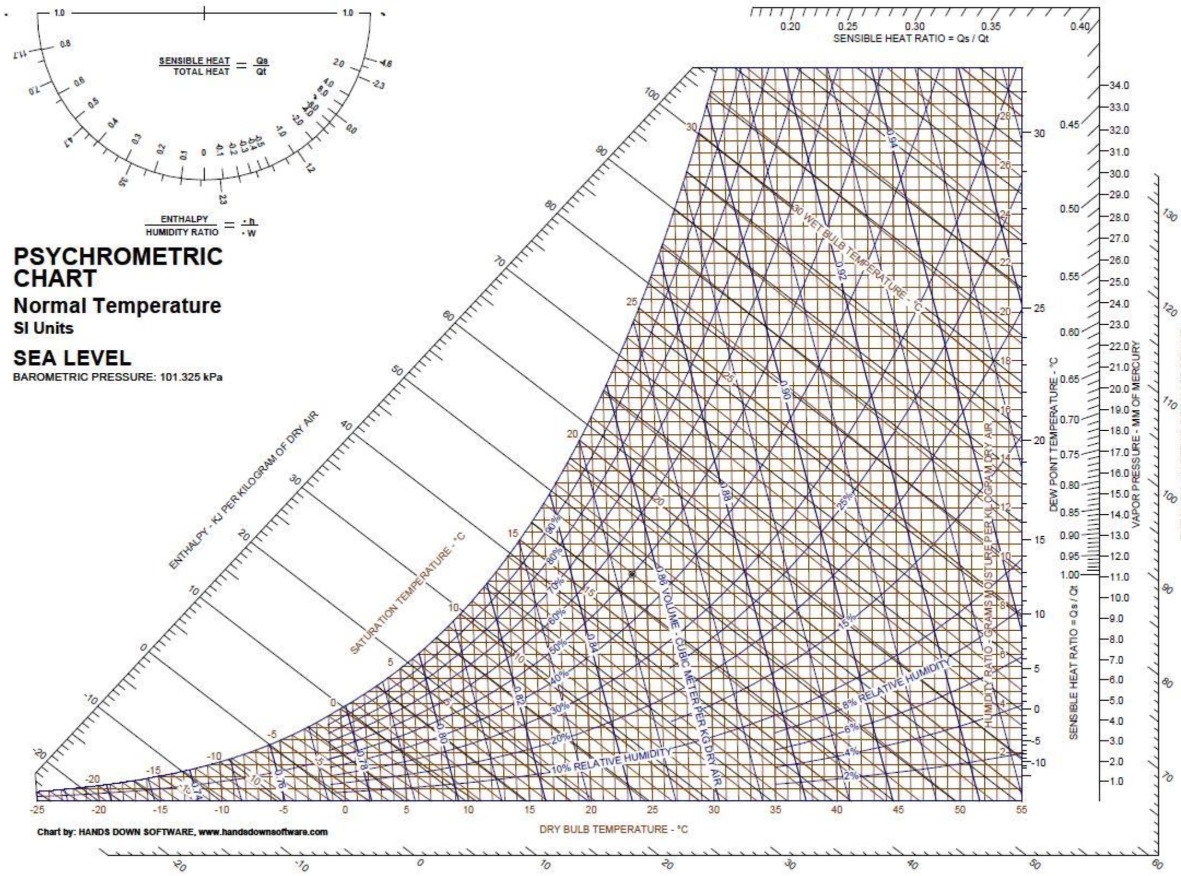
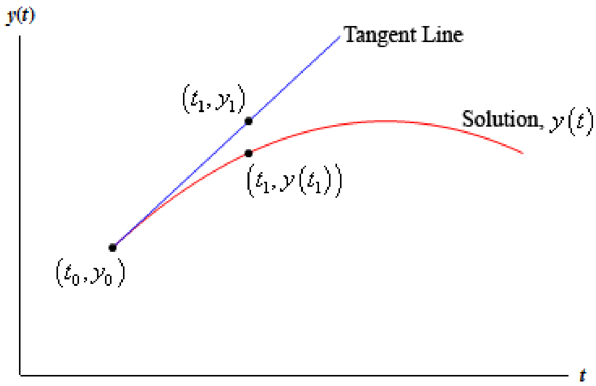


FIGURE 38: PSYCHOMETRIC CHART



In linear equations the change in y is constant, while in non-linear equations it varies with t. This gives that the derivative (dy/dt) at a specific point on the graph of non-linear equations is variable. Each point on a non-linear graph can be described by a tangent line. When sufficiently small steps are taken, the slope of this line can be taken as the slope of the non-linear line since the difference will be negligible. The Euler method is based on this concept. The differential equations describing the temperature and humidity gradient in the packed bed are linearized using MAPLE. The equation for air temperature and humidity are then combined such that an expression for the increase in humidity can be found. This then makes it possible to solve the equations for air and temperature gradients.

Equations derived from MAPLE

$$dTl := dz \cdot \left(\frac{G \cdot dw \cdot (hfg - hl)}{L \cdot dz \cdot Cpl} + \frac{U \cdot a \cdot (Tl - Ta)}{Cpl \cdot L} \right);$$

$$dTa := dz \cdot \left(-\frac{1}{1+w} \cdot \frac{dw \cdot hl}{dz \cdot Cpg} + \frac{U \cdot a}{Cpg \cdot G} \cdot \left(\frac{1}{1+w} \right) \cdot (Tl - Ta) \right);$$

$$dw = dTa \cdot \frac{P}{P - Psat} \cdot w \cdot \left(b - 2 \cdot c \cdot Ta + 3 \cdot x \cdot Ta^2\right);$$

$$dw = \frac{1}{P - Psat} \left(dz \left(-\frac{dw\, hl}{(1+w)\, dz\, Cpg} + \frac{U\, a\, (Tl - Ta)}{Cpg\, G\, (1+w)} \right) P\, w \left(b - 2\, c\, Ta + 3\, x\, Ta^2\right) \right)$$

$$\xrightarrow{\text{isolate for dw}}$$

$$dw = \left(w\, P\, U\, a\, dz\, Tl\, b - 2\, w\, P\, U\, a\, dz\, Tl\, c\, Ta + 3\, w\, P\, U\, a\, dz\, Tl\, x\, Ta^2 - w\, P\, U\, a\, dz\, Ta\, b + 2\, w\, P\, U\, a\, dz\, Ta^2\, c - 3\, w\, P\, U\, a\, dz\, Ta^3\, x\right) / \left(Cpg\, G\, P - Cpg\, G\, Psat + Cpg\, G\, w\, P - Cpg\, G\, w\, Psat + w\, P\, hl\, G\, b - 2\, w\, P\, hl\, G\, c\, Ta + 3\, w\, P\, hl\, G\, x\, Ta^2\right)$$

Symbols used in MAPLE

Tl	*fresh water temperature*
Ta	*humid air temperature*
w	*humidity*
b, c, x	*coefficients*
z	*packed bed height*

A.3 CALCULATION APPROACH FOR ADIABATIC EVAPORATIVE COOLING IN A PACKED BED

Calculation method for saturation temperature of air

The expressions described in this section are based on a calculation method described in a paper by Tahri et al (19). First, the saturation temperature of the air is estimated. In the first iteration, this value is used to determine the enthalpy of vaporization (Equation A1).

$$hfg_{ec} = 2502 * \left(\frac{1-\frac{(T_{G,sat,ec})}{T_{cr}}}{1-\frac{273.15}{T_{cr}}}\right)^{0.38} \quad \text{(A1)}$$

The humid heat is calculated according to Equation A2 and the absolute humidity at the saturation temperature is found with Equation A3.

$$Cs = Cp_A + w_{db} * Cp_V \quad \text{(A2)}$$

$$w_{sat,ec} = \left(T_{G,sat,ec} - T_{G,db}\right) * -\frac{Cs}{hfg} + w_{db} \quad \text{(A3)}$$

This value is used to obtain the saturation pressure of water vapour in mm Hg (Equation A4) and the saturation temperature in degrees Celsius (Equation A5).

$$P_{V,sat,ec} = \frac{760 * w_{sat,ec}}{\frac{18}{29} + w_{sat,ec}} \quad \text{(A4)}$$

$$T_{G,sat,ec} = \frac{1689.52}{8 - \mathrm{LOG}(P_{V,sat,ec})} - 230 \quad \text{(A5)}$$

The calculated saturation temperature is then compared to the estimated saturation temperature. If these are not equal, a second iteration will start in which the calculated saturation temperature is used as initial value.

Calculation method for required packed bed height to reach saturation

Applying a steady state mass balance on the water vapour in the packed bed during evaporation gives Equation A6. The amount of water evaporated is equal to the difference between the mass of water vapour at the outlet and at the inlet.

$$\Phi M_{V,evap} = \Phi M_{V,out} - \Phi M_{V,in} \tag{A6}$$

The humidity ratio is given as the mass of water vapour per unit mass of dry air (Equation A8). Since dry air makes up more than 95% of the total humid air, the mass flow rate of air is approximately similar to the mass flow rate of dry air (equation A7).

$$\Phi M_A = G * A_{cross} \tag{A7}$$

$$w = \frac{M_A}{M_V} \tag{A8}$$

Combining Equations A7 and A8 gives the total evaporation rate in the bottom part of the tower.

$$\Phi M_{V,evap} = G * A_{cross} * (w_{out} - w_{in}) \tag{A9}$$

The evaporation can also be described using the formulation of the mass transfer coefficient, given in Equation A10. This formula is derived from the book Heat Transfer by Nellis and Klein (33).

$$\Phi M_{V,evap} = km_G * A * \left(Cn_{v,surf} - Cn_{v,air}\right) \tag{A10}$$

$$with\, A = a * H * A_{cross}$$

The mass transfer coefficient of air is calculated according to Equation A11 (14). The values used in this formula are analysed at the average temperature of the air in the bottom part of the packed bed.

$$km_G = 5.23 * Re^{0.7} * Sc^{\frac{1}{3}} * (a * dp)^{-2} * a * D_G \tag{A11}$$

The concentrations of water vapour at the surface and in the air is the density at the partial pressure and temperature. The humidity ratio is then given by Equation A12.

$$w = \frac{Cn_V\,(\frac{kg\,vapor}{m3\,air})}{\rho_A\,(\frac{kg\,air}{m3\,air})} \tag{A 12}$$

The humidity ratio of the air changes over the height of the packed bed and therefore the average humidity ratio is taken as

$$w_{avg} = \frac{w_{in} + w_{out}}{2} \tag{A 13}$$

It is assumed that cooling at constant enthalpy takes place and therefore the saturation humidity and temperature of the air are defined. At this point no evaporation takes place and the water vapour concentration and humidity ratios at the surface and the air is equal.

$$w_{surf} = w_{air,sat} \tag{A14}$$

Assuming that the humidity ratio at the surface is constant gives Equation A15.

$$Height = \frac{G * (w_{air,sat} - w_{in})}{k_{mG} * a * (w_{avg} - w_{surf})} \tag{A 15}$$

A.4 Measurement data of a Seawater Greenhouse

Table 18, 19 and 20 present the data provided by the University of Wageningen. The temperature measurements are taken at the top, middle and bottom of the evaporator, first tube of the condenser and last tube of the condenser. The temperatures are indicated in degrees Celsius. Table 21 presents additional measurement data on the performance of the tube condenser.

Table 18: Temperature measurements at evaporator in seawater greenhouse

	East	West
Top	31	31.5
Middle	29	29.5
Bottom	26.5	26.5

Table 19: Temperature measurements at first tube in seawater greenhouse

	East	West
Top	28	28
Middle	27	27
Bottom	26.5	26.5

Table 20: Temperature measurements at last tube in seawater greenhouse

	East	West
Top	25.5	24.5
Middle	23.5	23
Bottom	22.5	22.5

Table 21: Additional measurements data on the performance of the tube condenser in a Seawater Greenhouse

Air	
Air velocity (m/s)	5
Relative humidity at inlet (%)	89
Water Production	
Temperature cold water tank (C)	21.3
Production rate (L/min)	0.825

A.5 Design parameters ocean thermal dehumidifiers

A.5.1 Direct Contact Dehumidification

Leading design parameters

The analysis done by the University of Florida indicated that the air mass flux, the fresh water mass flux and the height of the tower can impact the performance of the condensation tower. Since the DCD system makes use of (non-heated) atmospheric air, the temperature of the fresh water entering the tower is another important variable in the design. This temperature needs to be low enough for a suitable temperature difference between air and water, but also within the cost limitations of the external heat exchanger.

Selected system configurations

The packed bed height of the condensation tower at the University of Florida was 0.3 m. Part of the packed bed tower will also be used for evaporative cooling which requires more packed bed height and it can therefore be assumed that more height is needed compared to the tower in the Florida configuration. The initial value for the tower height is therefore 0.5 m and to determine the effect of the tower height on the performance, this value will be compared to a height of 1.0 m. The same approach is used to determine the values for the air mass flux and the water mass flux. In previous experiments (which used heated and humidified air) an air mass flux of 0.6 kg/m2*s was used with a fresh water mass flux ranging from 0.5-1.2 kg/m2*s. Assuming the temperature difference between the water and the air remains similar, these values seem a reasonable starting point. The effect of the mass flow rates on the performance will be tested by using a minimum and maximum value. These values are subject to the limitations of the heat exchanger and configurations which establish a similar pressure drop and heat transfer coefficient as in the commercial design will only be used.

Table 22: Configurations direct contact dehumidifier

Adjusted parameter	Configurations direct contact dehumidifier			
	Basic	**DCD-Height**	**DCD-Air rate**	**DCD-Water rate**
Packed Bed Height (m)	0.5	1.0	0.5	0.5
Air Mass Flux (kg/m2*s)	0.5	0.5	1.0	0.5
Fresh Water Mass Flux (kg/m2*s)	0.8	0.8	0.8	0.4

A.5.2 TUBE CONDENSER

Leading design parameters

Testing of the tube condenser has already been done in the Seawater Greenhouse. The system is therefore already optimized for the conditions in which the seawater greenhouse operates and only the components affected by the temperature of the deep seawater should be used in this analysis. This means that the tube sizing and material remain similar to the system used in the Seawater Greenhouse and the optimum air velocity and the number of tubes should be reconsidered in this new design. This is shown in Figure 39 the temperature difference between the air and seawater is much larger when using deep seawater than when using surface water. This additional temperature difference can be used to increase production by increasing the number of tubes or by adjusting the air velocity.

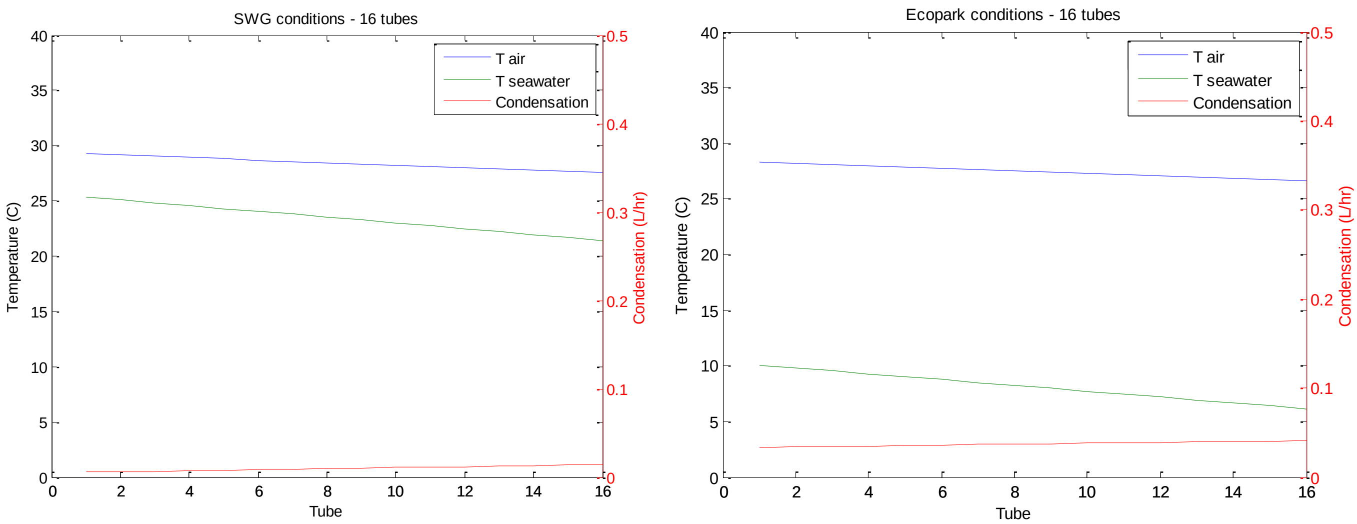


FIGURE 39: DIFFERENCE IN TEMPERATURE GRADIENT AND PRODUCTION FOR A TUBE CONDENSER WITH SURFACE SEAWATER AND DEEP OCEAN WATER

Selected system configurations

Increasing the number of tubes, when using a counter current flow between air and seawater, leads to a smaller temperature difference but more cooling of the air and therefore more production. In the computational analysis, the minimum value for number of tubes will therefore be the number of tubes used in the seawater greenhouse and the maximum value will be 32. The air velocity is another factor of influence since the air fan makes up the majority of the energy requirement of the system. The air velocity used in the Seawater Greenhouse is 5m/s, to investigate the influence of the air velocity on the performance of the system, an air velocity of 1 m/s will also be tested.

TABLE 23: CONFIGURATIONS TUBE CONDENSER

Adjusted parameter	Configurations tube condenser		
	Basic	**Tube-tubes**	**Tube-velocity**
Tubes in a row	16	65	16
Air velocity (m/s)	5	5	1

A.5.3 Compact Plastic Heat Exchanger

Data on the compact plastic heat exchanger is provded by Heat Matrix B.V. They provided two datasets: one with a high air velocity and one with a lower air velocity. The configuration in which a high velocity is used maximizes condensation, while the system with a low velocity maximizes cooling of the air.

Table 24: Configurations compact plastic heat exchanger

Adjusted parameter	Configurations compact plastic heat exchanger	
	Basic	**Compact-velocity**
Velocity (m/s)	9	1.1

A.6 SENSITIVITY ANALYSIS RATING IN MULTI-CRITERIA ANALYSIS

rating 25 - 50 - 100

	Direct contact dehumidifier			
	Importance	**Performance**	**Improvement Potential**	**Total**
Energy	5	50	100	25000
Component Cost	5	25	100	12500
Air temperature	4	50	50	10000
Efficiency	3	50	25	3750
Size	3	25	25	1875
Maintenance and Cleaning	3	100	25	7500
Reliability	4	100	50	20000
			TOTAL	**80625**

	Tube Condenser			
	Importance	**Performance**	**Improvement Potential**	**Total**
Energy	5	50	100	25000
Component Cost	5	50	25	6250
Air temperature	4	25	100	10000
Efficiency	3	25	50	3750
Size	3	25	25	1875
Maintenance and Cleaning	3	25	25	1875
Reliability	4	25	50	5000
			TOTAL	**53750**

Addition

	Direct contact dehumidifier			
	Importance	**Performance**	**Improvement Potential**	**Total**
Energy	5	1,5	2	8,5
Component Cost	5	1	2	8
Air temperature	4	1,5	1,5	7
Efficiency	3	1,5	1	5,5
Size	3	1	1	5
Maintenance and Cleaning	3	2	1	6
Reliability	4	2	1,5	7,5
			TOTAL	**47,5**

	Tube Condenser			
	Importance	**Performance**	**Improvement Potential**	**Total**
Energy	5	1,5	2	8,5
Component Cost	5	1,5	1	7,5
Air temperature	4	1	2	7
Efficiency	3	1	1,5	5,5
Size	3	1	1	5
Maintenance and Cleaning	3	1	1	5
Reliability	4	1	1,5	6,5
			TOTAL	**45**

A.7 CALCULATIONS WATER REQUIREMENT IN CASE STUDY

Greenhouse
Surface area = 720 m2 (20)
Water requirement greenhouse plant = 2.5 L/day
Surface area requirement tomato = 4 ft2 or 0.372 m2 (34)
Number of plants in greenhouse = 1935
Actual water requirement = 4838 L/d or 0.0560 L/s

Capacity Factor = 1
Average temperature and humidity used.

Availability Factor = 0.979
The downtime of an RO system which produces 7200 L/d is 1 hour per day. Since this is a smaller system, less components make use of seawater and there are less replacements (packing uses fresh water, lifetime of titanium heat exchanger is 30 years). Therefore, the downtime of the DCD system is taken as 0.5 hr per day.

Degradation factor =1

The theoretical water production which needs to be produced by the model is therefore
0.0560 / 0.979 = 0.0572 L/s
4838 / 0.979 = 4942 L/d