

**AN INNOVATIVE AND PRACTICAL DEFINITION OF
ENERGY EFFECTIVENESS FOR HUMIDIFICATION
DEHUMIDIFICATION DESALINATION SYSTEMS**

BY

ABDULAZIZ ABDULLAH ALSAIF

A Thesis Presented to the
DEANSHIP OF GRADUATE STUDIES

KING FAHD UNIVERSITY OF PETROLEUM & MINERALS

DHAHRAN, SAUDI ARABIA

In Partial Fulfillment of the
Requirements for the Degree of

MASTER OF SCIENCE

In

MECHANICAL ENGINEERING

December 2020

KING FAHD UNIVERSITY OF PETROLEUM & MINERALS

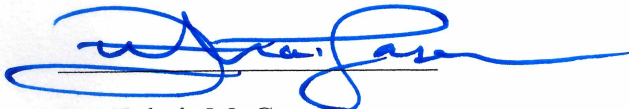
DHAHRAN- 31261, SAUDI ARABIA

DEANSHIP OF GRADUATE STUDIES

This thesis, written by **ABDULAZIZ ABDULLAH ALSAIF** under the direction of his thesis advisor and approved by his thesis committee, has been presented and accepted by the Dean of Graduate Studies, in partial fulfillment of the requirements for the degree of **MASTER OF SCIENCE IN MECHANICAL ENGINEERING.**



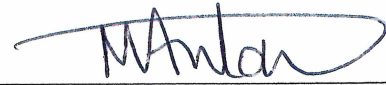
Dr. Syed M. Zubair
(Advisor)



Dr. Zuhair M. Gasem
Department Chairman



Dr. Suliman S. Al-Homidan
Dean of Graduate Studies



Dr. Mohammed A. Antar
(Member)

27/12/2020

Date



Dr. Syed A. Said
(Member)

© ABDULAZIZ ABDULLAH ALSAIF

2020

This Thesis Is Dedicated

To My Dear Parents, Brothers, and Sisters

To My Lovely Wife and Children

To My Friends and Colleagues

*To All Those Who Aspire To Improve Our Life
on This Planet*

With Love and Profound Gratitude.

ACKNOWLEDGMENTS

All praises and thanks be to Allah for giving me the power, health, blessing me with the intellect, strength, and patience to complete this work and attain this education level.

I have been fortunate that I have been granted the chance to pursue my Masters at King Fahd University of Petroleum & Minerals. I am taking this chance to direct my genuine thanks to my advisor Dr. Syed M. Zubair for his excellent guidance, advice, inspiration, diligence, assistance, patience, and support. Throughout carrying out this study, his guidance and advice aided and assisted me. Exceptional appreciation to both members of the committee, Dr. Mohammed A. Antar and Dr. Syed M. Said for their encouragement and helpful comments during the research.

Lastly and not least, my appreciation is extended to the Mechanical Engineering Department for their support and help. I want to express my thanks to my parents, wife, son, and daughters for their encouragement and support during the current research.

TABLE OF CONTENTS

ACKNOWLEDGMENTS	V
TABLE OF CONTENTS.....	VI
LIST OF TABLES.....	X
LIST OF FIGURES.....	XI
LIST OF ABBREVIATIONS.....	XIV
ABSTRACT.....	XV
ملخص الرسالة	XVII
CHAPTER 1 INTRODUCTION.....	1
1.1 Freshwater sources.....	3
1.2 Freshwater properties	4
1.3 The problem of freshwater scarcity	6
1.4 Resolution of freshwater scarcity	8
1.5 Desalination of freshwater	9
1.5.1 Historical background of freshwater desalination	9
1.5.2 Categorization of freshwater desalination techniques	10
1.5.3 Evolution of freshwater desalination techniques	11
1.6 The innovative part of this research	13
1.7 Objectives of the research	14

CHAPTER 2 LITERATURE REVIEW	16
2.1 Main desalination techniques	16
2.2 Humidification dehumidification systems	19
2.3 Evolution of performance parameters in humidification dehumidification systems	21
2.4 Thermodynamic balancing in humidification dehumidification systems	23
2.5 Design optimization of cost-effective HDH system	26
CHAPTER 3 PINCH-BASED ENERGY EFFECTIVENESS DEFINITION	27
Summary.....	27
3.1 Introduction.....	28
3.2 Description of the system under study.....	29
3.3 Temperature-enthalpy (T - h) diagram	30
3.4 Humidification dehumidification model.....	32
3.4.1 Mass and energy conservation	32
3.4.2 Terminal-based energy effectiveness definition	33
3.4.3 Proposed definition of humidifier effectiveness	35
3.4.4 Approximation of saturated air thermodynamic properties	39
3.4.5 Humidifier/dehumidifier sizing.....	43
3.4.6 Performance matrix.....	45
3.5 Verification of the proposed method	46
3.5.1 Maximum allowable terminal-based energy effectiveness.....	46
3.5.2 Enthalpy pinch.....	48
3.6 Comparison between terminal- and pinch-based energy effectiveness	49
3.7 Design procedure of the HDH system.....	55
3.7.1 Practical range of pinch-based energy effectiveness	55
3.7.2 Proxy indicator (BTC) optimization	57

3.7.3	Case study.....	60
3.8	Concluding remarks	63

CHAPTER 4 ANALYTICAL SOLUTION FOR THE PINCH-BASED ENERGY EFFECTIVENESS DEFINITION 64

Summary	64
4.1 Introduction	65
4.2 Performance and operating parameters.....	67
4.3 Analytical modeling	70
4.3.1 The balanced system, $MR = MR_{opt}$	73
4.3.2 The unbalanced system, Case 1: $MR > MR_{opt}$	78
4.3.3 The unbalanced system, Case 2: $MR < MR_{opt}$	80
4.3.4 Illustrative procedure.....	84
4.4 Validation and performance of the proposed analytical model	88
4.4.1 Impact of humidifier energy effectiveness	88
4.4.2 Impact of dehumidifier energy effectiveness	88
4.4.3 Impact of water top temperature	91
4.4.4 Impact of water bottom temperature	93
4.4.5 Validation of the proposed model.....	93
4.5 Concluding remarks	95

CHAPTER 5 THERMODYNAMIC BALANCING USING THE PROPOSED PINCH-BASED EFFECTIVENESS DEFINITION 96

Summary	96
5.1 Introduction	97
5.2 CAOW water heated humidification dehumidification arrangement under thermodynamic balancing (extraction)	98
5.3 Thermal balance in the humidifier and dehumidifier	100

5.3.1	Control volume balance – optimum operation (zero extraction)	100
5.3.2	Mass injection- and extraction-based (single extraction).....	101
5.4	Modeling of the proposed model.....	103
5.4.1	Zero extraction cycle	103
5.4.2	Single extraction cycle	106
5.4.3	Operational variables	108
5.5	Results and discussion	109
5.5.1	Impact of energy effectiveness	109
5.5.2	Impact of seawater temperature at inlets of humidifier/dehumidifier	113
5.6	Concluding remarks	119
CONCLUSION AND RECOMMENDATIONS		121
Recommendations		123
Nomenclature.....		124
REFERENCES.....		128
VITAE.....		140

LIST OF TABLES

Table 2.1. Comparison between MED and HDH desalination techniques [22].	17
Table 2.2. Evolution in the definition of performance parameters for humidification dehumidification desalination systems.	22
Table 3.1. Operational conditions basic values.....	46
Table 3.2. Seawater and air terminal temperatures (pinch-based model).	47
Table 3.3. Seawater and air terminal temperatures (terminal-based model).....	47
Table 3.4. Comparison between Elmutasim et al. [68] and the current work (pinch-based model).....	48
Table 3.5. Direct performance and sizing table for the design procedure.	59
Table 3.6. Monthly weather forecast and climate Dhahran, Saudi Arabia [69].....	61
Table 4.1. Main operational variables basic values and variations.....	68
Table 5.1. Suggested arrangement of high and low effectiveness segments.	111
Table 5.2. Suggested arrangement of high and low Terminal-based effectiveness segments.....	112

LIST OF FIGURES

Figure 1.1. Different salinities of water resources [11].	4
Figure 1.2. Projection of globe populace, 1900 – 2100 [12].	7
Figure 1.3. Categorization of desalination techniques [15].	11
Figure 1.4. The process of freshwater desalination [17].	12
Figure 3.1. CAOW W-H humidification dehumidification cycle [56].	29
Figure 3.2. T–h diagram of the CAOW-WH HDH cycle [61].	31
Figure 3.3. Pinch point on T – h diagram and temperature cross.	37
Figure 3.4. Temperature derivative vs. humid air enthalpy based on [63] and current temperature derivative of Equation (3.23).	40
Figure 3.5. Temperature derivative vs. humid air enthalpy; (a) ln - ln scale; (b) ($\sqrt{\ln - \ln}$) scale [63].	42
Figure 3.6. Temperature distribution for thermodynamically-balanced CAOW HDH cycle; $T_{w,0} = 20\text{ }^{\circ}\text{C}$; $T_{w,2} = 80\text{ }^{\circ}\text{C}$, $\varepsilon_{\text{deh}} = 0.9$; (a) $\varepsilon_{\text{hum,Tb}} = 0.94$; (b) $\varepsilon_{\text{hum,Pb}} = 0.94$; (c) $\varepsilon_{\text{hum,Tb}} = 1.00$; (d) $\varepsilon_{\text{hum,Pb}} = 1.00$	49
Figure 3.7. Temperature distribution for unbalanced ($\text{MR} < \text{MR}_{\text{opt}}$) CAOW HDH cycle; $T_{w,0} = 20\text{ }^{\circ}\text{C}$; $T_{w,2} = 80\text{ }^{\circ}\text{C}$, $\varepsilon_{\text{deh}} = 0.9$; (a) $\varepsilon_{\text{hum,Tb}} = 0.94$; (b) $\varepsilon_{\text{hum,Pb}} = 0.94$; (c) $\varepsilon_{\text{hum,Tb}} = 1.00$; (d) $\varepsilon_{\text{hum,Pb}} = 1.00$	50
Figure 3.8. Temperature distribution for unbalanced ($\text{MR} > \text{MR}_{\text{opt}}$) CAOW HDH cycle; $T_{w,0} = 20\text{ }^{\circ}\text{C}$; $T_{w,2} = 80\text{ }^{\circ}\text{C}$, $\varepsilon_{\text{deh}} = 0.9$; (a) $\varepsilon_{\text{hum,Tb}} = 0.94$; (b) $\varepsilon_{\text{hum,Pb}} = 0.94$; (c) $\varepsilon_{\text{hum,Tb}} = 1.00$; (d) $\varepsilon_{\text{hum,Pb}} = 1.00$	51
Figure 3.9. Normalized temperature distribution for thermodynamically-balanced CAOW HDH cycle; $\varepsilon_{\text{deh}} = 0.9$; (a) $\varepsilon_{\text{hum,Tb}} = 0.94$; (b) $\varepsilon_{\text{hum,Pb}} = 0.94$; (c) $\varepsilon_{\text{hum,Tb}} = 1.00$; (d) $\varepsilon_{\text{hum,Pb}} = 1.00$	53
Figure 3.10. Impact of mass flowrate ratio of seawater to air on the GOR of the CAOW humidification dehumidification cycle; $T_{w,0} = 20\text{ }^{\circ}\text{C}$; $T_{w,2} = 80\text{ }^{\circ}\text{C}$, $\varepsilon_{\text{deh}} = 0.9$; (a) $\varepsilon_{\text{deh}} = \varepsilon_{\text{hum,Tb}} = \varepsilon_{\text{hum,Pb}} = 1.00$; (b) $\varepsilon_{\text{deh}} = \varepsilon_{\text{hum,Tb}} = \varepsilon_{\text{hum,Pb}} = 0.90$	54

Figure 3.11. Effect of humidifier height (H) on the gain output ratio and humidifier effectiveness; $T_{w,0} = 20\text{ }^{\circ}\text{C}$; $T_{w,2} = 80\text{ }^{\circ}\text{C}$	56
Figure 3.12. Effect of dehumidifier area (A) on the gain output ratio and dehumidifier effectiveness; $T_{w,0} = 20\text{ }^{\circ}\text{C}$; $T_{w,2} = 80\text{ }^{\circ}\text{C}$	56
Figure 3.13. Effect of dehumidifier/humidifier pinch-based energy effectiveness on the gain output ratio and BTC; $T_{w,0} = 20\text{ }^{\circ}\text{C}$; $T_{w,2} = 80\text{ }^{\circ}\text{C}$	58
Figure 4.1. The relation between specific enthalpy and saturated air temperature pressure of 1 atm [40].	71
Figure 4.2. The relation between saturated air natural logarithm of specific enthalpy and temperature at a pressure of 1 atm [40].	73
Figure 4.3. Absolute humidity ratio vs. specific enthalpy for saturated air.	82
Figure 4.4. Analytical method algorithm flowchart for the balanced CAOW water-heated HDH system.	85
Figure 4.5. Analytical method algorithm flowchart for the unbalanced CAOW water-heated HDH system (Case 1).	86
Figure 4.6. Analytical method algorithm flowchart for the unbalanced CAOW water-heated HDH system (Case 2).	87
Figure 4.7. Impact of mass flowrate ratio and humidifier effectiveness; $T_{w,2} = 80\text{ }^{\circ}\text{C}$; $T_{w,0} = 30\text{ }^{\circ}\text{C}$; $\epsilon_{\text{deh}} = 80\%$; (a) $\epsilon_{\text{hum}} = 60\%$; (b) $\epsilon_{\text{hum}} = 80\%$; (c) $\epsilon_{\text{hum}} = 100\%$. 89	
Figure 4.8. Impact of mass flowrate ratio and dehumidifier effectiveness; $T_{w,2} = 80\text{ }^{\circ}\text{C}$; $T_{w,0} = 30\text{ }^{\circ}\text{C}$; $\epsilon_{\text{hum}} = 80\%$; (a) $\epsilon_{\text{deh}} = 60\%$; (b) $\epsilon_{\text{deh}} = 80\%$; (c) $\epsilon_{\text{deh}} = 100\%$	90
Figure 4.9. Impact of mass flowrate ratio and top water temperature; $T_{w,0} = 30\text{ }^{\circ}\text{C}$; $\epsilon_{\text{hum}} = \epsilon_{\text{deh}} = 92\%$; (a) $T_{w,2} = 60\text{ }^{\circ}\text{C}$; (b) $T_{w,2} = 70\text{ }^{\circ}\text{C}$; (c) $T_{w,2} = 80\text{ }^{\circ}\text{C}$	92
Figure 4.10. Impact of mass flowrate ratio and bottom water temperature; $T_{w,2} = 80\text{ }^{\circ}\text{C}$; $\epsilon_{\text{hum}} = \epsilon_{\text{deh}} = 92\%$; (a) $T_{w,0} = 30\text{ }^{\circ}\text{C}$; (b) $T_{w,0} = 35\text{ }^{\circ}\text{C}$; (c) $T_{w,0} = 40\text{ }^{\circ}\text{C}$	94
Figure 5.1. Water heated, CAOW-WH humidification dehumidification configuration under thermodynamical balancing (single extraction) [56].	99

Figure 5.2. Local energy effectiveness in HDH's dehumidifier with and without applying extraction.....	102
Figure 5.3. Impact of applying extraction in reducing the generated entropy.	102
Figure 5.4. A CV comprising the seawater heater, part of the humidifier, and the equivalent part of the dehumidifier [40].	104
Figure 5.5. Zero extraction algorithm flowchart.	105
Figure 5.6. T - h chart of CAOW – WH HDH desalination cycle after applying the injection and extraction technique [56].....	106
Figure 5.7. Single extraction algorithm flowchart.	107
Figure 5.8. Impact of energy effectiveness on GOR.....	109
Figure 5.9. Impact of energy effectiveness on extraction level location.	110
Figure 5.10. Impact of energy effectiveness on GOR.....	112
Figure 5.11. Impact of the top seawater temperature; $\varepsilon_{deh} = \varepsilon_{hum} = 100\%$	113
Figure 5.12. Impact of the top seawater temperature; $\varepsilon_{deh} = \varepsilon_{hum} = 97\%$	115
Figure 5.13. Top seawater temperature impact; $\varepsilon_{deh} = \varepsilon_{hum} = 91\%$	116
Figure 5.14. Impact of the top seawater temperature; $\varepsilon_{deh} = \varepsilon_{hum} = 85\%$	117

LIST OF ABBREVIATIONS

BTC	:	Benefit to cost ratio
CAOW	:	Closed-air open-water
GOR	:	Gained Output Ratio
HCR	:	Heat Capacity Ratio
HDH	:	Humidification Dehumidification
HME	:	Heat and mass exchange
MED	:	Multi-effect Distillation
MSF	:	Multi-Stage Flash
RO	:	Reverse Osmosis
RR	:	Recovery Ratio
WH	:	Water heated

ABSTRACT

Full Name : [ABDULAZIZ ABDULLAH ALSAIF]
Thesis Title : [AN INNOVATIVE AND PRACTICAL DEFINITION OF ENERGY EFFECTIVENESS FOR HUMIDIFICATION DEHUMIDIFICATION DESALINATION SYSTEMS]
Major Field : [MECHANICAL ENGINEERING]
Date of Degree : [December 2020]

Humidification dehumidification (HDH) cycles are reliable, simple, and easy to be maintained. In this thesis, the closed-air open-water (CAOW) HDH cycle is investigated. The energy and mass balance equations are effectively used in the humidification dehumidification cycle utilizing an implicit/explicit model that is founded on charted/correlated properties of saturated air specific enthalpy and temperature. The current study proposes a compact, analytical, and novel model that facilitates using the humidification dehumidification model. Based on these correlations, the humidification dehumidification system is modeled analytically. This explicit model is utilized for getting a better understanding of the impact of operating conditions on the performance of the closed-air open-water water-heated HDH system. Pinch-based energy effectiveness is introduced for the analysis of the humidifier and dehumidifier as a replacement to the terminal-based energy effectiveness, enthalpy pinch, and temperature pinch. Energy effectiveness combines the impact of both temperature and concentration (salinity and humidity of seawater and air, respectively). An assessment of the evaluated results based on the proposed method and the literature is presented and good agreement is noted

between the current model and theoretical data with a minor error of $\pm 2.29\%$. Also, a direct design procedure is proposed in this research in order to size the closed-air open-water water-heated humidification dehumidification desalination system. The city of Dhahran is taken as a case study. The system selected for this location is designed to produce 5 kg hr^{-1} of freshwater at average minimum and maximum temperature of seawater assumed 25 and 70°C , respectively. It is found that to run the system optimally around the year, energy effectiveness should be at average of 78%.

Thermodynamic balancing is a powerful technique to minimize the entropy generation in any thermal system. This technique is used to improve the CAOW HDH system performance by extracting the recirculated air from the humidifier to the dehumidifier. The objective of the current thesis is to study the performance of dehumidifier and humidifier with/without extraction using the proposed energy effectiveness definition analytically. It is found that the gained output ratio of the dehumidification humidification desalination system with and without extraction increases as energy effectiveness rises. It is important to emphasize that applying extraction and injection technique improves the system performance after a certain value of energy effectiveness. Below this point, the system performance after applying injection and extraction overlaps with that of zero extraction i.e. injection and extraction technique has zero impact at low energy effectiveness values. The gained output ratio of the dehumidification humidification desalination system with/without extraction rises as top seawater temperature increases to a certain level where it is optimum and then it decreases after it. At $\varepsilon = 97\%$ of the humidifier and dehumidifier, the performance of the humidification dehumidification system performance after applying extraction and injection technique is significantly enhanced by 83.1%.

ملخص الرسالة

الاسم الكامل: عبد العزيز عبد الله السيف

عنوان الرسالة: تعريف عملي مبتكر لكفاءة الطاقة في أنظمة تحلية المياه المعتمدة على الترطيب والتجفيف

التخصص: الهندسة الميكانيكية

تاريخ الدرجة العلمية: ديسمبر، 2020

أنظمة تحلية المياه المعتمدة على الترطيب والتجفيف تعتبر قوية، عالية الاعتمادية، بسيطة، ومعروفة لتحمل مدى واسع للمياه المالحة دون الحاجة إلى صيانة معقدة. في هذه الرسالة البحثية، يتم فحص ودراسة نظام تحلية المياه ذو دورة المياه المفتوحة ودورة الهواء المغلقة لكونه من أفضل أنظمة الترطيب والتجفيف. تم استخدام معادلات توازن الطاقة والكتلة بنجاح لنظام تحلية المياه المعتمد على الترطيب والتجفيف باستخدام نموذج ضمني/صريح يعتمد على الخصائص المجدولة/المرسومة بيانياً لدرجة الحرارة والمحتوى الحراري النوعي للهواء المشبع ببخار الماء بنسب 100%. يقدم هذا البحث نموذجاً متماسكاً، واضحاً، مباشراً، ومبتكراً يسهل استخدامه لدراسة النظام تحت الدراسة. أيضاً، بناءً على العلاقة بين درجة الحرارة والمحتوى الحراري النوعي للهواء المشبع، تم نمذجة النظام بشكل تحليلي. أستخدم هذا النموذج التحليلي الصريح للحصول على فهم عميق لتأثير ظروف التشغيل على أداء نظام تحلية المياه تحت الدراسة. تعريف الطاقة المقترح تم تقديمه لتحليل كلاً من المرطب وجهاز إزالة الرطوبة كبديل لتعريف الطاقة السابقة، فرق المحتوى الحراري النوعي، و فرق درجة الحرارة. من المهم توضيح أن تعريف الطاقة المقترح يشمل تأثير كلاً من درجة الحرارة والتركيز (درجة الملوحة والرطوبة للماء والهواء، على التوالي). تم تقييم النتائج المحسوبة بناءً على الطريقة المقترحة مع البحوث السابقة المنشورة وتمت ملاحظة مطابقة ممتازة بين النموذج الحالي والبيانات النظرية بخطأ $\pm 2.29\%$. أيضاً، تم تقديم إجراء تصميمي مباشر في هذا البحث من أجل تحديد حجم المرطب

والمجفف في نظام تحلية المياه ذو دورة المياه المفتوحة ودورة الهواء المغلقة مع تسخين المياه. مدينة الظهران تم إتخاذها كحالة دراسية مع فرض درجة الحرارة الصغرى والكبرى لماء البحر في الجهاز 25° و 70° درجة مئوية، على التوالي. تم تصميم نظام التحلية بالشكل الأمثل بسعة إنتاجية قدرها 5 كجم/ساعة من الماء القابل للشرب ووجد أن كفاءة الطاقة المثلى هي 78%.

الموازنة الديناميكية الحرارية هي تقنية قوية لتقليل توليد الإنتروبيا في أي نظام حراري. تم استخدام هذه التقنية من أجل تحسين أداء النظام عن طريق إستخراج الهواء المعاد تدويره من المرطب إلى جهاز إزالة الرطوبة. الهدف من هذا البحث هو التحقيق عددياً وتحليلياً في أداء جهاز إزالة الرطوبة والمرطب مع أو بدون تقنية الحقن والاستخراج باستخدام تعريف كفاءة الطاقة المقترح. وُجد أن أداء (نسبة الإخراج المكتسب) نظام التحلية بالترطيب والتجفيف مع أو بدون تقنية الحقن والاستخراج يزيد بزيادة كفاءة الطاقة. من المهم توضيح أن استخدام الموازنة الحرارية مع تطبيق تقنية الحقن والإستخراج تحسّن من أداء النظام فقط اذا كانت كفاءة الطاقة في المرطب والمجفف أعلى من حد معين وهذا الحد يعتمد على ظروف التشغيل. بمعنى آخر، تشغيل النظام مع تقنية الحقن والإستخراج بكفاءة أقل من هذا الحد عند نفس ظروف التشغيل سيؤدي لأداء مماثل لنفس النظام من غير تقنية الحقن والإستخراج أي أن تقنية الحقن والإستخراج لها تأثير صفري عند قيم كفاءات الطاقة المنخفضة نسبياً. تمت أيضاً دراسة تأثير درجة الحرارة الكبرى على أداء (نسبة الإخراج المكتسب) نظام التحلية بالترطيب والتجفيف مع/بدون استخدام الموازنة الحرارية بتقنية الحقن والإستخراج وُجد أن العلاقة بينهما طرية حتى مستوى معيّن من درجة الحرارة الكبرى يكون عنده أداء النظام أمثل ثم يبدأ بعدها بالإنخفاض كلما زادت درجة الحرارة الكبرى. عند كفاءة طاقة 97% لكل من المرطب والمجفف، أداء نظام التحلية بالترطيب والتجفيف بعد تطبيق الموازنة الحرارية بالحقن والإستخراج يتحسن بشكل كبير بنسبة 83.1% أكثر من النظام التقليدي.

CHAPTER 1

INTRODUCTION

Kingdom of Saudi Arabia is located between latitudes 16° and 33° N, and longitudes 34° and 56° E. In Kingdom regions, there is an acute lack of freshwater resources. Therefore, freshwater desalination is a necessity in the Kingdom with a major share in the Kingdom's energy consumption and due to population increase, it will keep rising in the future and it could be doubled by 2030 [1]. In this manner, saltwater distillation strategies in combination with preservation endeavors are required in regions wherever quantities of natural assets of drinking (consumable) water are lessening [2]. Saudi Arabia is considered a major producer of desalinated water with a vast share in the Middle East where more than 15% of the total national energy is consumed in freshwater production [3].

Desalination can be characterized into electrical, thermal, and mechanical procedures, for example, multi-effect desalination (MED) is frequently utilized as a thermal desalination technique [4]. These techniques are extremely successful for desalinating water. Though, it cannot be introduced as a unit of small-scale since their maintenance expenses are moderately high. A different alternative to desalinate water might be used in situations when capital and running/maintenance assets are inadequate. The humidification

dehumidification cycle is ecologically friendly as well as favorable when it comes to smaller-scale uses.

From physics, the air is used as a bearer to desalinate seawater utilizing the cycle of rainwater, in which seawater temperature is raised by sun radiations and vaporizes in the atmospheric air. At that point, the moist air rises, and the structure cloud mists. Then, each cloud condenses in the form of rain that can be utilized later. The humidification dehumidification (HDH) system can be considered as a descendant for sun oriented still (solar still). The humidification dehumidification cycle mimics the cycle of rain. This system has gotten steady consideration recently where a couple of investigators have examined some configurations of these HDH systems [5].

HDH cycles can be characterized into two general categories [6]. To start with, based on the type of energy utilized, for example, sun oriented, geothermal, and thermal, etc. The second category relies upon how HDH is designed. Based on the clear indication in the name [7] running in a continuous cycle, dehumidified and humidified as opposite to CWOA cycle [8] where the temperature of the air is raised by adding heat to it.

1.1 Freshwater sources

Freshwater is very important to mankind. It has different chemical properties, which play an essential role in all organic life. A long time ago, water was the strategic source to build the nations and growth of civilizations. Therefore, nations lived nearby lakes, waterfalls, and rivers. At present time, many countries are wealthy in water sources whereas others are suffering from water shortage or pollution. Water is the most used chemical compound for many applications in industries, agriculture growth, and human usage. Availing adequate drinking water for many countries is a real problem such that many poor countries are forced to utilize unhealthy water for drinking and their daily use, which might lead to several related diseases and infections. Some countries are rich with adequate drinking water either from natural sources or through industrial processes (desalination plants) that have helped to achieve rapid growth in population and the standard of living in the last decades.

A large portion of the earth's water (97%) is seawater in seas and oceans and around 2% in ice caps and glaciers. The rest of that is available underground, in rivers and lakes. However, nature is cycling the freshwater from the solar distillation of the seawater to the grounds and again to oceans (natural water cycle). The process occurs such that the sun heats and evaporates the surface water of the seas forming water vapor that is generated and raised into the upper layer of the atmosphere. Then it is condensing to form clouds and again returns to the earth in the form of rain and snow which flows back to the oceans [9].

1.2 Freshwater properties

Water has universal solvent properties. Dissolved chemical materials to such an extent that it is not suitable for human consumption. Also, water sources in nature are exposed to pollutants from industries and domestic areas. According to the Water Health Organization (WHO), suitable water for drinking is determined based on the concentration of the dissolved substances. The WHO standard for the maximum dissolved salt in drinking water is 500 ppm with the limit of chloride ions at 250 ppm. However, groundwater with salinity from 3000 to 4000 ppm in some dry areas might be used for drinking [10].

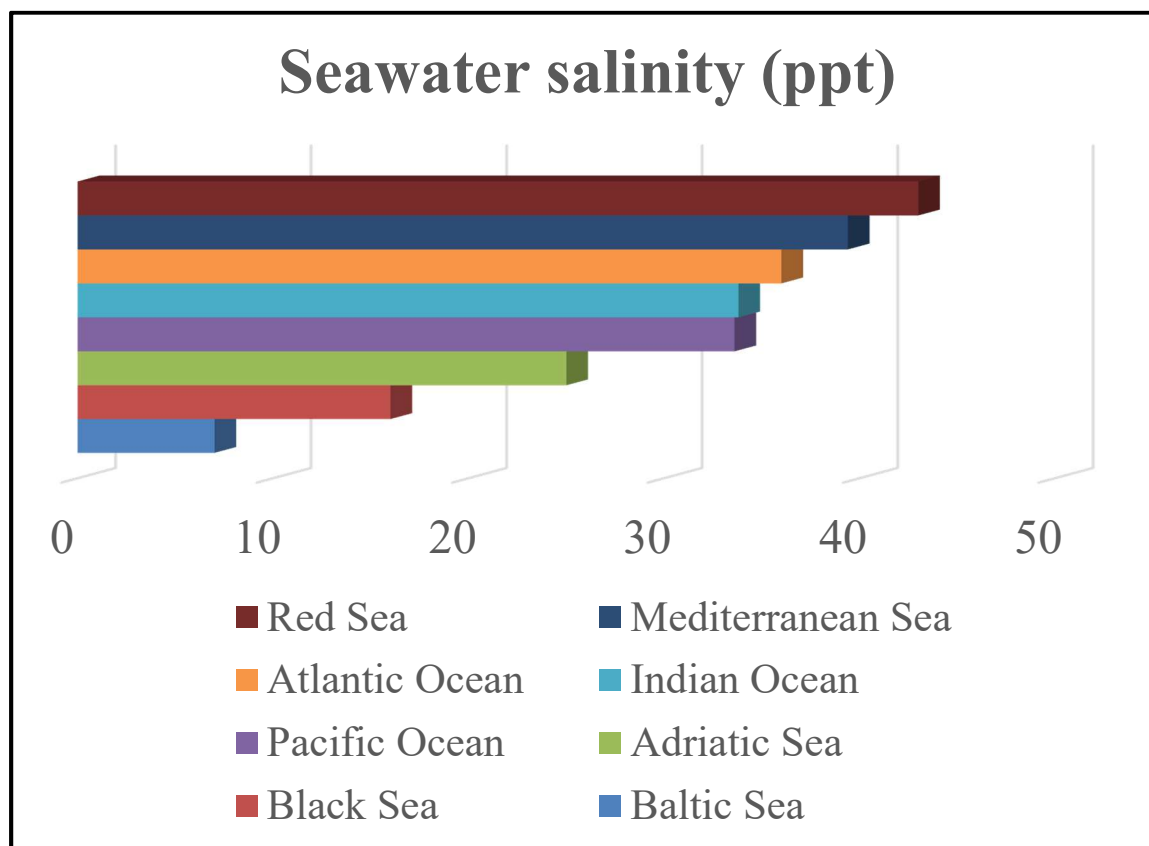


Figure 1.1. Different salinities of water resources [11].

The seawater in many parts of the world is the same in water chemistry. The salinity factor is a characteristic of seawater that determines the demand for treatment. For the analysis of the total concentration of dissolved salts, which varies from one location to another, the lowest value is 7000 at the Baltic Sea whereas the maximum value is recorded at the Red sea with 43000 ppm as shown in Fig. 1.1 [11] .

1.3 The problem of freshwater scarcity

The rainfall and water ground storage quantity is adequate for the demand of the world population but unfortunately, the distribution of rainfall is not uniform in different areas around the globe. Particularly in the MENA region, which has no natural water sources in some areas like Gulf countries and a low quantity of rainfall over the seasons of the year. Another important issue is the growth of the world population where the statistical data shows rapid growth over the past fifty years as shown in Fig 1.2. The trend also predicts that by 2025, the world population would be 7.5 billion. This growth is more pronounced in water scarcity areas. Water shortage is growing by many other factors. For instance, the raising of the standard of living in many cities all over the world, the rapid expansion of industries, the agriculture and the demand of many food products, lack of clean water and variation of culture and water reclamation awareness.

After oil exploration in Middle East countries, the oil sales and income growth were reflected in the standard of living and the large-scale development of the water engineering solutions to fulfill the huge water demand in the industry sector and supply more above what the local population had consumed. The development of water research also discovers the needs of some countries, which have plenty of available water but are not uniformly distributed to focus on the solution for water demands in the areas that experience water shortages. However, the rapid agriculture expansion in many countries is based on water availability. Some organizations are prohibiting random groundwater exploration and enhance irrigation processes to maintain the storage of groundwater mainly in the dry areas. Moreover, water shortages are considered political issues, especially in the countries that share one natural water source such as lakes, rivers, or springs.

The international memorandum of understanding and agreements are governing the share of each country besides the essential agreement from all water shareholders for acceptance of the other water applications like building a power plant, scientific military experiments, or new channel branching, especially for rivers and lakes.

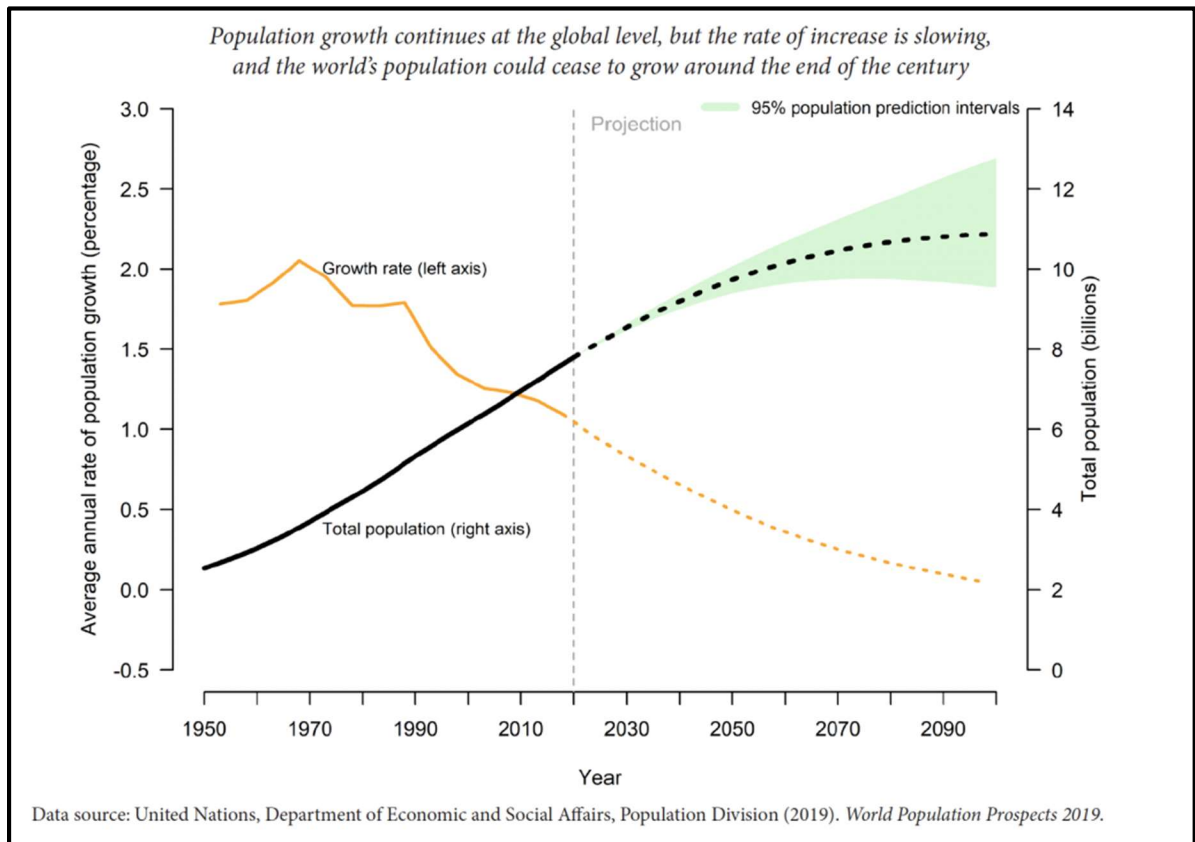


Figure 1.2. Projection of globe populace, 1900 – 2100 [12].

1.4 Resolution of freshwater scarcity

Economic policy, in many countries, is strongly dependent on water security and insurance of water storage and transportation. Water security is also considered as the top priority for countries where water might be threatened by many factors such as oil leakage, political war, chemical toxic material, etc.

Many alternatives solutions to water problems are being popularly used like controlling water consumption, proactive water distribution, reclamation, purification and water reuse, strategic water storage...etc. When all of the preceding solutions of water control are not working effectively, seawater desalination demand is addressed and considered to be implemented.

Many countries especially in the Mena region have built desalination plants to encounter the expansion in the consumption of freshwater. The aim of constructing these desalination plants is to produce freshwater with low salinity to be suitable for human consumption e.g less than 500 ppm. However, the major problem related to desalination plants is the high capital cost for the plants and the associated systems. From a comparison point for the cost of the freshwater and desalinated water, it's still too high due to operation cost as well. Despite all the previous research and development of new desalination technologies, the cost of one meter cubic of desalinated water is still higher than the freshwater from natural sources which require a minimum treatment process to make it suitable for human consumption [13].

1.5 Desalination of freshwater

Seawater is an aqueous solution, of which 95.5 ~ 96.5 percent is water and the remaining 4.5 ~ 3.5 percent consists of various types of dissolved minerals such as salts. Seawater desalination means the removal of freshwater from seawater. The method used for this process is called seawater (saline water) desalination or saline water conversion.

To remove freshwater from seawater, processes like distillation, crystallization, reverse osmosis, solvent extraction, electro dialysis, and ion exchange resin, are well known. The distillation process and the crystallization process separate the water content in seawater by making use of a phase change of water.

While the reverse osmosis and the electro dialysis processes separate the water content employing pressure difference and potential difference by making use of specially made membranes (Reverse osmosis membranes and ion exchange membranes), which are known generally as membrane processes.

1.5.1 Historical background of freshwater desalination

200 years ago, the first distillation method was used on trade ships. The sailors used to provide drinking water while the ships are sailing in the sea. Later on, many desalination systems were developed and commercialized. However, that was only based on the thermal distillation principle where the evaporation of seawater takes place then the generated vapor condenses to form freshwater. Gradually, land-based plants came into service. Historically, the first water desalination plant was reported installed in Aden before 100 years ago. In the early 1930s, oil exploration in the Middle East was started and some working companies built several small desalination systems to support the oil industries.

Unfortunately, these plants were not too efficient and had a high cost of operation and maintenance. In the early 1950s, Arabian Gulf countries were realized with the development of oil industries and sales leading to constructing the large capacity of a desalination plant. Therefore, they pushed the desalination industry [14].

1.5.2 Categorization of freshwater desalination techniques

The water desalination principle means removing freshwater from saline water. This is called seawater desalination. However, the terminology desalination is applied for salty water with salinity above 1000 ppm. The technique used to desalinate the water depends on the process that occurs to discretize the salt out of freshwater such that distillation, crystallization process, mechanical filtration, or what is commonly known by reverse osmosis process, electro dialysis, ion exchange, etc. Physically the desalination process depends on the driving force for the separation like the evaporation process and condensation in the distillation systems or in a different way like high pressure on an osmotic membrane in the reverse osmosis process or what is commonly known by (Membrane process).

The desalination field has a variety of desalination processes. Some of them can be applied for large scale while the other is for low scale or even laboratory scale (refer to Fig 1.3). The classification shown in the figure is based on the energy used where there are four types of energy used for desalination namely, electrical, thermal, mechanical, and chemical.

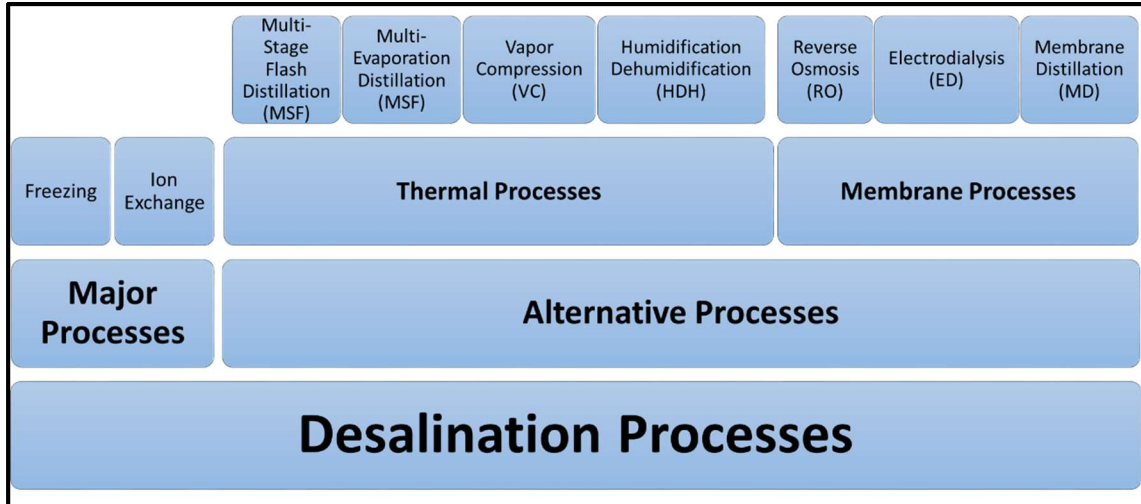


Figure 1.3. Categorization of desalination techniques [15].

From a design viewpoint, the optimal selection of desalination technology is based on feasibility studies for many important factors before implementation. These studies should include seawater chemistry, operation period, production capacity, location, and operation and maintenance experience.

1.5.3 Evolution of freshwater desalination techniques

The desalination field has delivered many thoughts and methods that we can use to desalinate saline water, but only a few of them were able to be commercialized and operated with a large capacity. Over the past years, many desalination units were built with good experiences of operation and maintenance especially thermal desalination work by HDH (humidification dehumidification), MED (multiple effect distillation), or by RO membrane desalination plant (reverse osmosis).

The majority of freshwater produced by the distillation process, which is presented in many countries are desalination plants that heat the seawater to be condensed as freshwater as illustrated in Fig.1.4 [16].

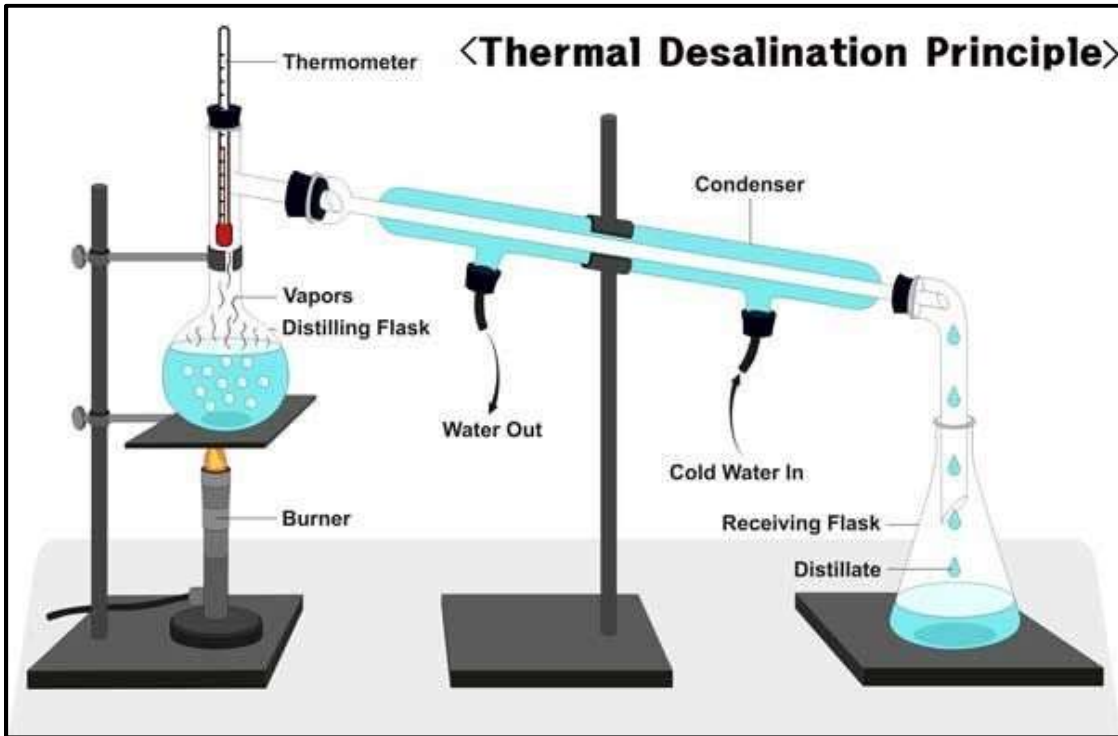


Figure 1.4. The process of freshwater desalination [17].

In this thesis, an innovative definition of energy effectiveness is presented for the HDH cycle. In the proposed systems the downsides of the old definition are taken care of. These downsides are second law violation in some cases and/or temperature cross in the humidifier in other extreme cases which defeats the humidification purpose.

1.6 The innovative part of this research

This thesis aims to model the CAOW HDH system by using the proposed definition of energy effectiveness. In this research, the CAOW HDH cycle performance is studied based on the proposed energy effectiveness definition.

This research may encourage other researchers utilizing the proposed definition to be used widely in modeling all HDH system as well as heat and mass exchangers in other thermal applications. In this research, the system of interest is studied theoretically to get rid of the old energy effectiveness definition downsides. The proposed definition has not been carried out before and the results are expected to improve the modeling and understanding of the HDH cycles.

1.7 Objectives of the research

The end objective of the proposed research is to establish a clear and robust definition of the proposed energy effectiveness for the main components (humidifier and dehumidifier) of the HDH cycle. In this research, the extraction technique is studied (thermodynamic balancing using the proposed energy effectiveness as the performance index) between the dehumidifier and the humidifier to increase the system performance. The specific objectives of the current research are ordered in sequence and can be summarized as follows:

1. An updated literature review is conducted to examine the state of the performance parameters associated with the HDH system. Performance evaluation studies for different types of HDH configurations in the literature and the best type is selected to apply the proposed method.
2. Steady-state mathematical formulation and investigation of simultaneous transfer of mass and heat processes for both humidifier and dehumidifier using thermodynamic conservation laws plus the new definition for energy effectiveness as the performance index.

In this regard, a computer program is written to simulate the system using both MATLAB® and EES® software.

3. Solving the humidification dehumidification model with and without extraction (using the new definition of energy effectiveness as the performance index) by using the iterative method to solve the model.
4. The zero-extraction model is solved analytically (using energy effectiveness as the performance index). Also, the infinity extraction system is modeled analytically.

5. Parametric study for the impact of operating conditions is carried out so that the critical operating variables are evaluated and highlighted. Also, results and findings are presented, interpreted, and reported.

CHAPTER 2

LITERATURE REVIEW

2.1 Main desalination techniques

Water desalination technology like RO with advanced membranes is getting widely expanded over the past decay by 44% of desalination water contribution, especially in the Middle East. This is due to ease of operation, which directly utilizes electrical power to run the high-pressure pump [18].

Unlike RO technology, the thermal-based plants are not independent in operation style. They require a steam power plant that operates using either oil or natural gas as a source of energy, which is highly un-economical [19].

Recognized thermal desalination technologies (HDH, MED, etc) still exist in the Gulf countries, the source of fresh water where no other natural resources can meet the demand for water required.

The oil exploration in Gulf countries led to vast industrial expansion, especially in water desalination. Unfortunately, the nature of Gulf water chemistry is very poor and can be characterized by high salinity (more than 45000 ppm) and it is geographically having some shallow area and high turbidity. Moreover, the area has high differences in temperature from a season to another in addition to high marine growth life [20].

Saudi Arabia is the largest producer of desalinated water in the world recorded daily production of 5.2 million m³. Nearly 70 % of this production is from thermal-based plants where the rest of production is by RO (Reverse Osmosis) plants and with lower production contribution of MED (Multi-effect distillation) plants [21].

Table 2.1. Comparison between MED and HDH desalination techniques [22].

	Multi-Effect Distillation (MED)	Humidification Dehumidification (HDH)
Operational Temperature (⁰ C)	< 110	60 - 90
Energy source	Steam	Steam
	Electric Power	Electric Power
Product quality (ppt)		< .025
Scale Control	Acid cleaning	Simple maintenance
Chemical Consumption	\$42,000/yr/MIGD	None

Table 2.1 that was prepared by Doosan Company in 2012, shows a comparison between thermal-based (MED for example) and RO systems in terms of energy resources, specific power consumption, capacity, etc. [22].

Moreover, the operation of thermal-based desalination technology is more reliable and sustainable since many problems were experienced and solved successfully over the past decades. Recently, many researchers are focusing on power consumption and performance optimization where the MED technology is showing good features in this by utilizing a portion of generated vapor and again recirculating it by mixing with the

motive steam [23], due to many operations constraints that need to be solved; like the limitation of top brine temperature $<66\text{ }^{\circ}\text{C}$, as well as critical operation conditions [23].

2.2 Humidification dehumidification systems

HDH desalination systems are comprised of two main components namely humidifier and dehumidifier, which can be driven by a variety of heat sources (Figure 1). HDH process functions based on the natural desalination cycle (rain cycle) in which the seawater is heated by solar energy. Clouds are formed by evaporation of the seawater to the atmosphere and raining occurs by dehumidification of the air at higher sea levels. The man-made version of this cycle is termed as HDH process. HDH process benefits from the moisture carrying capability of air in which air absorbs the moisture from saline water in the humidifier and rejects it as freshwater in the dehumidifier. To explain more, saline water in the humidifier is brought into contact with air and diffuses to it due to the concentration difference between the water vapor in the air and water vapor interface. This results in an increase in the moisture content of the saturated air and salinity level of the brine exiting the humidifier.

In the dehumidifier, the saturated air preheats the inlet saline water by condensing to the freshwater. Generally, three different configurations of HDH systems are defined as closed-air open-water (CAOW), open-air open-water (OAOW), and open-air closed-water (OACW) cycles. Moreover, the other classification of HDH systems is based on the heating of each stream, water-heated, or air-heated. Therefore, considerable efforts have been made to enhance the performance of each component for various configurations of HDH systems powered by different heat sources. A comprehensive study comparing the performance of the aforementioned cycles was conducted in [24]. It was illustrated that the CAOW water heated cycle exhibited good performance compared to other

arrangements not to mention the arrangement that allows applying thermodynamic balancing technique on it. It is worth mentioning that if the air heater is placed after the humidifier, higher GOR can be obtained compared to the condition that the heater is located before the humidifier [25].

A huge number of investigations were also performed in terms of improvement of HDH processes and therefore enhancing the performance of the cycle. A summary of the main technical results of HDH systems in terms of GOR, yield, and cost of freshwater is reported. GOR is defined as the ratio of the total latent heat of evaporation of the produced fresh water to the input energy to the desalination system. This parameter is a common index representing the amount of heat recovery in the desalination systems and is calculated as in Eq. (1). It can be said that different configurations of HDH cycles have been investigated from both theoretical and experimental viewpoints.

A variety of humidifiers have been employed in which the packed-bed humidifiers were the most commonly used ones. In these humidifiers, different packing materials have been used to enhance the effectiveness of the humidification process. Regarding the dehumidifiers, direct contact types, in which the air is dehumidified by spraying cold freshwater into the hot and humid air stream, exhibited better performance compared to conventional types. Furthermore, it can be concluded that utilizing the multistage and bubble column humidifiers and dehumidifiers enhanced the system performance. To benefit from the capability of HDH cycles in coupling with low-grade heat sources, geothermal, biomass energy, different solar technologies as flat plate collectors, evacuated tube collectors, and photovoltaic panels, etc. have been examined. Also, several researchers have focused on HDH cycles powered by external waste heat sources.

2.3 Evolution of performance parameters in humidification dehumidification systems

Table 2.2 presents the evolution of humidifier/dehumidifier performance parameters in the closed air open water seawater heated (CAOW - WH) HDH system. Narayan et al. [26] discussed different types of energy effectiveness definitions and concluded that the most utilized one is the terminal-based. However, this definition had two main issues: a high possibility for temperature cross in the humidifier, especially at high energy effectiveness. It also failed in modeling an HDH system with multiple extractions. To solve this problem, McGovern et al. [27] proposed using temperature pinch as an alternative performance parameter to model a thermodynamically-balanced HDH system with multiple extractions and injections. It is important to highlight that temperature pinch has a high possibility for temperature cross and does not take mass transfer into account. To eliminate these downsides, a new index called enthalpy pinch was proposed [28]. However, a significant drawback is associated with this parameter. This is because it is not normalized, a fixed numerical value of enthalpy pinch (e.g., $10 \text{ kJ kg}_{\text{da}}^{-1}$) has a different meaning at various operating conditions.

Table 2.2. Evolution in the definition of performance parameters for humidification dehumidification desalination systems.

A Terminal-based energy effectiveness (2010)

B Temperature pinch (2013)

C Enthalpy pinch (2013)

D Pinch-based energy effectiveness (current research)

Advantage	Performance Parameter			
	A	B	C	D
	[26]	[27]	[28]	
Takes account of both heat and mass transfer processes	✓	✗	✓	✓
Have more sense (0 – 100%) and familiarity with it	✓	✗	✗	✓
Successful in modeling thermodynamic balancing	✗	✓	✓	✓
Zero possibility for Temperature cross	✗	✗	✓	✓
Normalized w.r.t operating conditions	✓	✗	✗	✓
Model balanced and unbalanced systems	✓	✓	✗	✓

2.4 Thermodynamic balancing in humidification dehumidification systems

Irrespective of the previously mentioned advantages, the humidification dehumidification cycle has a low performance which leads to more freshwater cost. This is because of the generated entropy in instantaneous mass and heat exchange devices (HME) e.g. the humidifier and dehumidifier are significant. Thermal balance for the mass/heat exchange devices in the HDH cycle enhances energy efficiency [27] by minimizing the difference in enthalpy of exchanging streams namely, seawater and recirculated air which consequently reduces the total rate of entropy generation [29].

Using the injection and extraction method takes full benefit of the humidification dehumidification cycle by rising the cycle performance to the maximum probable limit. Alongside HDH seawater purification cycles, thermal balance is also utilized to improve the coefficient of performance of an innovative air-cooling system [30]. There are other methods [31]–[36] that have been used to improve the performance of HDH desalination systems, but they have not increased the performance as promised in the method of extraction and injection.

Wu and his colleagues [31] conducted an experimental investigation of an innovative 3-stage humidification dehumidification cycle where the humidifier and the solar seawater heater are one whole component. They maximized the production of freshwater to more than 3 kg h^{-1} and with an optimum gained output ratio of slightly more than 2. In a different try to improve the performance of the cycle, He et al. [32] recovered energy from the waste heat to drive the HDH system and they achieved a 2.5 GOR at design conditions.

The hybridization of HDH with other cycles [33], [34], [36] is utilized as an approach to improve the cycle performance. Qasem and Zubair [33] hybridized a conventional HDH cycle with an adsorption desalination cycle for air conditioning and freshwater production with a maximum GOR of 7.8. Pairing conventional [34] and absorption-compression [36] heat pump cycles with HDH systems showed potential in enhancing the performance of the cycle.

In this research, the focus is on the CAOW water-heated cycle as it is considered one of the most cycles in which thermodynamic balancing is applied comprehensively and has been reviewed thoroughly in the literature [28], [37]–[40]. Narayan et al. [28] suggested thermal balance utilizing injection and extraction as a probable method for reducing energy consumption by decreasing the generated entropy in humidification dehumidification cycles. Chehayeb et al. [37] improved the research in [28] by using extra points for extraction and injection to enhance the humidification dehumidification cycles performance to higher levels of GOR. After this study, Chehayeb and his colleagues [38] investigated the impact of changing energy effectiveness with operational variables at a fixed area of heat and mass transfer on applying the thermal balancing method. Qasem et al. [39] utilized a thermal balance method using multiple extraction and injection points on an innovative humidification dehumidification cycle that uses the desiccant solutions as an alternative to seawater with a 100% enhancement in the cycle performance by using one extraction. Significant to be noted that open knowledge is numerously focusing on the conventional humidification dehumidification frameworks to estimate gained-output-ratio (GOR) utilizing thermodynamical balancing (extraction) using enthalpy and temperature pinch methods as the performance index.

Using energy effectiveness was, however, not investigated to the points of the thermodynamic balancing approach due to some divergence issues. Therefore, one of the primary objectives of the current research is investigating thermodynamical balancing (extraction) using the proposed energy effectiveness definition.

In the current research, a novel model based on numerical techniques is utilized to solve the thermodynamically-balanced humidification dehumidification cycles due to the high level of nonlinearity. The proposed numerical technique is expected to make the model more robust while facilitates using simultaneous equation solvers, for example, EES® to resolve the model. The generated methods are expected to be direct, compact, and precise; hence, these methods could be used by designers and researchers to improve the humidification dehumidification systems performance.

2.5 Design optimization of cost-effective HDH system

Humidification-dehumidification technology has been proven to be a cost-effective approach to fulfill the freshwater demand. In the review reported in [41], the impact of hybridization of HDH cycles with refrigeration, power, and other desalination technologies in reducing the freshwater production cost is elaborated. Abbasi and Pourrahmania [42] suggested a system in which the HDH is powered by a Kalina cycle and thermoelectric generator; then two different configurations are compared by levelized cost with fossil fuel power plants. Xu et al. [43] studied the cost reduction in compressed-based HDH due to performance enhancement. Dhahiru et al. [34], [44], [45] conducted both numerical and experimental investigations on the heat pump-driven HDH systems performance. They [44] concluded that this hybridization is economically competitive compared to other HDH arrangements. To reduce the operational cost of the HDH system, brine recirculation is used by Ghofrani, and Moosavi [46] as a method of heat recovery to improve the system performance.

All studies [41]–[44], [46] used performance enhancement methods such as hybridization or heat recovery to reduce only the operational cost. Unlike the aforementioned studies, the current study end objective is to optimize the operational conditions by maximizing the system performance and minimizing the system size (capital cost) in the design process.

CHAPTER 3

PINCH-BASED ENERGY EFFECTIVENESS DEFINITION

Summary

In this chapter, the closed-air open-water water-heated humidification dehumidification system is examined. This chapter proposes an innovative and robust model that facilitates a practical utilization of the humidification dehumidification model. This study aims to investigate the performance of the dehumidifier and humidifier using the proposed energy effectiveness that proves unrestricted by the downsides of the terminal-based definition. These downsides violate the second law in some cases with a high possibility of temperature cross in the humidifier which defeats the heating/humidification purpose. This study may encourage other researchers to utilize the proposed definition widely in modeling all humidification dehumidification systems. The proposed definition is verified with the literature and showed an excellent agreement with a maximum discrepancy of 2.29 %. The objective of this research is to optimize the operational conditions by maximizing the system performance and minimizing the system size. Afterward, a direct and simple scheme is provided to estimate dehumidifier and humidifier sizes based on the location and amount of freshwater needed. A case study for a system that produces 5 kg hr^{-1} of freshwater is presented. The optimal energy effectiveness is found to be 78% in the city of Dhahran where the average minimum and maximum temperature of seawater in the cycle are assumed 25 and 70 °C, respectively.

3.1 Introduction

It is worth mentioning that for a long time, systems of solar distillation have been used before humidification dehumidification (HDH) techniques, however, they performed poorly with a limited and discrete freshwater production [47]. HDH cycles are considered as an enhanced form of solar distillation systems that operate using low-grade energy sources. The process of the HDH cycle is attained by humidifying the circulated air through the humidification device to the saturation point. Then, saturated air is condensed to obtain freshwater [48]. It is to be noted that HDH is a mimic cycle of the natural rainwater cycle [49].

In this chapter, an innovative definition of energy effectiveness is presented for the HDH cycle. This definition eliminates the downsides of the terminal-based definition that lead to whether an occurrence of the temperature cross in the humidifier or violation of the second law in other extreme cases. The proposed pinch-based energy effectiveness is used as an appropriate alternative to temperature pinch [50]–[52] and enthalpy pinch [53]–[55] in modeling HDH systems with both balanced and unbalanced conditions.

3.2 Description of the system under study

To demonstrate the proposed definition of pinch-based energy effectiveness, the CAOW-WH HDH system is used. This system comprises a humidifier, dehumidifier, and saltwater heater [56]. Figure 3.1 shows that seawater ‘coolant’ (0) flows in the dehumidifier in this arrangement. Afterward, it is pre-warmed (1), then, is heated to state 2 utilizing the seawater heater. Heated seawater (2) is sprayed above the packing material in the humidifier. The loop of circulated air is close through both the dehumidifier and humidifier, as depicted in Fig. 3.1. The circulated air stream flows into the humidifier and leaves as hot and humid air. Then, air (a,2) is dehumidified to produce freshwater (fw). The dehumidified air is directed back to the humidifier to close the air loop.

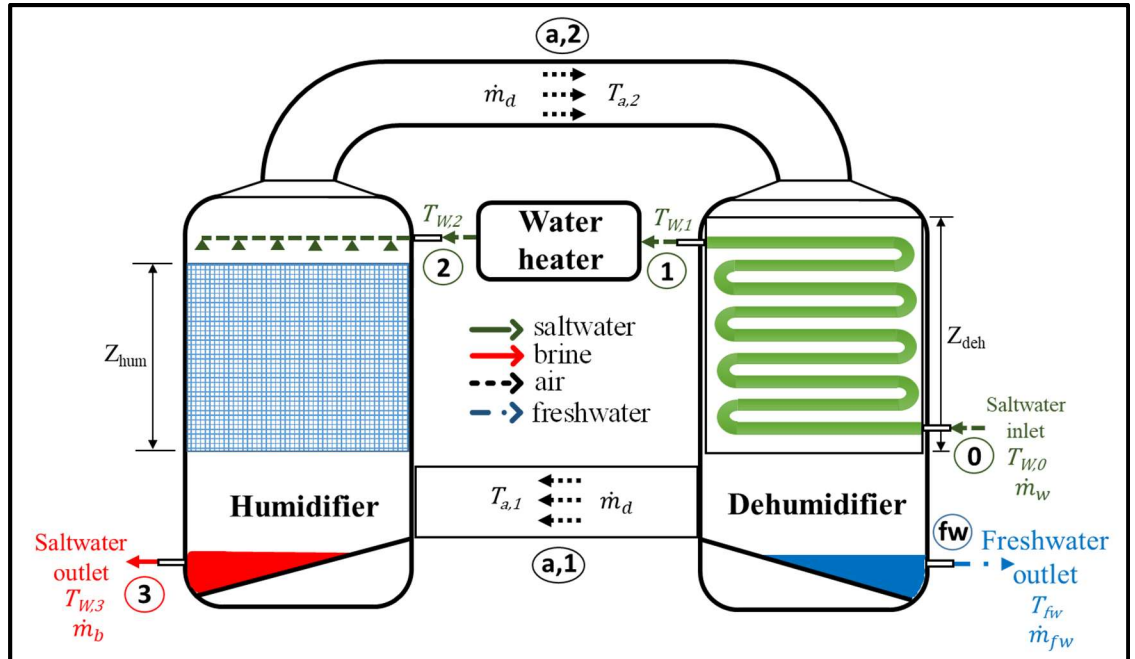


Figure 3.1. CAOW W-H humidification dehumidification cycle [56].

3.3 Temperature-enthalpy (T - h) diagram

The concept of energy effectiveness is well-established in the literature. Still, it is also intricate because its definition is based on both maximum and minimum enthalpies of the recirculated air between the humidifier and dehumidifier. Due to this complexity, the temperature pinch was used as an alternative performance parameter [27], [57], analogous to heat exchangers. However, this performance parameter considers only sensible heat exchange between water and air streams. Thus, an alternative performance index was introduced; called enthalpy pinch [58], which considers both mass and heat transfer between air and water in the dehumidifier and humidifier.

Based on the concept of enthalpy pinch, a temperature-enthalpy (T - h) diagram was developed [59], [60] as a graphical method to model HDH systems. The temperature-enthalpy chart for the HDH cycle is depicted in Fig. 3.2. This diagram represents the temperature change for water and air streams through the humidifier and dehumidifier. It is worth noting that the x-axis in these diagrams (refer to Fig. 3.2) is specific enthalpy (h^*) of recirculated dry air in preference to the height (Z) of the humidifier and dehumidifier. The assumption here is that normalizing specific enthalpy based on dry air basis is used as an equivalent to the height of both towers [28].

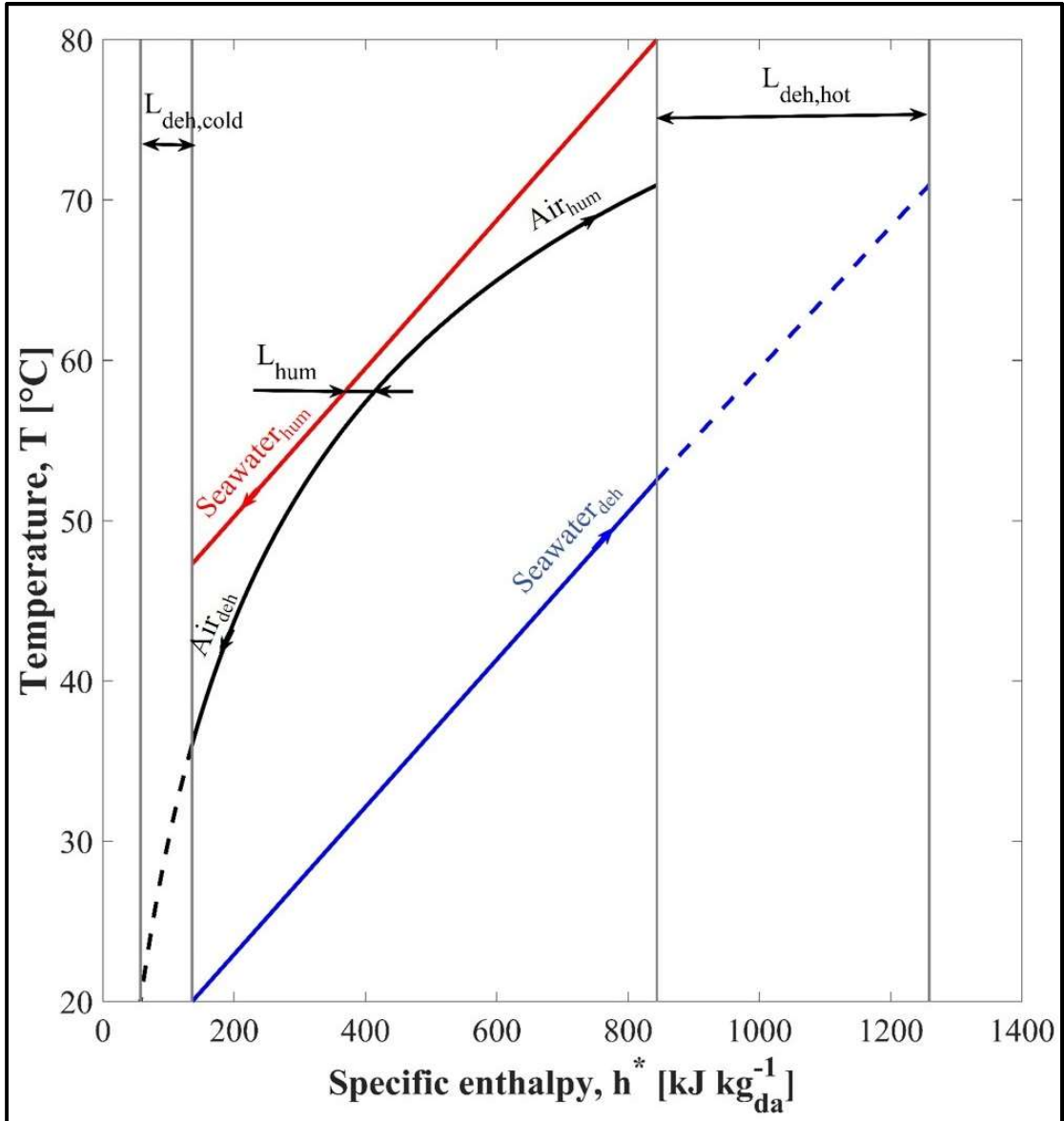


Figure 3.2. T-h diagram of the CAOW-WH HDH cycle [61].

3.4 Humidification dehumidification model

In this section, the approach of energy effectiveness is introduced for evaluating the CAOW-WH HDH systems under balanced and unbalanced operating conditions. Therefore, the following assumptions are implemented.

- All processes are steady-state.
- Compared to the heater's required energy, the saltwater pump and air blower's power is insignificant.
- Losses of heat and mass from the humidifier, dehumidifier, and heater to surrounding ambiances are neglected.
- The impact of potential and kinetic energies are neglected.
- The recirculated air is saturated [62].
- Freshwater exits the dehumidification side with the average temperature of the humid air.

3.4.1 Mass and energy conservation

All equations related to the transfer of mass and heat through the components of the humidification dehumidification arrangement are expressed as follow,

3.4.1.1 Humidifier

$$\dot{m}_w - \dot{m}_{da} (\omega_{a,2} - \omega_{a,1}) = \dot{m}_b \quad (3.1)$$

$$\dot{m}_w h_2 - \dot{m}_{da} (h_{a,2} - h_{a,1}) = \dot{m}_b h_3 \quad (3.2)$$

3.4.1.2 Dehumidifier

$$\dot{m}_{fw} = \dot{m}_{da} (\omega_{a,2} - \omega_{a,1}) \quad (3.3)$$

$$\dot{m}_w (h_1 - h_0) + \dot{m}_{fw} h_{fw} = \dot{m}_{da} (h_{a,2} - h_{a,1}) \quad (3.4)$$

3.4.1.3 Seawater heater

$$\dot{Q}_{ht} = \dot{m}_w (h_2 - h_1) \quad (3.5)$$

Equations (3.1 – 3.5) can't be solved because the number of variables is more than the number of equations by two. Therefore, dehumidifier and humidifier energy effectiveness are defined to close the equations set to solve them.

3.4.2 Terminal-based energy effectiveness definition

In all heat/mass exchange devices, the change in concentration (salinity and humidity ratio for water and air, respectively) and temperature are the driving force for energy transfer to take place. Energy effectiveness can be defined based on the ideal probable enthalpy change at terminal points, which is attained in mass and heat exchange devices with energy effectiveness of 100%. The maximum saltwater temperature at the dehumidification device outlet which can be obtained is the inlet temperature of the recirculated air inlet temperature as stated in the following equation,

$$\Delta \dot{H}_{w,max} = \dot{m}_w (h_{1,id} - h_0) \quad (3.6)$$

The required energy to achieve $\Delta \dot{H}_{w,max}$ will be drawn from the air side, which might not provide sufficient energy amount. In case of air, the side does not provide the needed energy, this ideal change in enthalpy will cool the recirculated air to saltwater inlet (bottom) temperature as expressed in the following equation,

$$\Delta \dot{H}_{da, \max} = \dot{m}_{da} (h_{a,2} - h_{a,1,id}) \quad (3.7)$$

Similar to heat exchange devices, heat/mass exchange devices effectiveness is defined as the proportion of actual to probable ideal enthalpy change as expressed in the following equation,

$$\varepsilon_{deh} = \frac{\Delta \dot{H}}{\Delta \dot{H}_{\max}} \quad (3.8)$$

Based on the value of $\Delta \dot{H}_{\max}$ for both water and air sides, Eq. (3.8) can be rewritten as follows,

Caes (1): if $\Delta \dot{H}_{da, \max} < \Delta \dot{H}_{w, \max}$

$$\varepsilon_{deh} = \frac{h_{a,2} - h_{a,1}}{h_{a,2} - h_{a,1,id}} \quad (3.9)$$

Case (2): if $\Delta \dot{H}_{da, \max} > \Delta \dot{H}_{w, \max}$

$$\varepsilon_{deh} = \frac{h_1 - h_0}{h_{1,id} - h_0} \quad (3.10)$$

Equations (3.9) and (3.10) can be merged as in Eq. (3.11),

$$\varepsilon_{deh} = \max \left\langle \frac{h_1 - h_0}{h_{1,id} - h_0}, \frac{h_{a,2} - h_{a,1}}{h_{a,2} - h_{a,1,id}} \right\rangle \quad (3.11)$$

In addition, the effectiveness of a humidifier can be defined using the aforementioned investigation. However, it is incorrect for the humidifier because the previous definition is based on the assumption that losses due to the limit in heat and mass transfer area are at both terminal points. Actually, this is correct for the dehumidifier all the time and incorrect for the humidifier mostly, since the losses in the humidifier occur at an intermediate point (pinch point, refer to L_{hum} in Fig. 3.2). This point can travel in some extreme cases to one of the terminal points. Therefore, the definition of the humidifier's energy effectiveness needs to be corrected based on the aforementioned facts (refer to section 3.3).

3.4.3 Proposed definition of humidifier effectiveness

The current study is utilizing the model that is based on the pinch-based (new) definition of energy effectiveness (as a suitable alternative to pinch change either enthalpy-based [28] or temperature-based [50]–[52]) for the humidifier and dehumidifier. It is to be noted that temperature pinch is a performance index usually used in heat exchangers due to its lack of mass transfer process. Then, enthalpy pinch is used as an alternative performance index, which considers both heat and mass transfer as a replacement for the former definition of energy effectiveness that failed in modeling thermodynamic balancing with multiple extractions. In brief, the authors believe that the pinch-based (new) definition is a better performance index for the humidifier in HDH systems. For a balanced humidifier, heat capacity ratio (HCR) = 1, the losses occur only in one intermediate point. This point (tan_{id}) is the closest point on the humidifier line to the air curve (see Fig. 3.2). State 'tan' is the tangent point on the air curve, as shown in Fig. 3.2, where state ' tan_{id} ' represents the maximum enthalpy that can be reached by air in a 100% pinch-based effectiveness humidifier.

Therefore, the relation between humidifier losses and effectiveness can be expressed as,

$$\mathcal{E}_{hum,Pb} = \frac{h_{a,2} - h_{a,1}}{h_{a,2} - h_{a,1} + LOSS_{hum,Pb}} \quad (3.12)$$

Equation (3.12) in its current form is insufficient to close the system of equations because it contains an extra unknown; that is pinch losses $L_{hum,Pb}$. To eliminate this misrepresentation, the terminal-based definition is used for guidance as shown in Eq. (3.13),

$$\mathcal{E}_{hum,Tb} = \frac{h_{a,2} - h_{a,1}}{h_{a,2,id} - h_{a,1}} \quad (3.13)$$

where,

$$h_{a,2,id} = \text{specific enthalpy} (@T = T_{w,2}) \quad (3.14)$$

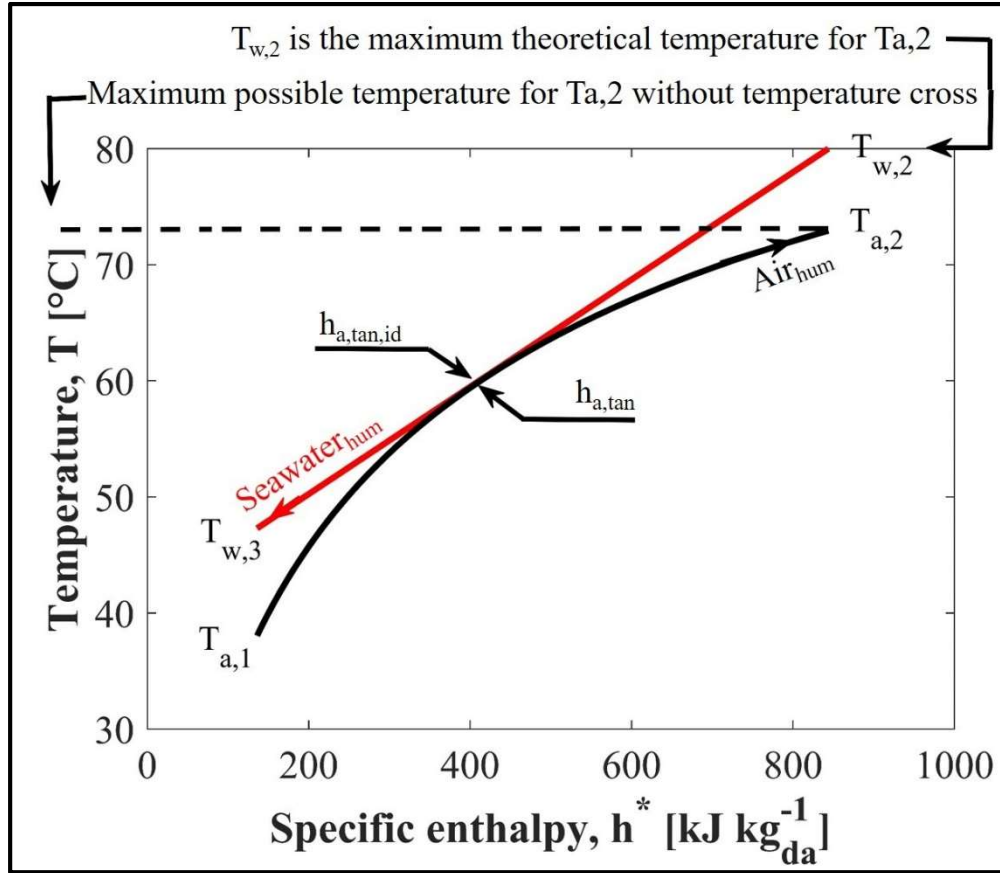


Figure 3.3. Pinch point on T – h diagram and temperature cross.

Figure 3.3 shows the temperature distribution in the humidifier. This figure is generated using a pinch-based effectiveness value equal to 100%. It is clear that there is an initiation of temperature cross between the air curve and water line at the pinch point. In the terminal (previous) definition, the air temperature at humidifier outlet $T_{a,2}$ can reach the water temperature $T_{w,2}$ at the humidifier inlet at 100% terminal effectiveness. While in the pinch-based definition, a different air temperature is defined as the new ceiling; it characterizes the maximum possible temperature that the outlet air can reach without the occurrence of temperature cross. To do so, the best statement can be defined by comparing both Eqs. (3.12) and (3.13) and the pinch-based effectiveness definition becomes,

$$\mathcal{E}_{hum,Pb} = \frac{h_{a,2} - h_{a,1}}{(h_{a,2} + L_{hum,Pb}) - h_{a,1}} = \frac{h_{a,2} - h_{a,1}}{(h_{a,2,id}) - h_{a,1}} \quad (3.15)$$

From Fig. 3.2, the humidifier energy losses at the pinch point can be expressed as,

$$L_{hum,Pb} = h_{a,tan} - h_{a,tan,id} \quad (3.16)$$

where the definition of pinch point (*tan*) is a point on the air curve at which the temperature derivative is equal to the water line slope. In other words the closest distance between the air curve and water line. This can be written as,

$$h_{a,tan} = \text{specific enthalpy} \left(@ \frac{dT}{dh} = slp_{hum} \right) = \text{specific enthalpy} \left(@ \frac{dT}{dh} = \frac{T_{w,2} - T_{w,3}}{h_{a,2} - h_{a,1}} \right) \quad (3.17)$$

Similarly, the temperature at the pinch point can be found using the psychrometric chart as follows,

$$T_{a,tan,id} = T_{a,tan} = \text{Temperature} \left(@ \frac{dT}{dh} = \frac{T_{w,2} - T_{w,3}}{h_{a,2} - h_{a,1}} \right) \quad (3.18)$$

From Fig. 3.2, using a single point ($h_{a,2}$, $T_{w,2}$) on the humidifier water line with the

$slp_{hum} = \frac{T_{w,2} - T_{w,3}}{h_{a,2} - h_{a,1}}$, specific enthalpy along the humidifier can be stated as a function of the

local temperature, which can be written as,

$$h = h_{a,2} - \left(\frac{h_{a,2} - h_{a,1}}{T_2 - T_3} \right) (T_{w,2} - T) \quad (3.19)$$

Applying the ideal pinch temperature ($T_{a,tan,id}$) in Eq. (3.19), $h_{a,tan,id}$ can be expressed as follows,

$$h_{a,tan,id} = h_{a,2} - \left(\frac{h_{a,2} - h_{a,1}}{T_2 - T_3} \right) \left(T_{w,2} - \text{Temperature} \left(@ \frac{dT}{dh} = \frac{T_{w,2} - T_{w,3}}{h_{a,2} - h_{a,1}} \right) \right) \quad (3.20)$$

3.4.4 Approximation of saturated air thermodynamic properties

Air thermo-physical properties are significant towards modeling the thermodynamically-balanced HDH systems. Namely, these properties are saturated air temperature, specific enthalpy, and temperature derivative with respect to specific enthalpy. These three parameters can be linked to the relation between saturated air specific enthalpy and temperature. Because of the importance of this correlation in modeling cycles, Ahmed et al. [40] simplified this relation to reduce the model complexity. Saturated air specific enthalpy at atmospheric pressure is stated as follows [40],

$$h = \left(\frac{\zeta_1}{\zeta_2 - T} - \zeta_3 \right)^{\zeta_4} \quad (3.21)$$

where,

$$\zeta_1 = 4920.1593$$

$$\zeta_2 = 104.4580$$

$$\zeta_3 = 41.9112$$

$$\zeta_4 = 1.4473$$

From Eq. (3.21), the saturated air temperature and its derivative with respect to specific enthalpy can be expressed in Eqs. (3.22) and (3.23), respectively,

$$T = \zeta_2 - \frac{\zeta_1}{\zeta_3 + h^{1/\zeta_4}} \quad (3.22)$$

$$\frac{dT}{dh} = \frac{\zeta_1}{\zeta_4} \frac{h^{-1+1/\zeta_4}}{\left(\zeta_3 + h^{1/\zeta_4} \right)^2} \quad (3.23)$$

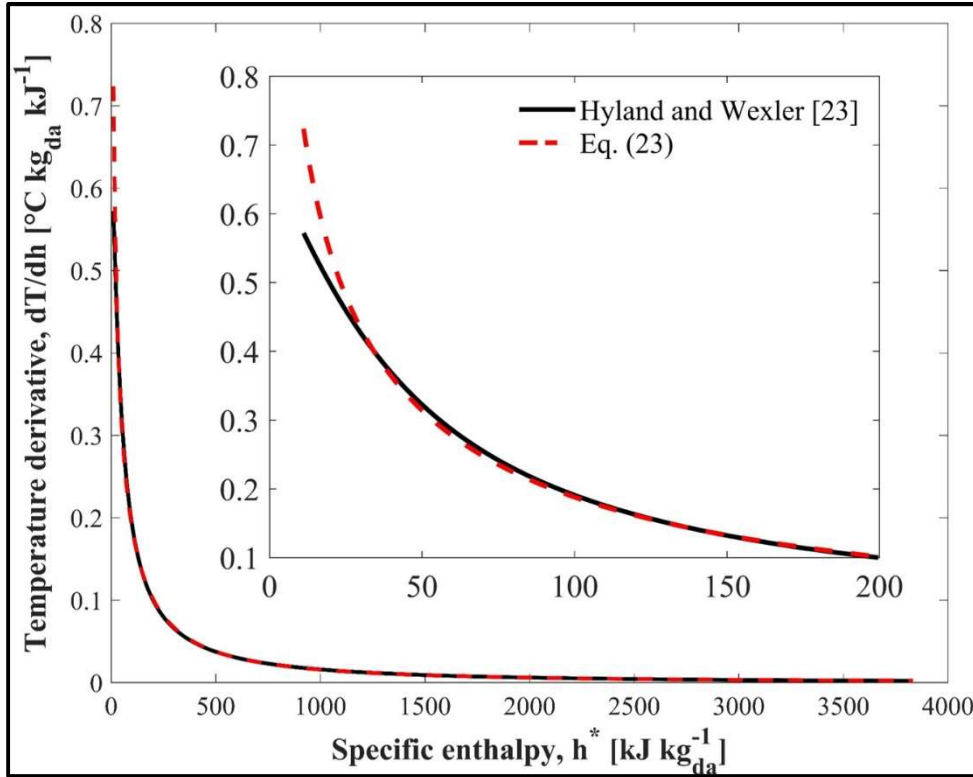


Figure 3.4. Temperature derivative vs. humid air enthalpy based on [63] and current temperature derivative of Equation (3.23).

The problem with Eq. (3.23) is that it is complex and difficult to be rearranged. For instance, to obtain 'h' as an explicit function of 'dT/dh', Eq. (3.23) cannot be used. Also, it is to be noted that Eq. (3.23) is not a good representation of the temperature derivative in the temperature range of 1 to 10 °C, as shown in Fig. 3.4. Therefore, further improvement is needed for this equation that covers the possible range of specific enthalpy (10 – 50000 kJ kg_{da}⁻¹ at atmospheric pressure) for the saturated air, which almost corresponds to a temperature range of 1 – 99 °C.

Figure 3.4 shows that as the specific enthalpy of saturated air increases, the temperature derivative decreases rapidly until a particular value of h (~ 500 kJ kg_{da}⁻¹). After this point, the temperature derivative decreases gradually as the specific enthalpy rises till it

approaches zero at $h = 2500 \text{ kJ kg}_{\text{da}}^{-1}$. Due to the extreme change of temperature derivative trend, this behavior as illustrated in Fig. 3.4 is noticeably difficult to formulate. However, some of the fitting approximations are used, such as polynomials up to the 3rd order, power, exponential, and logarithmic formulae, to form the temperature derivative and the specific enthalpy. Those approximations are not representative of the data. However, the logarithmic formula was the best; nevertheless, it represents the original data poorly. Therefore, Fig. 3.4 is re-demonstrated in a $\ln - \ln$ and $\sqrt{-\ln} - \ln$ forms for further investigation, as shown in Fig. 3.5 (a) and (b), respectively.

As presented in Fig. 3.5 (a), the $\ln - \ln$ relationship between temperature derivative 'dT/dh' and specific enthalpy 'h' is not linear. It has a behavior similar to the square root. Therefore, the natural logarithm of temperature derivative is compressed using the square root. It is to be noted that the 'minus sign' is added to avoid generating complex numbers, as shown in Fig. 3.5 (b). The behavior in Fig. 3.5 (b) is linear for air temperature range = 10 – 99 °C (which is wider than the expected temperature range of air in the system under study having water temperature range = 25 ~ 85 °C).

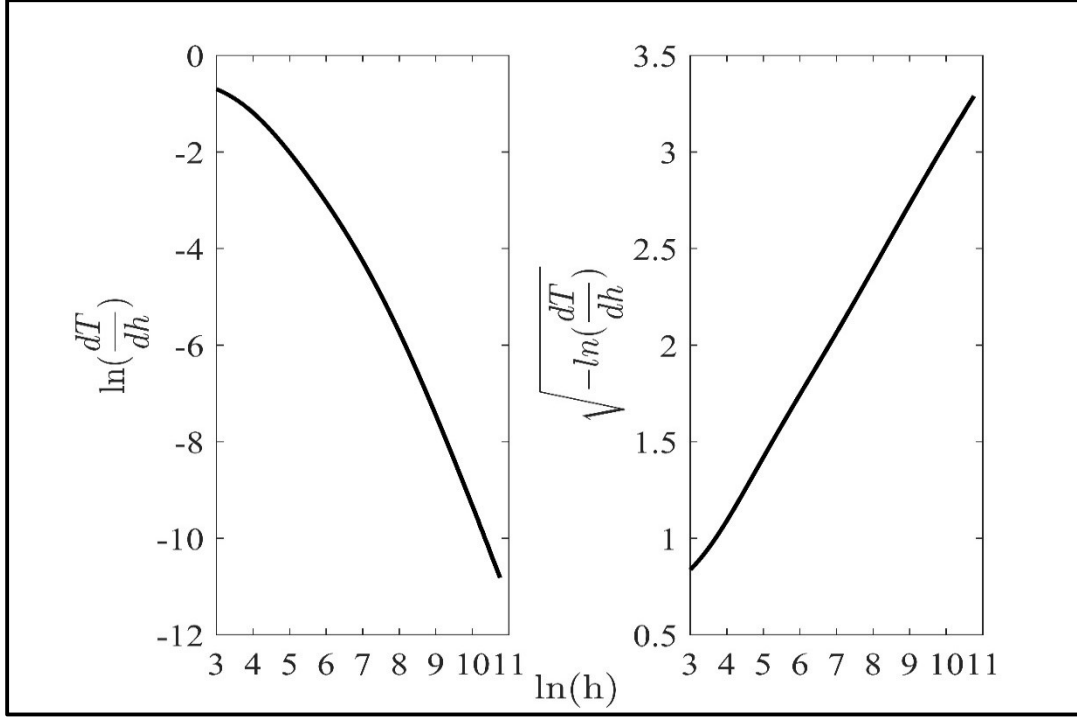


Figure 3.5. Temperature derivative vs. humid air enthalpy; (a) ln - ln scale; (b) ($\sqrt{-\ln - \ln}$) scale [63].

Using Fig. 3.5 (b) with the Hyland and Wexler [63] data, the relationship between the saturated air specific enthalpy and the temperature derivative can be expressed as,

$$h = \exp\left(\zeta_5 + \zeta_6 \sqrt{-\ln\left(\frac{dT}{dh}\right)}\right) \quad (3.24)$$

where,

$$\zeta_5 = 0.59033$$

$$\zeta_6 = 3.08635$$

Eq. (3.24) represents a $\ln - \sqrt{-\ln}$ approximation of the correlation between specific enthalpy and temperature derivative. This estimation is utilized in simplifying Eqs. (3.17) and (3.20) as,

$$h_{a,\tan} = h_{a,\tan} = \exp \left(\zeta_5 + \zeta_6 \sqrt{-\ln \left(\frac{T_{w,2} - T_{w,3}}{h_{a,2} - h_{a,1}} \right)} \right) \quad (3.25)$$

$$h_{a,\tan,id} = h_{a,2} - \left(\frac{h_{a,2} - h_{a,1}}{T_2 - T_3} \right) \left(T_{w,2} - \zeta_2 + \zeta_1 \left(\zeta_3 + \left(\exp \left(\zeta_5 + \zeta_6 \sqrt{-\ln \left(\frac{dT}{dh} \right)} \right) \right)^{\frac{1}{\zeta_4}} \right)^{-1} \right) \quad (3.26)$$

Using Eqs. (3.15), (3.16), (3.25), and (3.26), the air enthalpy at the maximum possible temperature that leaves the humidifier is stated as,

$$h_{a,2,id} = \exp \left(0.6 + 3.1 \sqrt{\ln \left(\frac{h_{a,2} - h_{a,1}}{T_{w,2} - T_{w,3}} \right)} \right) + \left(\frac{h_{a,2} - h_{a,1}}{T_{w,2} - T_{w,3}} \right) \left(T_2 + 43.4 - 57.3 \sqrt{\ln \left(\frac{h_{a,2} - h_{a,1}}{T_{w,2} - T_{w,3}} \right)} \right) \quad (3.27)$$

The final form for the proposed energy effectiveness for the humidifier can be stated as,

$$\mathcal{E}_{hum,Pb} = \frac{h_{a,2} - h_{a,1}}{\exp \left(0.6 + 3.1 \sqrt{\ln \left(\frac{h_{a,2} - h_{a,1}}{T_{w,2} - T_{w,3}} \right)} \right) + \left(\frac{h_{a,2} - h_{a,1}}{T_{w,2} - T_{w,3}} \right) \left(T_2 + 43.4 - 57.3 \sqrt{\ln \left(\frac{h_{a,2} - h_{a,1}}{T_{w,2} - T_{w,3}} \right)} \right) - h_{a,1}} \quad (3.28)$$

3.4.5 Humidifier/dehumidifier sizing

Merkel number can be stated [56] as shown in Eq. (3.29),

$$M e = \bar{C}_{p,w} \int_{T_{w,out}}^{T_{w,in}} \frac{dT_w}{\left(\zeta_1 (\zeta_2 - T_w)^{-1} - \zeta_3 \right)^{\zeta_4} - h_{a,in} - C_{p,w} M R (T_w - T_{w,out})} \quad (3.29)$$

Equation (3.29) integration is evaluated by the Simpson rule; it is applied to utilize a hundred intervals. The height of the humidifier depends on mass flow rate ratio and Merkel number [64],

$$Z = \zeta_7 MR^{\zeta_8} Me^{\zeta_9} \quad (3.30)$$

ζ_7 , ζ_8 , and ζ_9 are arbitrary coefficients that are a function of packing material type. Using the packing material (Brentwood manufacturer, CF1200MA), ζ_7 , ζ_8 , and ζ_9 are evaluated as 0.3214, 1.5823, and 1.2326, respectively [64].

Humidifier cross-section area is evaluated as seawater mass flowrate ($\dot{m}_w$) to the allowed flux (0.8 - 4.2 kg s⁻¹m⁻²) ratio. It is to be noted that humidifier recommended flux is around 1.5 kg s⁻¹m⁻² [65].

The surface area of the dehumidifier is evaluated utilizing the $\varepsilon - NTU$ routine. For a cross-flow dehumidifier that uses the Baclic method [66], the effectiveness of the dehumidifier is based on heat capacity ratio and number of transfer units (NTU) as,

$$\varepsilon_{deh} = 1 - e^{-(1+C)NTU} \left[I_0 (@ 2NTU \sqrt{C}) + \sqrt{C} I_1 (@ 2NTU \sqrt{C}) - \frac{1-C}{C} \sum_{i=2}^{\infty} C^{i/2} I_i (@ 2NTU \sqrt{C}) \right] \quad (3.31)$$

I_i is the Bessel function of ith-order and C is stated as follows,

$$C = \frac{\min[\dot{m}_w C_{p,w}, \dot{m}_d C_{p,a}]}{\max[\dot{m}_w C_{p,w}, \dot{m}_d C_{p,a}]} \quad (3.32)$$

Using the relation between NTU, C, overall heat transfer coefficient (U), the dehumidifier area can be stated as,

$$A_{deh} = \frac{NTU \min[\dot{m}_w C_{p,w}, \dot{m}_d C_{p,a}]}{U} \quad (3.33)$$

The overall heat transfer coefficient depends on the mass flow rate of the exchanging streams, the material type of the condensation tubes, and its fin geometry [67]. In this research, it is assumed to be fixed at $10 \text{ W m}^{-2} \text{ K}^{-1}$.

3.4.6 Performance matrix

To realize the analysis of the cycle under study, the following performance indices are used,

- i. Gained Output Ratio (GOR), a non-dimensional index represents the ratio of freshwater heat to the required energy to produce it.

$$GOR = \frac{\dot{m}_{fw} h_{fg}}{\dot{Q}} \quad (3.34)$$

- ii. Recovery ratio (RR), a non-dimensional variable represents the percentage of feedwater that transformed into freshwater.

$$RR = \frac{\dot{m}_{fw}}{\dot{m}_w} \quad (3.35)$$

- iii. Mass flowrate ratio (MR), a non-dimensional variable that is equivalent to the ratio of feedwater to recirculated dry air mass flowrate.

$$MR = \frac{\dot{m}_w}{\dot{m}_d} \quad (3.36)$$

3.5 Verification of the proposed method

This part is devoted to validating the proposed definition of humidifier energy effectiveness in the CAOW-WH HDH system.

3.5.1 Maximum allowable terminal-based energy effectiveness

The proposed method is novel, so it is difficult to validate it with the literature. Therefore, to verify the present model accuracy to use it with confidence, the CAOW-WH HDH system is tested under the extreme case, that is $\varepsilon_{hum,Pb} = 1$.

Table 3.1. Operational conditions basic values.

Operational variable	Typical value
The mass flow rate of freshwater production, $\dot{m}_w$ (kg hr^{-1})	10 (reference)
Seawater temperature, T_0 ($^{\circ}\text{C}$)	30
Seawater salinity, S_0 (ppt)	35
Seawater water maximum temperature, $T_{\max}$ or T_2 ($^{\circ}\text{C}$)	60
Dehumidifier energy effectiveness, ε_{deh}	0.9
Humidifier energy effectiveness, ε_{hum}	0.95

Considering the operating conditions in Table 3.1 with 100% humidifier pinch-based energy effectiveness, the water and air temperatures can be evaluated using Eqs. (3.1 – 3.5), (3.11), and (3.28), as shown in Table 3.2. Also, the maximum allowable terminal-based energy effectiveness is estimated using Eq. (3.30) to be $\varepsilon_{hum,Tb,\max} = 0.98$.

Table 3.2. Seawater and air terminal temperatures (pinch-based model).

Terminal points	Temperature (°C)
Seawater temperature at dehumidifier outlet ($T_{w,1}$)	56.54
Seawater temperature at humidifier outlet ($T_{w,3}$)	42.94
Bottom air temperature ($T_{a,1}$)	42.08
Top air temperature ($T_{a,2}$)	66.44

Using the value of 0.98 as humidifier terminal-based energy effectiveness with the typical values of Table 3.1 as well as the mass and energy balance equations (refer to Eqs. 3.1 – 3.5), the water and air temperatures can be evaluated as shown in Table 3.3. The deviation between the two methods is ranged between 0.20 to 1.92 %, which represents an excellent agreement. It is important to emphasize that this disagreement is due to the precision in calculating the maximum allowable terminal-based energy effectiveness.

Table 3.3. Seawater and air terminal temperatures (terminal-based model).

Terminal points	Temperature (°C)	Error (%)
Seawater temperature at dehumidifier outlet ($T_{w,1}$)	57.29	1.33
Seawater temperature at humidifier outlet ($T_{w,3}$)	42.15	1.84
Bottom air temperature ($T_{a,1}$)	41.27	1.92
Top air temperature ($T_{a,2}$)	66.57	0.20

3.5.2 Enthalpy pinch

For additional confidence in the proposed definition, the pinch-based model is validated with Elmutasim et al. [68], as shown in Table 3.4. The data taken from [68] is used to estimate the dehumidifier energy effectiveness and pinch-based effectiveness in the humidifier. Thereafter, the proposed model calculates air and water temperatures based on both the humidifier and dehumidifier effectiveness. The current model results show good agreement with the literature as the maximum deviation is 2.29%.

Table 3.4. Comparison between Elmutasim et al. [68] and the current work (pinch-based model).

Ref. [68]		Calculated using Ref. [68]		Current work			
Enthalpy	T_{w1}	T_{w3}	$\varepsilon_{hum,pb} = \varepsilon_{dch}$	T_{w1}	Error	T_{w3}	Error
pinch	(°C)	(°C)		(°C)	(%)	(°C)	(%)
(kJ kg ⁻¹ da)							
105	49.9	50.1	0.74	49.0	1.80	51.2	2.20
115	47.6	52.3	0.70	46.8	1.68	53.5	2.29
125	45.2	54.8	0.65	44.5	1.55	55.9	2.01
135	42.6	57.4	0.60	42.0	1.41	58.4	1.74
145	39.7	60.3	0.54	39.3	1.01	61.2	1.49
160	34.7	65.3	0.44	34.8	0.29	65.6	0.46

3.6 Comparison between terminal- and pinch-based energy effectiveness

Figures 3.6 – 3.8 show the temperature distribution for water and air streams in both the dehumidifier and humidifier in balanced (MR_{opt}) and unbalanced ($MR < \text{and} > MR_{opt}$) conditions, respectively. These estimations are conducted twice, once using the terminal- and then the pinch-based effectiveness model. In order to perform a reasonable comparison between both effectiveness models, bottom and top water temperatures are fixed at 20 and 80 °C, respectively, with the effectiveness of (a) 94% and (b) 100%.

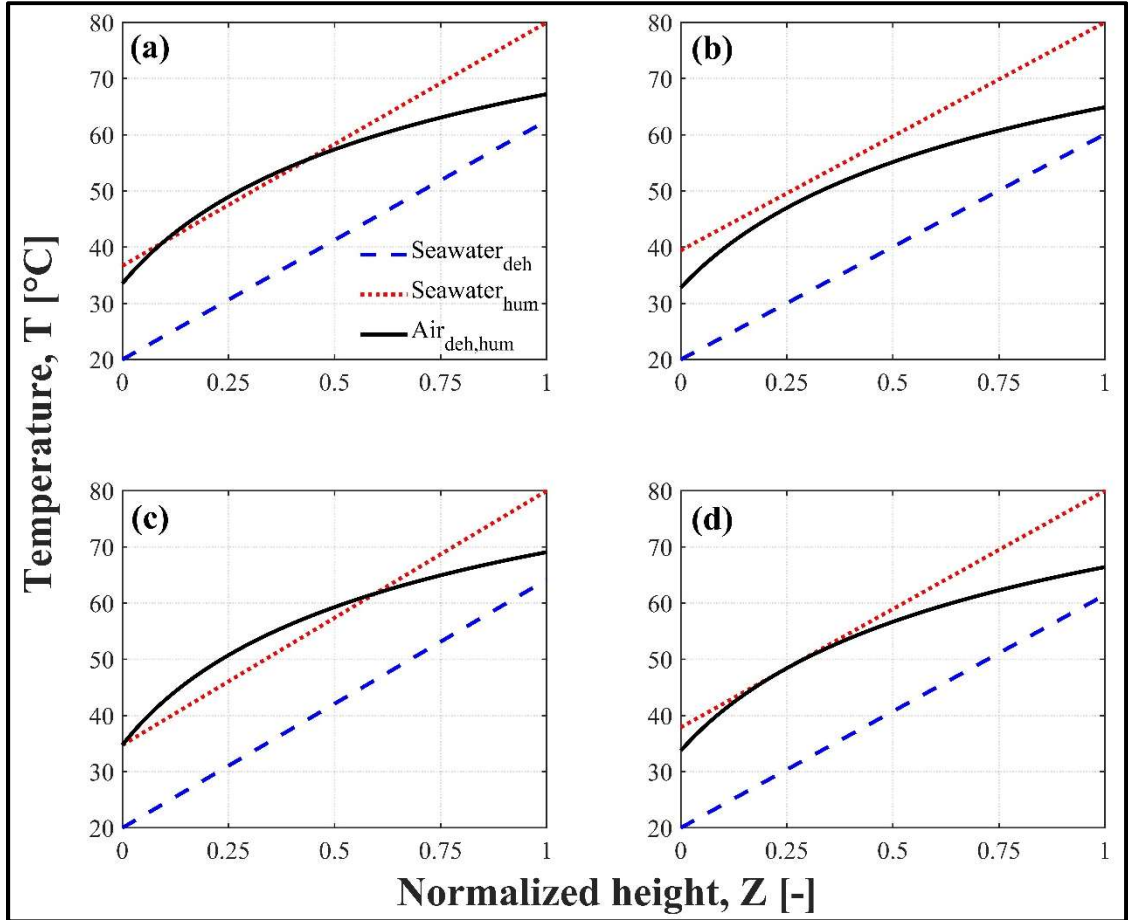


Figure 3.6. Temperature distribution for thermodynamically-balanced CAOW HDH cycle; $T_{w,0} = 20\text{ }^{\circ}\text{C}$; $T_{w,2} = 80\text{ }^{\circ}\text{C}$, $\epsilon_{deh} = 0.9$; (a) $\epsilon_{hum,Tb} = 0.94$; (b) $\epsilon_{hum,Pb} = 0.94$; (c) $\epsilon_{hum,Tb} = 1.00$; (d) $\epsilon_{hum,Pb} = 1.00$.

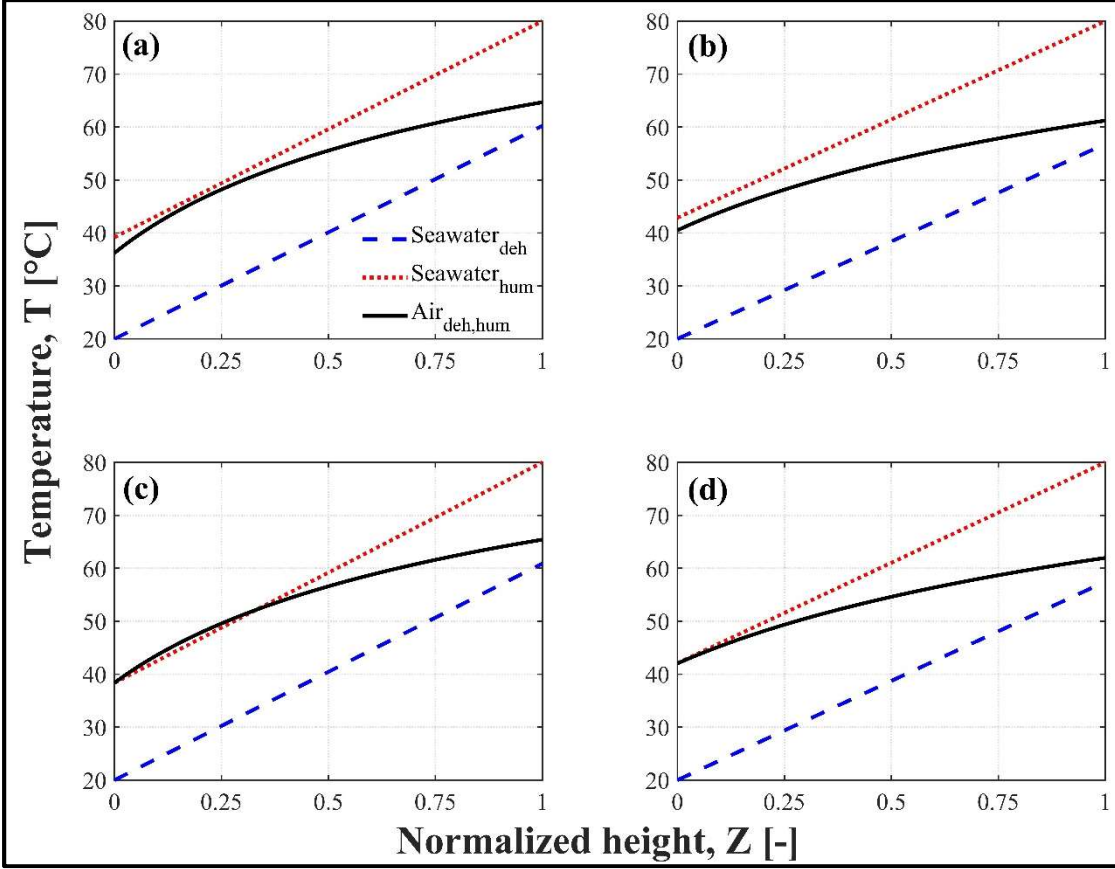


Figure 3.7. Temperature distribution for unbalanced ($MR < MR_{opt}$) CAOW HDH cycle; $T_{w,0} = 20\text{ }^{\circ}\text{C}$; $T_{w,2} = 80\text{ }^{\circ}\text{C}$, $\epsilon_{deh} = 0.9$; (a) $\epsilon_{hum,Tb} = 0.94$; (b) $\epsilon_{hum,Pb} = 0.94$; (c) $\epsilon_{hum,Tb} = 1.00$; (d) $\epsilon_{hum,Pb} = 1.00$.

For example, in the case of the balanced system, terminal-based effectiveness is taken as 94 and 100 % for both the dehumidifier and humidifier, where pinch-based effectiveness is also 94 and 100 %, as shown in Figs. 3.6 (a) and (b), respectively. From Fig. 3.6 (Terminal-based), it is clear there is a temperature cross between the seawater and air streams in the humidifier, which means the mass and heat transfer processes in the humidifier are reversed, which defeat the purpose of humidification; also, there is a high probability of second law violation ($S_{gen} < 0$) making the system not viable. On the other hand, there is no temperature cross in the humidifier when the temperature distribution is

estimated using the pinch-based effectiveness model. This can be attributed to the current definition of energy effectiveness based on the enthalpy difference at the terminal points. Where in the practical situation of the humidifier, it is based on a single intermediate point. Based on this, it can be said that the humidifier effectiveness (based on terminal points) has a ceiling value (84.5 % for the case in Fig. 3.6(b)) that cannot be exceeded to avoid temperature cross as well as the high probability of violating the second law.

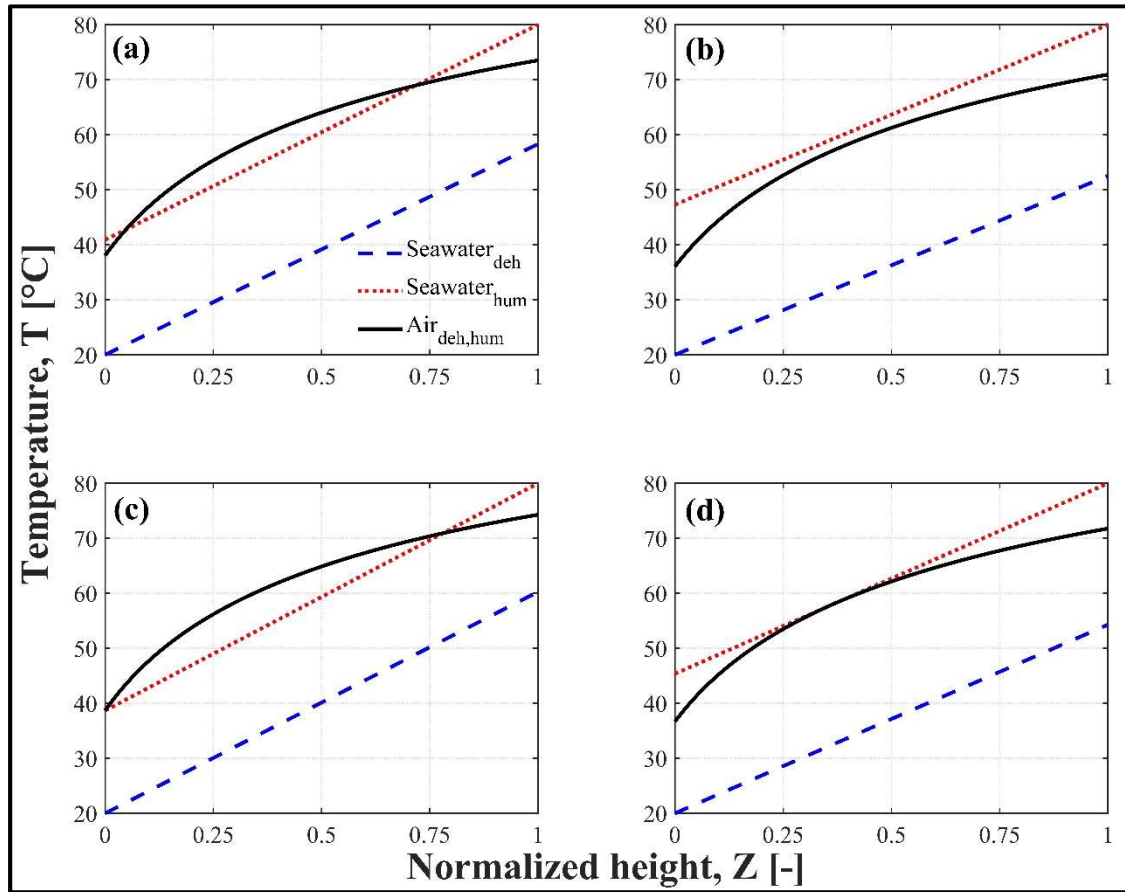


Figure 3.8. Temperature distribution for unbalanced ($MR > MR_{opt}$) CAOW HDH cycle; $T_{w,0} = 20\text{ }^{\circ}\text{C}$; $T_{w,2} = 80\text{ }^{\circ}\text{C}$, $\epsilon_{deh} = 0.9$; (a) $\epsilon_{hum,Tb} = 0.94$; (b) $\epsilon_{hum,Pb} = 0.94$; (c) $\epsilon_{hum,Tb} = 1.00$; (d) $\epsilon_{hum,Pb} = 1.00$.

This ceiling value can be calculated when the seawater line starts touching the air curve in the pinch point (losses at humidifier pinch point = 0 kJ kg_{da}⁻¹). However, this ceiling value is confusing, and it disturbs the basic common sense that the maximum potential value for the effectiveness should be 100 %. In order to solve this issue, a new energy effectiveness definition is used, as shown below in Eq. (3.12).

The definition in Eq. (3.1) is a mix between terminal-based and pinch-based models, which means using the pinch-based model is the best solution for the temperature cross issue, not to mention the pinch-based definition does not conflict with common sense.

In the case of the unbalanced system ($MR < MR_{opt}$), as shown in Fig. 3.7, the results of both models are consistent. This is because, as MR decreases, the humidifier's air temperature drops due to the increase in air quantity. It is important to note that the temperature cross becomes worse in the case of an unbalanced system ($MR > MR_{opt}$), as shown in Fig. 3.8 for the terminal-based effectiveness model. Regardless, it can be said that at higher values of energy effectiveness, the terminal-based model fails, particularly in the balanced condition. As the balanced condition is the design condition, especially for multiple extractions, it can be concluded that the pinch-based effectiveness model is the most suitable for the balanced CAOW HDH cycle with/without extraction.

Figures 3.6 – 3.8 are based on 20 and 80 °C minimum and maximum temperatures, respectively. In order to generalize these figures, they are normalized with respect to minimum and maximum temperatures. This is carried out to show the temperature distribution of water and air streams in both the dehumidifier and humidifier disregard to minimum and maximum temperatures as show in Fig. 3.9.

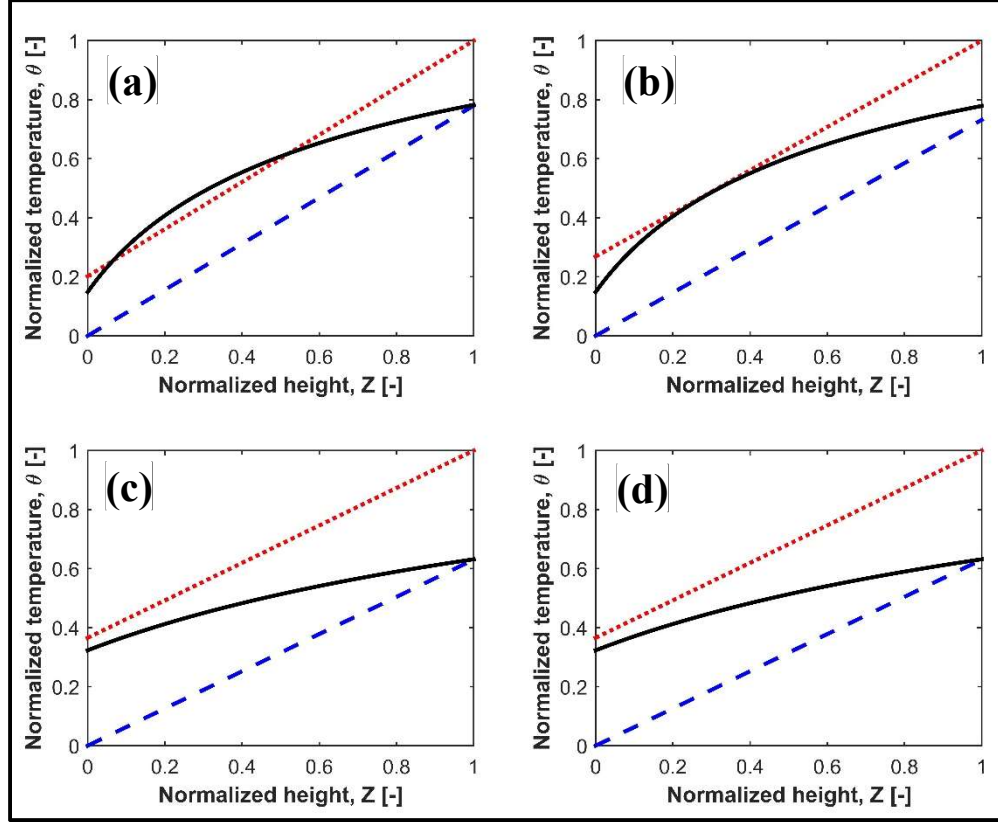


Figure 3.9. Normalized temperature distribution for thermodynamically-balanced CAOW HDH cycle; $\varepsilon_{deh} = 0.9$; (a) $\varepsilon_{hum,Tb} = 0.94$; (b) $\varepsilon_{hum,Pb} = 0.94$; (c) $\varepsilon_{hum,Tb} = 1.00$; (d) $\varepsilon_{hum,Pb} = 1.00$.

Figure 3.10 shows the effect of water-to-air mass flow rate ratio (MR) on the performance of the CAOW HDH system at different values of energy effectiveness 100% and 90% as illustrated in Fig. 3.10 a and b, respectively. The general trend shown in this Figure is that, as MR increases, the gain output ratio (GOR) increases until a certain point (optimum MR) where the GOR value is maximum. After this point, the GOR starts to decrease as the values of MR rises. It is to be noted that the performance of the system is over-estimated when the terminal-based energy effectiveness is used compared to pinch-based energy effectiveness. However, the terminal-based estimations are not reliable due to their affinity

to violate the second law especially at the optimum operating conditions as shown in Figure 3.10.

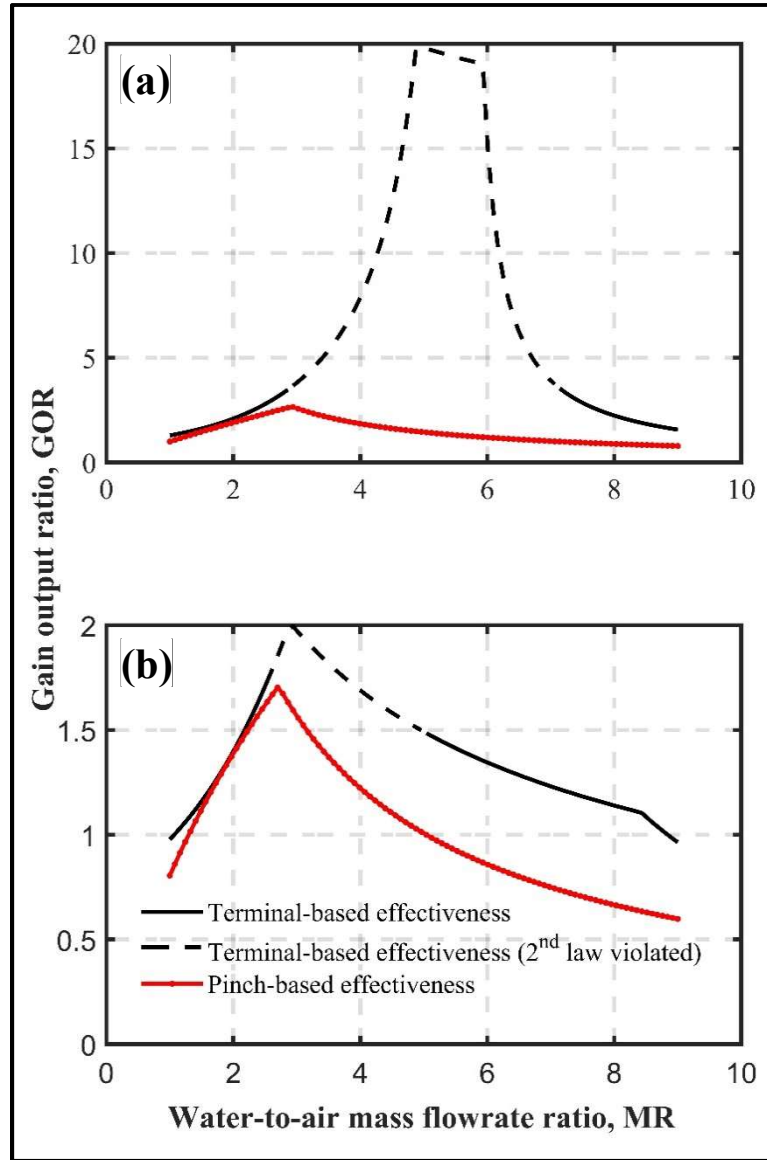


Figure 3.10. Impact of mass flowrate ratio of seawater to air on the GOR of the CAOW humidification dehumidification cycle; $T_{w,0} = 20\text{ }^{\circ}\text{C}$; $T_{w,2} = 80\text{ }^{\circ}\text{C}$, $\epsilon_{\text{deh}} = 0.9$; (a) $\epsilon_{\text{deh}} = \epsilon_{\text{hum,Tb}} = \epsilon_{\text{hum,Pb}} = 1.00$; (b) $\epsilon_{\text{deh}} = \epsilon_{\text{hum,Tb}} = \epsilon_{\text{hum,Pb}} = 0.90$.

3.7 Design procedure of the HDH system

In this section, an optimized, direct, and simple design process is proposed for the CAOW-WH HDH cycle. The sequence of the design procedure is to use the input data in the thermal model to specify the optimum operating conditions. Afterward, the size of the humidifier and dehumidifier is determined based on the location and amount of freshwater needed. The installation location of the system indicates the feedwater *temperature* and *salinity*, as well as the heating temperature assuming the energy source, is solar. These variables are design inputs. The effectiveness of both the humidifier and dehumidifier are optimized with the objectives of maximizing the performance and minimizing the system size for a specified value of freshwater production.

3.7.1 Practical range of pinch-based energy effectiveness

Figures 3.11 and 3.12 show the effect of humidifier height (H) and dehumidifier area (A), respectively, on the gain output ratio (GOR) and also the corresponding pinch-based energy effectiveness of the humidifier and dehumidifier, respectively. As shown in both figures increasing the humidifier height or dehumidifier area leads to enhancing the energy effectiveness of both components and consequently improving the GOR. From Figs. 3.11 and 3.12, in region (a), $H_{\text{hum}} = (0 - 1 \text{ m} \ \& \ A_{\text{deh}} = 0.4 - 0.48) \text{ m}^2$, both components sizes have significant effect on the GOR compared with regions (b) where $H = (1 - 5 \text{ m} \ \& \ A_{\text{deh}} = 0.48 - 0.8) \text{ m}^2$ and region (c) where $H > 5 \text{ m} \ \& \ A_{\text{deh}} > \text{m}^2$. On the other hand, the GOR in the region (a) is very low, less than 1.5. Therefore, it is not recommended to design the humidifier in this region. Moreover, in the region (c), the GOR does not change significantly with the system size. In other words, designing either the humidifier or

dehumidifier in these regions leads to an increase in capital costs without a significant improvement in the performance.

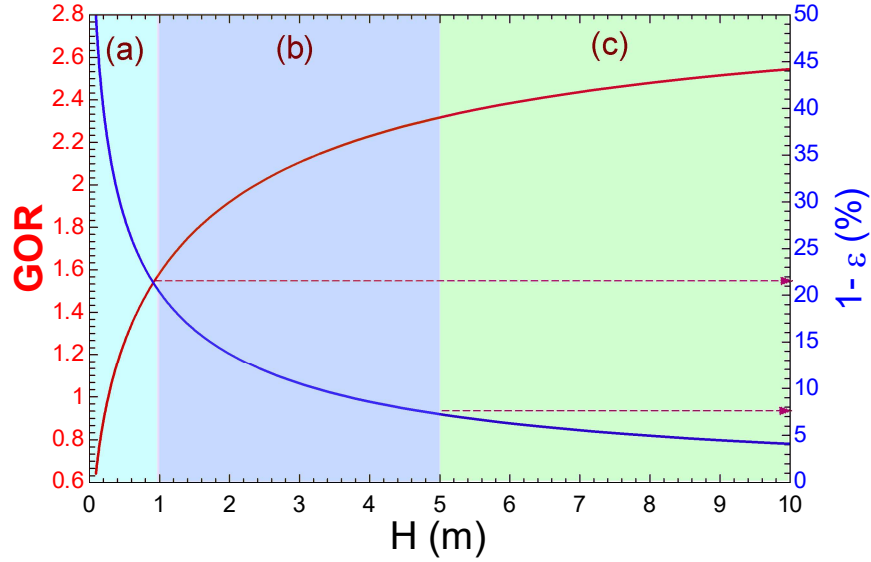


Figure 3.11. Effect of humidifier height (H) on the gain output ratio and humidifier effectiveness; $T_{w,0} = 20$ °C; $T_{w,2} = 80$ °C.

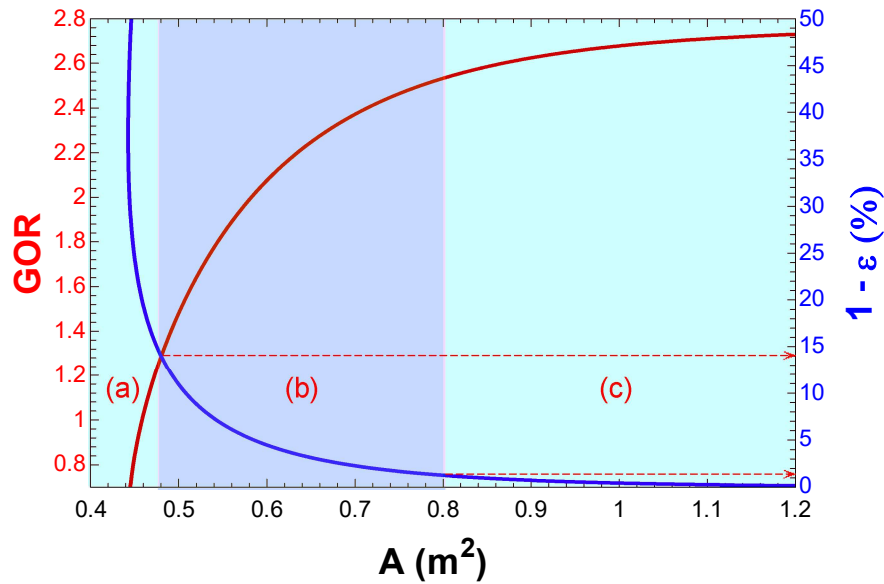


Figure 3.12. Effect of dehumidifier area (A) on the gain output ratio and dehumidifier effectiveness; $T_{w,0} = 20$ °C; $T_{w,2} = 80$ °C.

It is favorable to design the dehumidifier/humidifier in the region (b) due to its high performance compared to the moderated system size. In other words, based on the minimum and maximum temperature of 20 and 80 °C, the recommended effectiveness of humidifier and dehumidifier is in the range of (78 - 92.5) and (86 – 98.5), respectively.

3.7.2 Proxy indicator (BTC) optimization

The improvement in the HDH system performance by increasing both humidifier and dehumidifier energy effectiveness is attained with a toll that more mass and heat transfer areas are needed in both of them. These dimensions are converted to a proxy indicator that represents the cycle initial cost as stated by Elzayed et al. [56]. In order to make the HDH system viable, a humidifier/dehumidifier benefit to cost (BTC) indicator is utilized, as shown in Eq. (3.37). This indicator is defined as the system performance (GOR) normalized to the humidifier/dehumidifier size. As the proxy indicator rises, the performance of the HDH system increases at a lower capital cost. From Fig. 3.13, the HDH system is economically viable at a certain value of pinch-based energy effectiveness that is 20%; however, the GOR is very low ($GOR \approx 0.1$). Therefore, a condition of $GOR \leq 1$ is added and hence, it can be concluded that the optimum BTC occurs at energy effectiveness of 80%. It is to be noted that Fig. 3.13 is generated at a minimum and maximum temperature of 20 and 80 °C. In order to generalize the data in Fig. 3.13, it is regenerated for a range of (20 – 40 °C) and (60 – 80 °C) of the minimum and maximum temperatures, respectively.

$$BTC = \frac{GOR}{A_{c,hum} Z_{hum} A_{deh}} \quad (3.37)$$

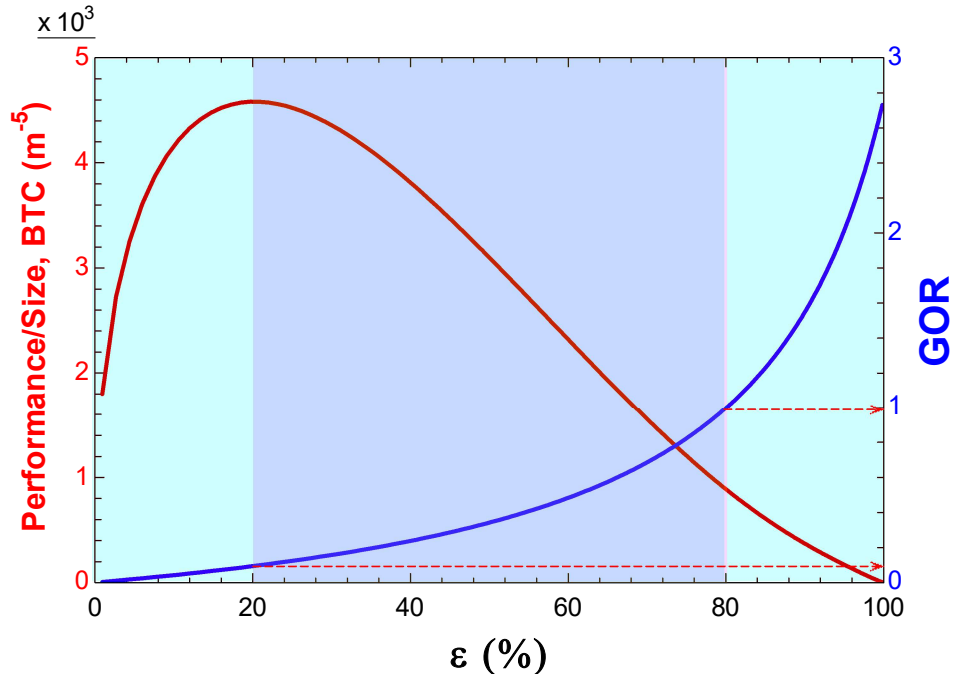


Figure 3.13. Effect of dehumidifier/humidifier pinch-based energy effectiveness on the gain output ratio and BTC; $T_{w,0} = 20\text{ }^{\circ}\text{C}$; $T_{w,2} = 80\text{ }^{\circ}\text{C}$.

Table 3.5. Direct performance and sizing table for the design procedure.

	Maximum temperature, T_2 ($^{\circ}\text{C}$)								
	60			70			80		
	20	30	40	20	30	40	20	30	40
Minimum temperature, T_0 ($^{\circ}\text{C}$)									
The production rate of freshwater (kg hr^{-1})	1 (reference value)								
Effectiveness of the dehumidifier/humidifier	77.5	75.8	74.0	78.8	77.2	75.5	80.0	78.6	77.3
BTC indicator (m^{-5})	192	173	119	431	439	383	887	974	969
Heat input rate (kW)	0.67								
Recovery ratio, RR	2.93	2.24	1.52	3.71	3.01	2.29	4.49	3.79	3.06
Feedwater flow rate (kg hr^{-1})	34.2	44.6	65.8	26.9	33.2	43.7	22.3	26.4	32.6
Optimum mass flow rate ratio, MR	1.74	2.3	3.09	2.13	2.83	3.85	2.64	3.54	4.87
Dry air flow rate (kg hr^{-1})	19.1	19.0	20.9	12.2	11.4	11.1	8.07	7.17	6.5
Dehumidifier NTU	1.69	1.55	1.44	1.72	1.59	1.49	1.76	1.65	1.55
The surface area of the dehumidifier	0.98	0.92	0.97	0.65	0.59	0.56	0.46	0.40	0.36
Merkel number at the humidifier	1.19	0.90	0.68	0.92	0.68	0.51	0.69	0.51	0.37
Height of packing material of the humidifier (m)	0.83	0.76	0.71	0.71	0.63	0.58	0.59	0.53	0.47

3.7.3 Case study

This part focuses on illustrating a demonstrative case that considers a system to be installed in Dhahran, Saudi Arabia with a freshwater production of $40 \text{ liter day}^{-1}$ that is equivalent to 5 kg hr^{-1} for 8 hours of system operation per day. It is needed to size the CAOW - WH humidification dehumidification cycle using the following parameters of design,

- The terminal-based effectiveness of the humidifier/dehumidifier (ϵ_{hum}).
- The required amount of input heat rate ($\dot{Q}$).
- The feedwater flow rate supplied to the system ($\dot{m}_w$).
- The dry air flow rate circulated between the humidifier and dehumidifier.
- The dehumidifier surface area (A_{deh}).
- The humidifier cross-section area and height of packing material (A_{pack} and H_{pack})

Consider Table 3.6, through the year, the average ambient temperature in Dhahran is 26.7°C . Due to evaporation at the Arabian Gulf surface, feedwater temperature is less than the ambient air temperature; it is assumed to be 25°C . Solar energy in Saudi Arabia is feasible, therefore, the maximum temperature can reach values over 70°C .

Table 3.6. Monthly weather forecast and climate Dhahran, Saudi Arabia [69].

Month	Ambient temperature (°C)		
	High	Low	Average
January	21	10	15.5
February	23	11	17
March	27	15	21
April	33	19	26
May	39	24	31.5
June	42	27	34.5
July	44	29	36.5
August	43	28	35.5
September	41	25	33
October	36	22	29
November	29	17	23
December	23	12	17.5

Considering minimum and maximum temperatures of 25 and 70°C, respectively, and freshwater production of 5 kg hr⁻¹, using Table 3.5,

- The humidifier and dehumidifier effectiveness are equal to 78%.
- The required $\dot{Q}=0.67*5=3.3$ kW.
- The $RR=3.36$ and the required $\dot{m}_w=30.05*5=150$ kg hr⁻¹.
- $MR=2.48$ and the required $\dot{m}_d=11.8*5=59$ kg hr⁻¹.
- The number of transfer unit (NTU) = 1.66 and considering $U=0.01$ kW m⁻² K⁻¹, the required $A_{deh}=0.62*5=3.1$ m².
- The Merkel number (Me) = 0.8 and the required $H_{pack}=0.67$ m

The cross-section area of the humidifier is, $A_{hum} = \frac{\dot{m}_w}{1.5} = \frac{150/3600}{1.5} = 0.028$ m²

which is a 16x16 cm area.

3.8 Concluding remarks

A novel method based on energy effectiveness as a performance index is suggested in the current research in order to investigate and model the water-heated, CAOW humidification dehumidification desalination cycle at balanced and unbalanced states. The proposed method objective is evading the possibility of temperature cross and second law violation as a probable result in the humidifier. It has been found the proposed method is tested and verified against the theoretical data in the reported literature. An assessment of the evaluated results based on the current model and the literature is presented and good agreement is noted between the current model and theoretical data with a minor error of $\pm 2.29\%$. A direct design procedure is proposed in this research in order to size the closed-air open-water water-heated humidification dehumidification desalination system. The city of Dhahran is taken as a case study where the average minimum and maximum temperature of seawater is assumed 25 and 70°C, respectively. The system is designed to produce 5 kg hr⁻¹ of freshwater and the optimal energy effectiveness is found to be 78%.

CHAPTER 4

ANALYTICAL SOLUTION FOR THE PINCH-BASED

ENERGY EFFECTIVENESS DEFINITION

Summary

The energy and mass balance equations are used to model the humidification dehumidification cycles by utilizing a simultaneous model based on the humid air thermophysical library. This implicit model is burdensome, especially, in modeling the complex arrangement of humidification dehumidification cycles. This chapter presents an explicit, novel, and robust model that facilitates the modeling and designing of the system. The correlation between the humid air enthalpy and temperature as an important factor in system modeling is made simpler in this chapter. On this basis, the system is modeled analytically. The proposed analytical model is tested and validated against the theoretical data in the reported literature. An assessment of the gained output ratio is evaluated utilizing the current analytical model and the numerical model presented in chapter 3. From this comparison, a good agreement is observed between the current model and theoretical data with an error of around $\pm 5\%$.

4.1 Introduction

In 2015, the United Nations adopted a set of 17 sustainable development goals to end poverty, protect the planet, and ensure prosperity for all. Among the 17 goals identified for sustainable growth Climate action is one of the most important goals that impact the entire world population and the delicate ecosystem of earth. The need for climate action arises due to the alarming rise in global temperatures since the industrial revolution in the 1700s [70]. The major reason for the temperature rise is attributed to the emission of Green House Gases (GHG), like carbon dioxide (CO₂) from burning fossil fuels. In 2014, about 68% of the global GHG emission was generated from energy production [71]. The electricity and heat sector alone produces 42% of the total CO₂ emission. In countries with hot climatic conditions such as Saudi Arabia, about 70% of the energy is used to meet the energy demands of domestic needs [72].

Another important goal identified by the UN for sustainable development is to meet the rising freshwater demand. Potable water is becoming a rare supply in different areas of the globe. Some estimates [73], [74] suggest that more than a third of the global populace suffer from freshwater scarcity while another [75] suggests the number could be as high as 60%. One of the major causes is the rise in population, which increases the consumption of freshwater in countries like India and Turkey that is expected to reach 60% by 2025 [76]. Thus, freshwater desalination technologies aggregated with preservation efforts are required especially in areas that have a low amount of natural freshwater resources [2]. Countries like Saudi Arabia which have few groundwater resources and utilize seawater, rely heavily on Desalination systems to meet the freshwater demand.

A major share of the global populace lives within 40 miles from coastlines [77], which meet the requirements for industrialized desalination as a potential resolution to this predicament. Humidification dehumidification cycles can be considered as favorable thermal-based desalination techniques. Therefore, it is appropriate for producing moderate potable water. Hence, it can be said that it is simple and therefore, appropriate for minority groups. Also, HDH cycles are acknowledged for enduring a widespread salinities gamut with a negligible amount of maintenance required [78].

The simplest form of closed-air open-water (CAOW) water-heated HDH system is studied in this chapter (please refer to Fig. 3.1).

This chapter incorporates an innovative method of modeling for avoiding the use of the thermos-physical library of fully saturated air. These techniques provide an easy and precise analytical solution; thus, designers and researchers can easily use them to improve HDH cycle performance.

4.2 Performance and operating parameters

The performance of HDH systems is measured using the gained output ratio (GOR) as a criterion that can be explained as the amount of energy carried by the produced H₂O (output) as a result of the energy supplied (input), it is expressed as,

$$GOR = \frac{\dot{m}_{fw} h_{fg}}{\dot{Q}_{ht}} \quad (4.1)$$

The operational variables needed to estimate the cycle performance are enlisted as follow,

- Seawater mass flowrate at dehumidifier inlet, $\dot{m}_w$.
- The mass flow rate ratio of seawater to air, MR.
- Seawater temperature and concentration at dehumidifier inlet, T_0, s_0 .
- The seawater top temperature, T_2 .
- Energy effectiveness of the humidifier and dehumidifier ($\varepsilon_{hum}, \varepsilon_{deh}$).

The basic values of the main operational variables are enlisted as depicted in Table 4.1.

Table 4.1. Main operational variables basic values and variations.

Operating parameter	Typical value	Range
The mass flow rate of the feed seawater (kg s^{-1})	1	-
Seawater bottom temperature, T_0 ($^{\circ}\text{C}$)	30	10 – 30
The salinity of the feed seawater, s_0 (ppt)	35	20 – 50
The mass flow rate ratio of seawater to air, $\dot{m}_w$ (kg s^{-1})	opt.	0.5 - 4
Heating temperature, T_2 ($^{\circ}\text{C}$)	80	70 – 90
Energy effectiveness, $\varepsilon_{deh}, \varepsilon_{hum}$ (kJ kg_{da}^{-1})	0.9	0.5 – 0.9

In this section, the thermodynamic analysis of the humidification dehumidification cycle is articulated analytically founded above the numerical method mentioned in chapter 3 to evaluate the humidification dehumidification cycle performance. To perform these manipulations, the following presumptions are implemented,

- The examined processes of heat and mass transfer are in steady-state.
- Insulation is applied to both cycle towers (humidifier and dehumidifier) as well as the seawater heater.

- The needed power for the recirculated air and seawater pumps is negligible compared to the rate of heat input.
- The temperature of freshwater production is assumed as the average recirculated air temperature.
- The recirculated air is relative humidity is assumed at 100% [62].

4.3 Analytical modeling

This part comprises an exact and innovative mathematical model which is utilized as an improvement for that mentioned in chapter 3. The conservation and balance equations mentioned in the previous chapter depend considerably on the thermos-physical data of the humid air, especially the correlation that governs air specific enthalpy and temperature. Because of the importance of the correlation on HDH modeling, a try is carried out to make this correlation simpler to overcome the complication of the model. Therefore, humid air enthalpy under a pressure of 1 atm can be stated as,

$$h(T) = C_{p,a}T + \omega(C_{p,v}T + h_{fg}) \quad (4.2)$$

where,

$$\omega = 0.62198 \frac{P_v}{101.325 - P_v} \quad (4.3)$$

The partial vapor pressure at saturation point, P_v , could be stated by utilizing Tetens' equation [79],

$$P_v = 0.61078 e^{\frac{17.27T}{T+237.3}} \quad (4.4)$$

With presumption of $C_{p,a} = 1.006 \text{ kJ kg}^{-1}\text{K}^{-1}$, $C_{p,v} = 1.84 \text{ kJ kg}^{-1}\text{K}^{-1}$, and $h_{fg} = 2501 \text{ kJ kg}^{-1}$, Eq. (4.2) is still complex and difficult to be handled. To illustrate, to inverse this relation and get 'T' as a function of 'h' is not possible and hence, it cannot be used as it is in this chapter. Therefore, for further comprehension of Eq. (4.2), this relation is depicted as

shown in Fig. 4.1 which covers the probable gamut of saturated air temperature which is, (1 – 99 °C) for the HDH system.

It is to be noted that the data illustrated in Fig. 4.1 was evaluated using the Hyland and Wexler data [63] to guarantee maximum precision. Figure 4.1 represents that as humid air enthalpy rises, the temperature rises abruptly until a level $h = 1000 \text{ kJ kg}_d^{-1}$. Afterward the value of $h = 1000 \text{ kJ kg}_d^{-1}$, the air temperature rises gradually as enthalpy rises till it is nearly constant at a level of $T = 98 \text{ }^\circ\text{C}$. Because of the great variation of behavior in the temperature of the air after the level of $h = 1000 \text{ kJ kg}_d^{-1}$, the correlation illustrated in Fig. 4.1 is difficult to be fitted.

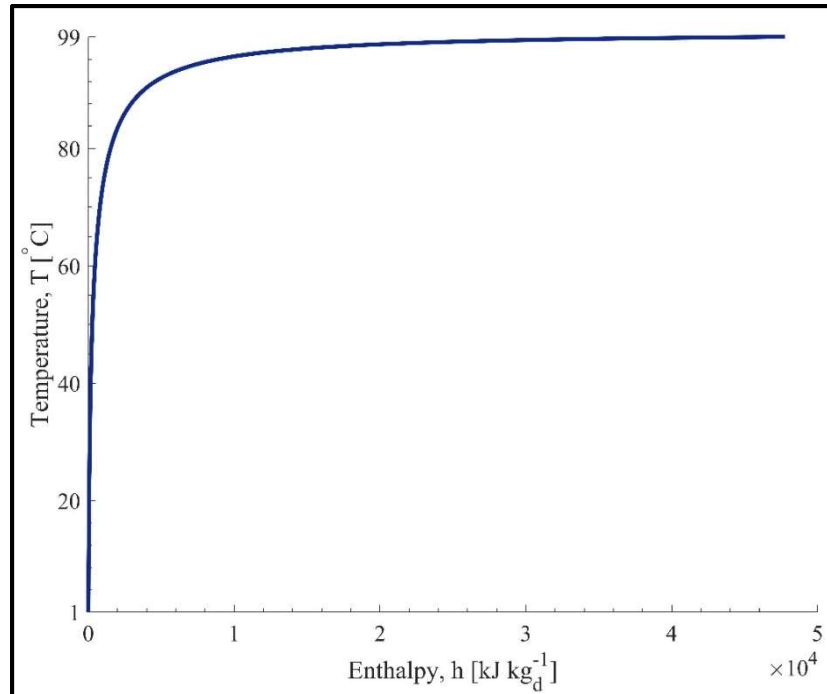


Figure 4.1. The relation between specific enthalpy and saturated air temperature pressure of 1 atm [40].

However, approximation forms of different order-polynomial, power, logarithmic, and exponential are tried for fitting the relation between specific enthalpy and saturated air temperature pressure of 1 atm. The aforementioned approximation performance was poor and the natural logarithm approximation was the least, however, it coincides with the Hyland and Wexler data [63] in only one point that is $T = 91\text{ }^{\circ}\text{C}$. Thus, Fig. 4.1 is re-drawn in a semi-ln scheme for additional study as depicted in Fig. 4.2.

As illustrated in Fig. 4.2, the semi-ln trend of ‘T’ and ‘h’ is impeccably linear in the range of 40 - 60 $^{\circ}\text{C}$. This trend varies regularly as the air temperature rises or declines from this range. Figure 4.1 shows that between $T_{\text{air}} = 25 - 65\text{ }^{\circ}\text{C}$ that is, $T_{\text{water}} = 20 - 90\text{ }^{\circ}\text{C}$ for the optimum system at high effectiveness of 0.9 this correlation can be stated as,

$$T = \zeta_0 + \zeta_1 \ln(h) \quad (4.5)$$

where,

$$\zeta_0 = -54.4727862$$

$$\zeta_1 = 18.4993347$$

Eq. (4.5) shows a ln estimation of the correlation of the humid air temperature and specific enthalpy. This estimation can be utilized to make the HDH system model simpler. Based on Eq. (4.5), saturated air specific enthalpy is stated as,

$$h = \exp\left(\frac{T - \zeta_0}{\zeta_1}\right) \quad (4.6)$$

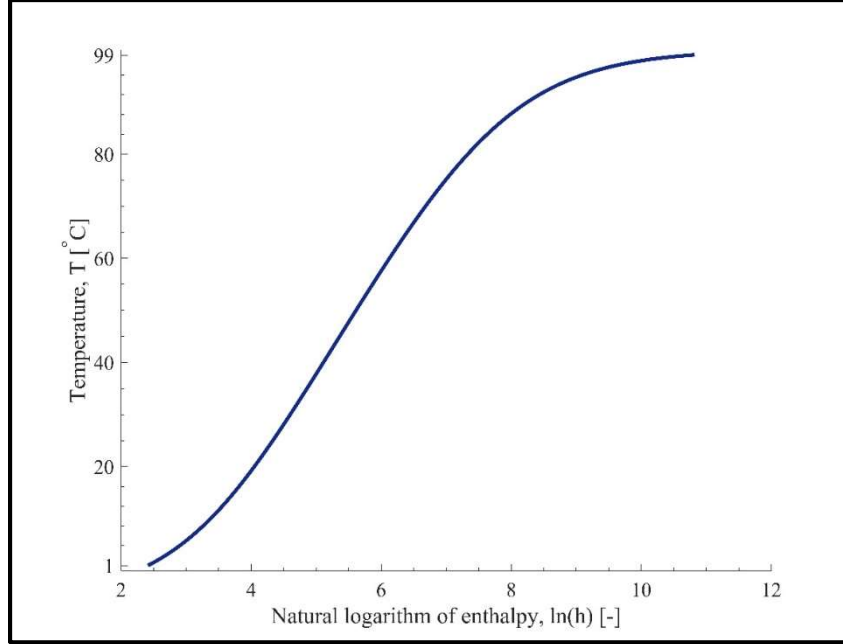


Figure 4.2. The relation between saturated air natural logarithm of specific enthalpy and temperature at a pressure of 1 atm [40].

4.3.1 The balanced system, $MR = MR_{opt}$

In this section, the model of the HDH system with optimum MR is re-examined and made simple by utilizing the $\ln$ estimation. The generated system of equations is resolved analytically for giving a better understanding of the humidification dehumidification cycle.

Based on the definition reported by Narayan et al. [54], the energy effectiveness for the towers of dehumidification and humidification can be expressed as in Eqs. (3.11) and (3.13), respectively. In balance state i.e. $HCR = 1$, the energy effectiveness in both air and water sides is the same, therefore, the air side effectiveness is considered for the dehumidifier due to its simplicity as stated in Eq. (4.7),

$$\varepsilon_{deh} = \frac{h_{a,2} - h_{a,1}}{h_{a,2} - h_{a,1,id}} \quad (4.7)$$

Similar to Eq. (4.7), the energy effectiveness for the humidifier can be stated as,

$$\varepsilon_{hum} = \frac{h_{a,2} - h_{a,1}}{h_{a,2,id} - h_{a,1}} \quad (4.8)$$

It is clear from Eqs. (4.7 – 4.8) that the only unknowns are $h_{a,1}$ and $h_{a,2}$, therefore, Eqs. (4.7 – 4.8) can be solved simultaneously to find $h_{a,1}$ and $h_{a,2}$ as follow,

$$h_{a,1} = \frac{\varepsilon_{deh} h_{a,1,id} + \varepsilon_{hum} (1 - \varepsilon_{deh}) h_{a,2,id}}{\varepsilon_{deh} + \varepsilon_{hum} (1 - \varepsilon_{deh})} \quad (4.9)$$

$$h_{a,2} = \frac{\varepsilon_{deh} (1 - \varepsilon_{hum}) h_{a,1,id} + \varepsilon_{hum} h_{a,2,id}}{\varepsilon_{deh} (1 - \varepsilon_{hum}) + \varepsilon_{hum}} \quad (4.10)$$

where,

$$h_{a,1,id} = \exp\left(\frac{T_0 - \zeta_0}{\zeta_1}\right) \quad (4.11)$$

$$h_{a,2,id} = \exp\left(\frac{T_2 - \zeta_0}{\zeta_1}\right) \quad (4.12)$$

Using Eqs. (4.11 – 4.12), Eqs. (4.9 – 4.10), both $h_{a,1}$ and $h_{a,2}$ can be expressed as follow,

$$h_{a,1} = \frac{\varepsilon_{deh} \exp\left(\frac{T_0 - \zeta_0}{\zeta_1}\right) + \varepsilon_{hum} (1 - \varepsilon_{deh}) \exp\left(\frac{T_2 - \zeta_0}{\zeta_1}\right)}{\varepsilon_{deh} + \varepsilon_{hum} (1 - \varepsilon_{deh})} \quad (4.13)$$

$$h_{a,2} = \frac{\varepsilon_{deh} (1 - \varepsilon_{hum}) \exp\left(\frac{T_0 - \zeta_0}{\zeta_1}\right) + \varepsilon_{hum} \exp\left(\frac{T_2 - \zeta_0}{\zeta_1}\right)}{\varepsilon_{deh} (1 - \varepsilon_{hum}) + \varepsilon_{hum}} \quad (4.14)$$

Applying the log approximation, air bottom, and top temperature can be expressed as follow,

$$T_{a,1} = \zeta_0 + \zeta_1 \ln \left(\frac{\varepsilon_{deh} \exp\left(\frac{T_0 - \zeta_0}{\zeta_1}\right) + \varepsilon_{hum} (1 - \varepsilon_{deh}) \exp\left(\frac{T_2 - \zeta_0}{\zeta_1}\right)}{\varepsilon_{deh} + \varepsilon_{hum} (1 - \varepsilon_{deh})} \right) \quad (4.15)$$

$$T_{a,2} = \zeta_0 + \zeta_1 \ln \left(\frac{\varepsilon_{deh} (1 - \varepsilon_{hum}) \exp\left(\frac{T_0 - \zeta_0}{\zeta_1}\right) + \varepsilon_{hum} \exp\left(\frac{T_2 - \zeta_0}{\zeta_1}\right)}{\varepsilon_{deh} (1 - \varepsilon_{hum}) + \varepsilon_{hum}} \right) \quad (4.16)$$

Repeating the same procedure for the seawater stream the energy effectiveness for the dehumidifier can be represented as,

$$\varepsilon_{deh} = \frac{h_1 - h_0}{h_{1,id} - h_0} \quad (4.17)$$

From Eq. (4.17), the only unknown is h_1 and it can be expressed as,

$$h_1 = h_0 + \varepsilon_{deh} (h_{1,id} - h_0) \quad (4.18)$$

where,

$$\begin{aligned} h_0 &= (\zeta_2 - \zeta_3 s_0 + (\zeta_4 s_0 - \zeta_5)(T_0 + \zeta_6)) T_0 \\ h_{1,id} &= (\zeta_2 - \zeta_3 s_0 + (\zeta_4 s_0 - \zeta_5)(T_{a,2} + \zeta_6)) T_{a,2} \\ \zeta_2 &= 5.328 \\ \zeta_3 &= 9.76 \times 10^{-2} \\ \zeta_4 &= 7.351 \times 10^{-4} \\ \zeta_5 &= 6.913 \times 10^{-3} \\ \zeta_6 &= 273.15 \end{aligned} \quad (4.19)$$

Applying Eqs. (4.16) and (4.19), Eq. (4.18) becomes,

$$h_1 = h_0 + \varepsilon_{deh} \left(\left(\zeta_2 - \zeta_3 s_0 + (\zeta_4 s_0 - \zeta_5) (T_{a,2} + 273.15) \right) T_{a,2} - \left(\zeta_2 - \zeta_3 s_0 + (\zeta_4 s_0 - \zeta_5) (T_0 + \zeta_6) \right) T_0 \right) \quad (4.20)$$

Using Eq. (4.20), the feed seawater temperature is stated as,

$$T_1 = \frac{\sqrt{h_1 (\zeta_4 s_0 - \zeta_5) - \zeta_7^2} + \zeta_7}{\zeta_4 s_0 - \zeta_5} \quad (4.21)$$

where,

$$\zeta_7 = \zeta_3 s_0 - \zeta_4 \zeta_6 s_0 - \zeta_2 + \zeta_5 \zeta_6$$

From energy balance in the dehumidifier, see Eq. (3.4), the dry air mass flowrate ratio can be expressed as,

$$\dot{m}_{da} = \frac{\dot{m}_w (h_1 - h_0)}{(h_{a,2} - h_{a,1})} \quad (4.22)$$

Applying the Eqs. (4.13 – 4.14) and (4.18 – 4.19), Eq. (4.22) can be depicted as,

$$\dot{m}_{da} = \frac{\dot{m}_w \left(\left(\zeta_2 - \zeta_3 s_0 + (\zeta_4 s_0 - \zeta_5) (T_{a,2} + 273.15) \right) T_{a,2} - \left(\zeta_2 - \zeta_3 s_0 + (\zeta_4 s_0 - \zeta_5) (T_0 + \zeta_6) \right) T_0 \right)}{\left(\varepsilon_{deh} (1 - \varepsilon_{hum}) + \varepsilon_{hum} \right)^{-1} \left(\varepsilon_{deh} (1 - \varepsilon_{hum}) \exp \left(\frac{T_0 - \zeta_0}{\zeta_1} \right) + \varepsilon_{hum} \exp \left(\frac{T_2 - \zeta_0}{\zeta_1} \right) \right) - \exp \left(\frac{T_0 - \zeta_0}{\zeta_1} \right)} \quad (4.23)$$

The mass flow rate of freshwater produced can be found by applying Eq. (4.23) in the dehumidifier mass balance, see Eq. (3.3) as,

$$\dot{m}_{pw} = \frac{\dot{m}_w \left((\zeta_2 - \zeta_{s0} + (\zeta_4 s_0 - \zeta_5)(T_{a2} + 273.15)) T_{a2} - (\zeta_2 - \zeta_{s0} + (\zeta_4 s_0 - \zeta_5)(T_0 + \zeta_6)) T_0 \right)}{(\varepsilon_{dh}(1 - \varepsilon_{hum}) + \varepsilon_{hum})^{-1} \left(\varepsilon_{dh}(1 - \varepsilon_{hum}) \exp\left(\frac{T_0 - \zeta_0}{\zeta_1}\right) + \varepsilon_{hum} \exp\left(\frac{T_2 - \zeta_0}{\zeta_1}\right) \right) - \exp\left(\frac{T_0 - \zeta_0}{\zeta_1}\right)} (\omega_{a2} - \omega_{a1}) \quad (4.24)$$

where,

$$\omega_{a,1} = \frac{0.62198}{165.894 \exp\left(-\frac{17.27 T_{a,1}}{237.3 + T_{a,1}}\right) - 1} \quad (4.25)$$

$$\omega_{a,2} = \frac{0.62198}{165.894 \exp\left(-\frac{17.27 T_{a,2}}{237.3 + T_{a,2}}\right) - 1} \quad (4.26)$$

From the mass balance in the humidifier, the mass flowrate of the rejected brine is stated as,

$$\dot{m}_b = \dot{m}_w - \frac{\dot{m}_w \left((\zeta_2 - \zeta_{sw} + (\zeta_4 s_w - \zeta_5)(T_{a2} + 273.15)) T_{a2} - (\zeta_2 - \zeta_{sw} + (\zeta_4 s_w - \zeta_5)(T_0 + \zeta_6)) T_0 \right)}{(\varepsilon_{dh}(1 - \varepsilon_{hum}) + \varepsilon_{hum})^{-1} \left(\varepsilon_{dh}(1 - \varepsilon_{hum}) \exp\left(\frac{T_0 - \zeta_0}{\zeta_1}\right) + \varepsilon_{hum} \exp\left(\frac{T_2 - \zeta_0}{\zeta_1}\right) \right) - \exp\left(\frac{T_0 - \zeta_0}{\zeta_1}\right)} (\omega_{a2} - \omega_{a1}) \quad (4.27)$$

Similar to Eq. (4.7), the energy effectiveness for the humidifier can be defined using the seawater stream as,

$$\varepsilon_{hum} = \frac{\dot{m}_w h_2 - \dot{m}_b h_3}{\dot{m}_w h_2 - \dot{m}_b h_{3,id}} \quad (4.28)$$

$\dot{m}_b$ is known from Eq. (4.27) which means the only unknown is h_3 and can be evaluated by manipulating Eq. (4.28) as

$$h_3 = \frac{\dot{m}_w h_2 - \varepsilon_{hum} (\dot{m}_w h_2 - \dot{m}_b h_{3,id})}{\dot{m}_b} \quad (4.29)$$

where,

$$\begin{aligned} h_2 &= (\zeta_2 - \zeta_3 s_w + (\zeta_4 s_w - \zeta_5)(T_2 + \zeta_6))T_2 \\ h_{3,id} &= (\zeta_2 - \zeta_3 s_b + (\zeta_4 s_b - \zeta_5)(T_{a,1} + \zeta_6))T_{a,1} \end{aligned} \quad (4.30)$$

Using Eq. (4.29), the brine temperature is stated as,

$$T_3 = \frac{\sqrt{h_3(\zeta_4 s_b - \zeta_5) - \zeta_7^2} + \zeta_7}{\zeta_4 s_b - \zeta_5} \quad (4.31)$$

4.3.2 The unbalanced system, Case 1: $MR > MR_{opt}$

In this section, the model of the HDH system with MR as input is studied and simplified using the log approximation, and then the generated model is solved analytically. Using the definition of energy effectiveness for case 1 ($MR > MR_{opt}$, see Eq. (3.9)), Eqs. (4.9 – 4.16) can be used to evaluate the recirculated air temperature and enthalpy at states a,1, and a,2.

The dry air mass flowrate can be estimated directly as expressed in Eq. (4.32),

$$\dot{m}_{da} = \frac{\dot{m}_w}{MR} \quad (4.32)$$

Accordingly, the freshwater mass flowrate can be evaluated as,

$$\dot{m}_{pw} = \frac{\dot{m}_w}{MR} (\omega_{a,2} - \omega_{a,1}) \quad (4.33)$$

where $\omega_{a,1}$ and $\omega_{a,2}$ are defined in Eqs. (4.25) and (4.26), respectively.

From the mass balance in the humidifier, the mass flowrate of the rejected brine is stated as,

$$\dot{m}_b = \dot{m}_w - \frac{\dot{m}_w}{MR} (\omega_{a,2} - \omega_{a,1}) \quad (4.34)$$

From the energy balance in the dehumidifier, see Eq. (3.4), the only unknown is h_1 and it can be expressed as,

$$h_1 = h_0 + \frac{(h_{a,2} - h_{a,1})}{MR} \quad (4.35)$$

where h_0 is defined in Eq. (4.19).

Using Eq. (4.35), the temperature of the seawater at the dehumidifier outlet can be stated as,

$$T_1 = \frac{\sqrt{\left(h_0 + \frac{(h_{a,2} - h_{a,1})}{MR} \right) (\zeta_4 s_w - \zeta_5) - \zeta_7^2} + \zeta_7}{\zeta_4 s_w - \zeta_5} \quad (4.36)$$

where,

$$\zeta_7 = (\zeta_3 - \zeta_4 \zeta_6) s_w - \zeta_2 + \zeta_5 \zeta_6$$

Similarly, using the energy balance in the humidifier, see Eq. (3.2), the only unknown is h_3 and it can be expressed as,

$$h_3 = \frac{\dot{m}_w h_2 - \dot{m}_{da} (h_{a,2} - h_{a,1})}{\dot{m}_b} \quad (4.37)$$

where, h_2 is defined in Eq. (4.30).

Using Eq. (4.37), the brine temperature is stated as,

$$T_3 = \frac{\sqrt{h_3(\zeta_4 s_b - \zeta_5) - \zeta_7^2} + \zeta_7}{\zeta_4 s_b - \zeta_5} \quad (4.38)$$

4.3.3 The unbalanced system, Case 2: $MR < MR_{opt}$

In this section, the model of the HDH system with MR as input is studied and simplified using the log approximation, and then the generated model is solved analytically. Using the definition of energy effectiveness for case 2 ($MR < MR_{opt}$, see Eq. (3.10)), Eqs. (4.18 – 4.21) can be used to evaluate the seawater temperature and enthalpy at state 1. The dry air mass flowrate can be estimated directly using Eq. (4.32).

Using the energy effectiveness for the humidifier and dehumidifier (see Eqs. (4.17) and (4.28)), the energy balance for the humidifier and dehumidifier can be expressed as in Eqs. (4.39) and (4.40), respectively,

$$\dot{m}_w h_2 - \dot{m}_b h_3 = \varepsilon_{hum} (\dot{m}_w h_2 - \dot{m}_b h_{3,id}) = \dot{m}_{da} (h_{a,2} - h_{a,1}) \quad (4.39)$$

$$\dot{m}_w (h_1 - h_0) = \dot{m}_w \varepsilon_{deh} (h_{1,id} - h_0) = \dot{m}_{da} (h_{a,2} - h_{a,1}) \quad (4.40)$$

Equation (4.39) can be simplified as follow,

$$\varepsilon_{hum} (\dot{m}_w h_2 - (\dot{m}_w - \dot{m}_{fw}) h_{3,id}) = \dot{m}_{da} (h_{a,2} - h_{a,1}) \quad (4.41)$$

or,

$$\varepsilon_{hum} \left(\dot{m}_w h_2 - \left(\dot{m}_w - \dot{m}_{da} (\omega_{a,2} - \omega_{a,1}) \right) h_{3,id} \right) = \dot{m}_{da} (h_{a,2} - h_{a,1}) \quad (4.42)$$

By dividing by $\dot{m}_{da}$ Equations (4.40) and (4.42) can be stated as,

$$\varepsilon_{deh} MR (h_{1,id} - h_0) = (h_{a,2} - h_{a,1}) \quad (4.43)$$

$$\varepsilon_{hum} \left(MR h_2 - \left(MR - (\omega_{a,2} - \omega_{a,1}) \right) h_{3,id} \right) = (h_{a,2} - h_{a,1}) \quad (4.44)$$

It is clear from the psychometric chart that the relation between the absolute humidity ratio and specific enthalpy is linear, yet, for further investigation, this relation is plotted for an air temperature range of 25 -65 °C as shown in Fig. 4.3.

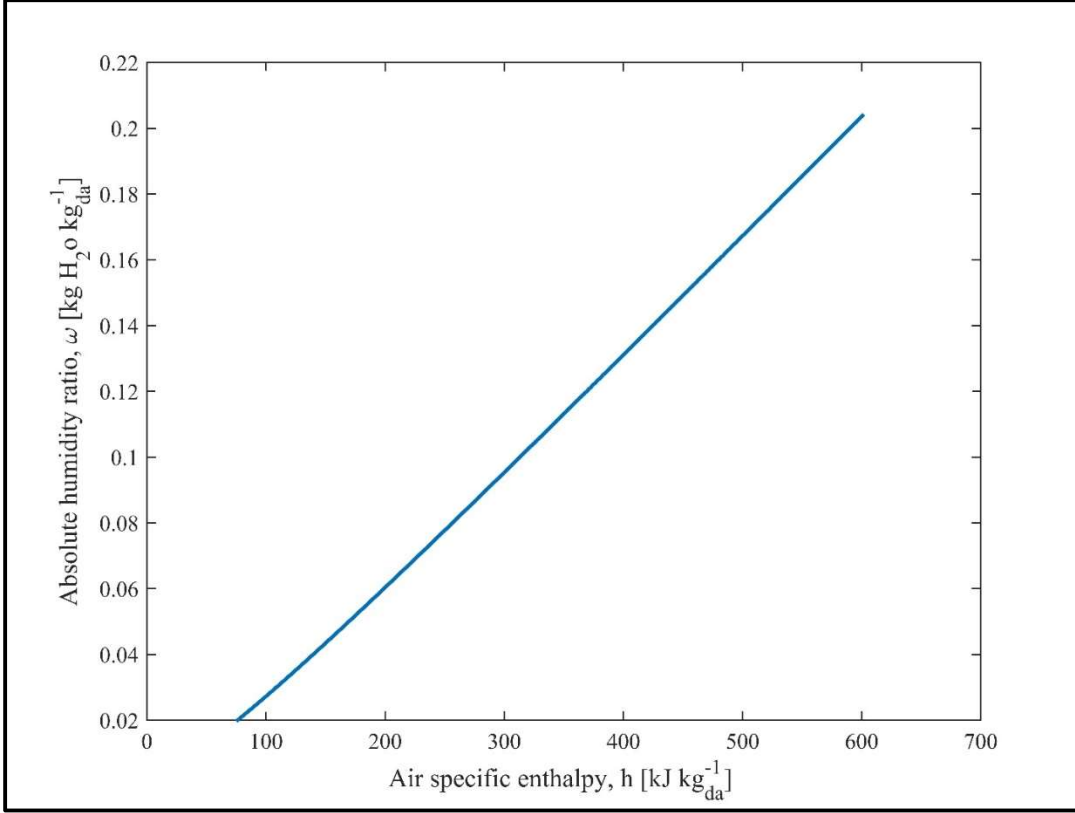


Figure 4.3. Absolute humidity ratio vs. specific enthalpy for saturated air.

The relation between absolute humidity ratio and specific enthalpy as shown in Fig. 4.3 is perfectly linear and hence, it can be fitted as follows,

$$\omega = \zeta_8 + \zeta_9 h \quad (4.45)$$

where,

$$\zeta_8 = -153.858138 \times 10^{-4}$$

$$\zeta_9 = 3.66008976 \times 10^{-4}$$

Applying Eq. (4.45), Eq. (4.44) becomes,

$$\varepsilon_{hum} MR \left(h_2 - \left(1 - \zeta_9 MR^{-1} (h_{a,2} - h_{a,1}) \right) h_{3,id} \right) = (h_{a,2} - h_{a,1}) \quad (4.46)$$

or,

$$\varepsilon_{hum} MR (h_2 - h_{3,id}) = (h_{a,2} - h_{a,1}) (1 - \varepsilon_{hum} \zeta_9 h_{3,id}) \quad (4.47)$$

Then the difference in absolute humidity ratio can be expressed as follows,

$$(\omega_{a,2} - \omega_{a,1}) = \frac{\varepsilon_{deh} MR (h_{1,id} - h_0)}{\zeta_9} \quad (4.48)$$

Applying Eq. (4.48) with Eq. (4.32), the mass flowrate of the freshwater produced is stated as,

$$\dot{m}_{pw} = \frac{\dot{m}_w}{MR} \frac{\varepsilon_{deh} MR (h_{1,id} - h_0)}{\zeta_9} \quad (4.49)$$

where,

h_0 and $h_{1,id}$ are defined in Eq. (4.19).

Subsequently, the mass flowrate for the rejected brine can be expressed as follows,

$$\dot{m}_b = \dot{m}_w \left(1 - \frac{\varepsilon_{deh} MR (h_{1,id} - h_0)}{MR \zeta_9} \right)$$

(4.50)

It is clear from Eqs. (4.43 and 4.47) that the only unknowns are $h_{a,1}$ and $h_{a,2}$, therefore,

these equations can be solved simultaneously to find $h_{a,1}$ and $h_{a,2}$ as follow,

$$h_{a,1} = \frac{\varepsilon_{deh} h_{a,1,id} + \varepsilon_{hum} (1 - \varepsilon_{deh}) h_{a,2,id}}{\varepsilon_{deh} + \varepsilon_{hum} (1 - \varepsilon_{deh})} = \frac{\varepsilon_{deh} \exp\left(\frac{T_0 - \zeta_0}{\zeta_1}\right) + \varepsilon_{hum} (1 - \varepsilon_{deh}) \exp\left(\frac{T_2 - \zeta_0}{\zeta_1}\right)}{\varepsilon_{deh} + \varepsilon_{hum} (1 - \varepsilon_{deh})} \quad (4.51)$$

$$h_{a,2} = \frac{\varepsilon_{deh} (1 - \varepsilon_{hum}) h_{a,1,id} + \varepsilon_{hum} h_{a,2,id}}{\varepsilon_{deh} (1 - \varepsilon_{hum}) + \varepsilon_{hum}} = \frac{\varepsilon_{deh} (1 - \varepsilon_{hum}) \exp\left(\frac{T_0 - \zeta_0}{\zeta_1}\right) + \varepsilon_{hum} \exp\left(\frac{T_2 - \zeta_0}{\zeta_1}\right)}{\varepsilon_{deh} (1 - \varepsilon_{hum}) + \varepsilon_{hum}} \quad (4.52)$$

The minimum and maximum temperature for the recirculated air $T_{a,1}$ and $T_{a,2}$ can be stated, respectively, as follow,

$$T_{a,1} = \zeta_0 + \zeta_1 \ln(h_{a,1}) = \zeta_0 + \zeta_1 \ln \left(\frac{\varepsilon_{deh} \exp\left(\frac{T_0 - \zeta_0}{\zeta_1}\right) + \varepsilon_{hum} (1 - \varepsilon_{deh}) \exp\left(\frac{T_2 - \zeta_0}{\zeta_1}\right)}{\varepsilon_{deh} + \varepsilon_{hum} (1 - \varepsilon_{deh})} \right) \quad (4.53)$$

$$T_{a,2} = \zeta_0 + \zeta_1 \ln(h_{a,2}) = \zeta_0 + \zeta_1 \ln \left(\frac{\varepsilon_{deh} (1 - \varepsilon_{hum}) \exp\left(\frac{T_0 - \zeta_0}{\zeta_1}\right) + \varepsilon_{hum} \exp\left(\frac{T_2 - \zeta_0}{\zeta_1}\right)}{\varepsilon_{deh} (1 - \varepsilon_{hum}) + \varepsilon_{hum}} \right) \quad (4.54)$$

4.3.4 Illustrative procedure

This part is made to describe the potential of the proposed analytical method and how to utilize it efficiently. This section provides a step-by-step procedure to show how to utilize the analytical method to model the CAOW HDH system. The input parameters for this system are seawater salinity, top temperature, bottom temperature, water-to-air mass flowrate ratio, and energy effectiveness for the humidifier and dehumidifier. The basic values for these variables are used as enlisted in Table 4.1. All the steps needed to apply the analytical model are summarized in the flowchart as in Figs. 4.4 – 4.6.

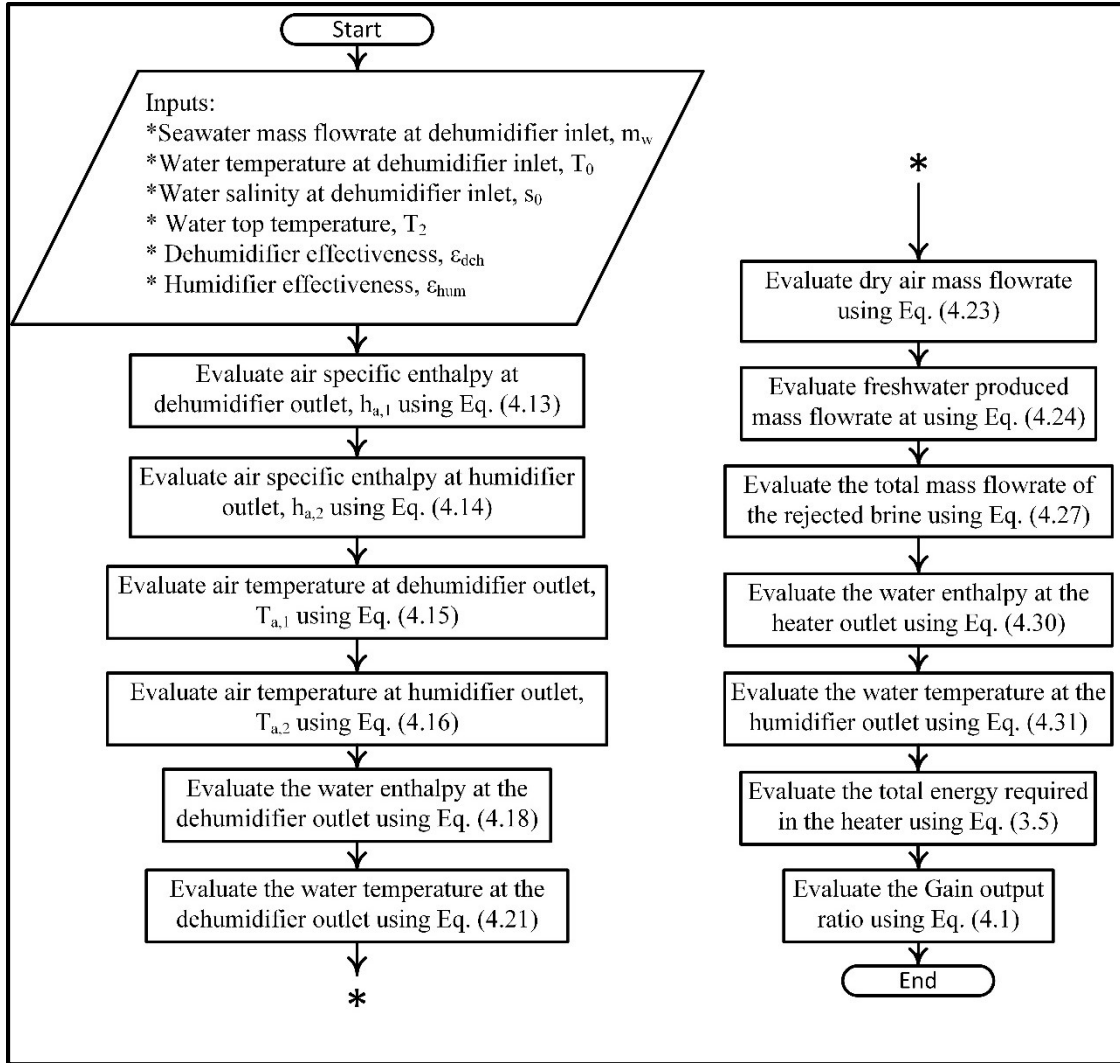


Figure 4.4. Analytical method algorithm flowchart for the balanced CAOW water-heated HDH system.

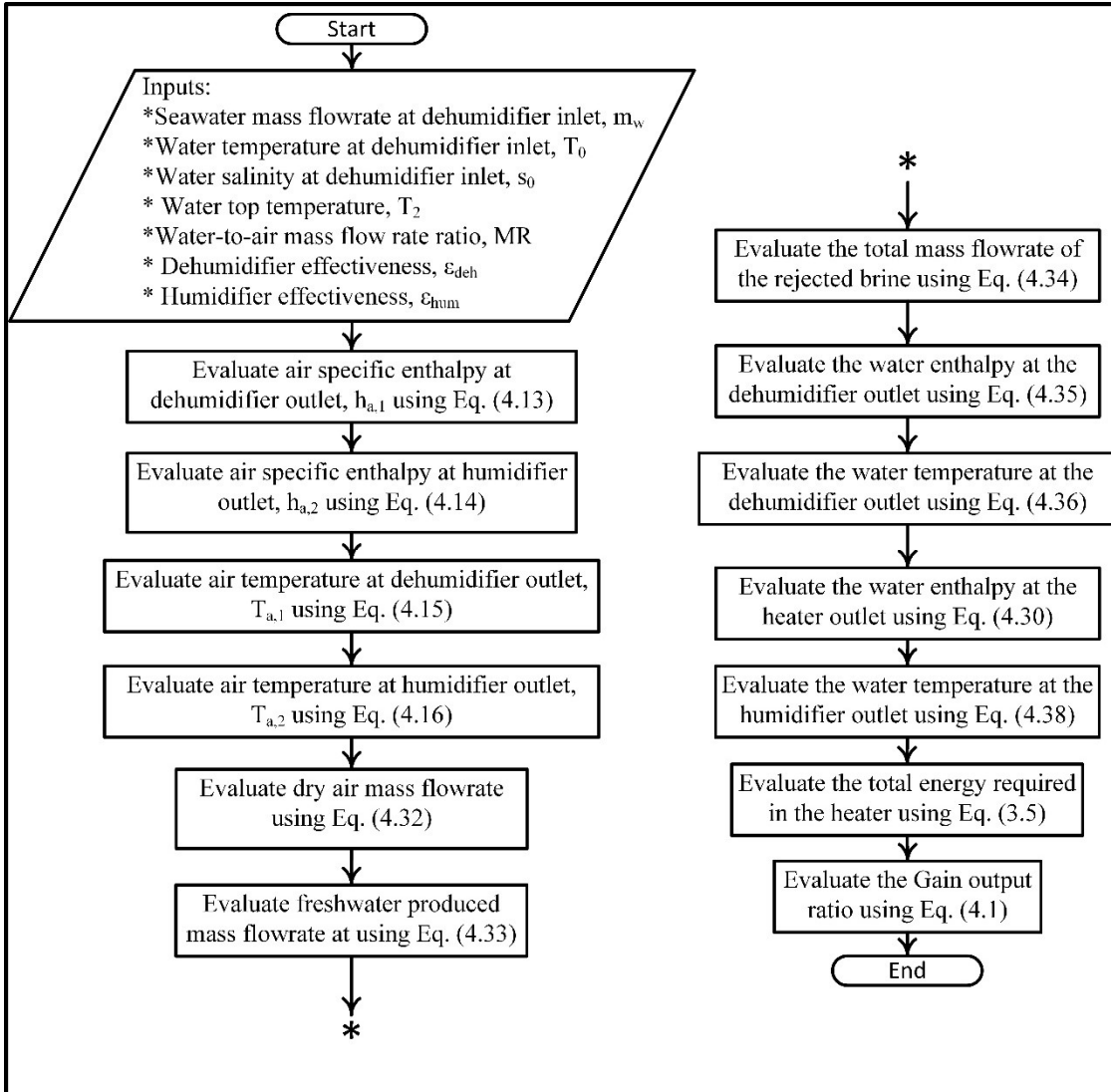


Figure 4.5. Analytical method algorithm flowchart for the unbalanced CAOW water-heated HDH system (Case 1).

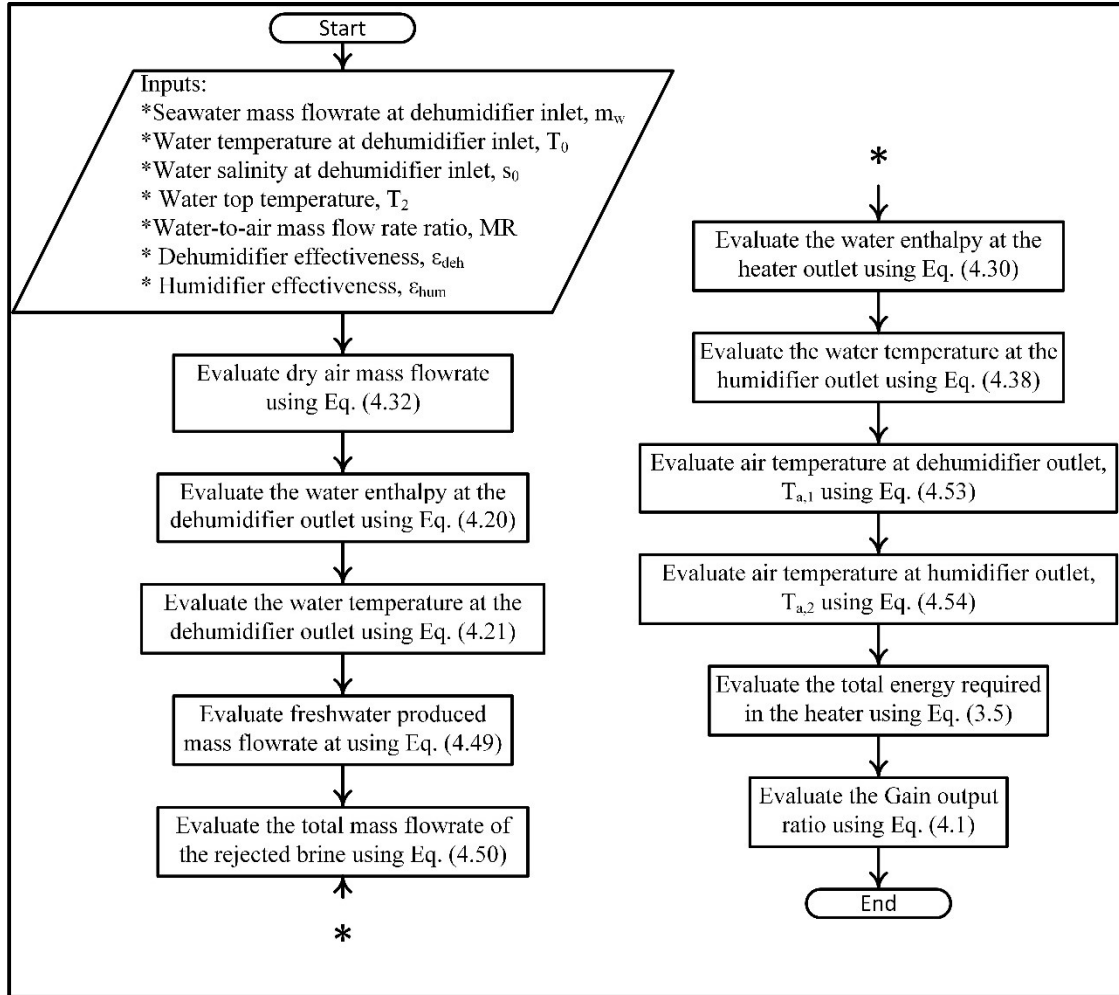


Figure 4.6. Analytical method algorithm flowchart for the unbalanced CAOW water-heated HDH system (Case 2).

4.4 Validation and performance of the proposed analytical model

4.4.1 Impact of humidifier energy effectiveness

Figures 4.7 and 4.8 represents the change in the cycle performance at different values of humidifier/dehumidifier effectiveness. From Fig. 4.7 it can be seen that the seawater temperature at humidifier inlet is set at 80 °C, the temperature of feed seawater temperature was set at 30 °C while energy effectiveness of the dehumidifier is set at 0.8. The mass flowrate ratio of seawater to air ranges from 1 - 4.5. It can be observed that there is an optimal point of mass flowrate ratio of seawater to air where the gained output ratio is maximum. Also, it is to be noted that the performance rises gradually as the energy effectiveness of the humidifier rises, ε_{hum} .

4.4.2 Impact of dehumidifier energy effectiveness

From Fig. 4.8 it can be seen that the seawater temperature at humidifier inlet is set at 80 °C, the temperature of feed seawater is set at 30 °C, and energy effectiveness of humidifier is set at 0.8. The trend of cycle performance of the cycle varies abruptly at higher energy effectiveness in the dehumidifier. This behavior is consistent with different values of seawater temperatures at the inlet of both dehumidifier and humidifier. Therefore, the high energy effectiveness of the dehumidifier impact is more than that of the humidifier on the cycle performance. In the literature [25], it was reported that the air humidity ratio at the dehumidifier outlet is more significant than that at the humidifier outlet. Therefore, based on these results, it can be said that for a CAOW W–H system the dehumidifier performance is more essential than the performance of the humidifier.

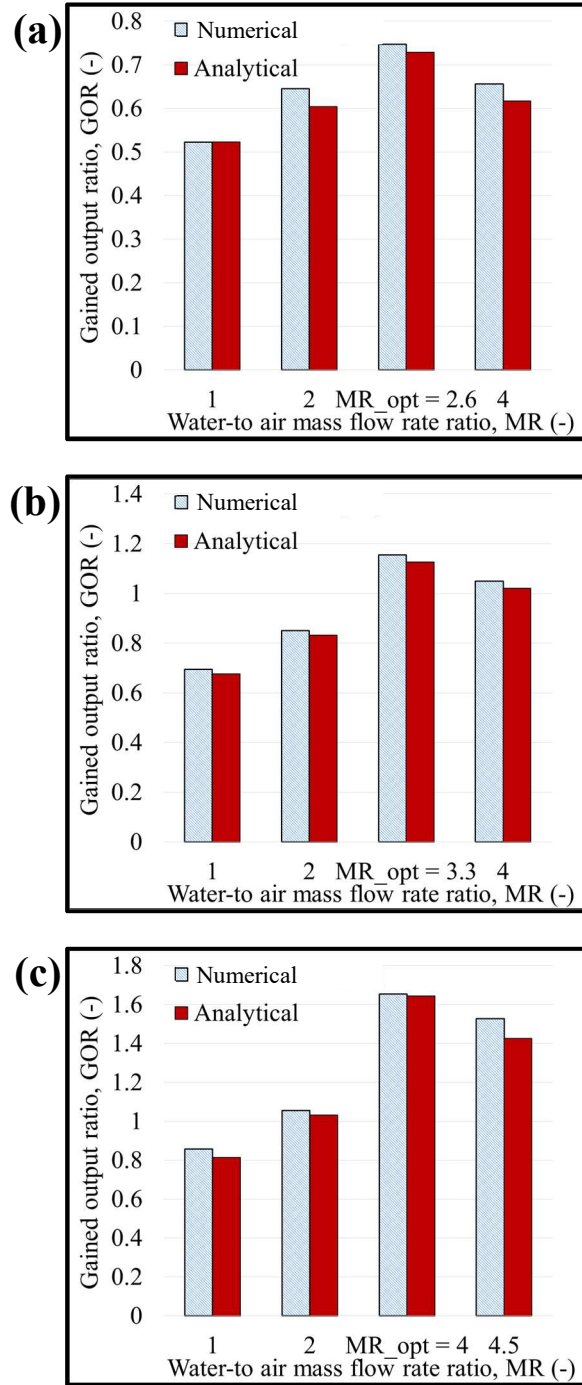


Figure 4.7. Impact of mass flowrate ratio and humidifier effectiveness; $T_{w,2} = 80\text{ }^{\circ}\text{C}$; $T_{w,0} = 30\text{ }^{\circ}\text{C}$; $\epsilon_{deh} = 80\%$; (a) $\epsilon_{hum} = 60\%$; (b) $\epsilon_{hum} = 80\%$; (c) $\epsilon_{hum} = 100\%$.

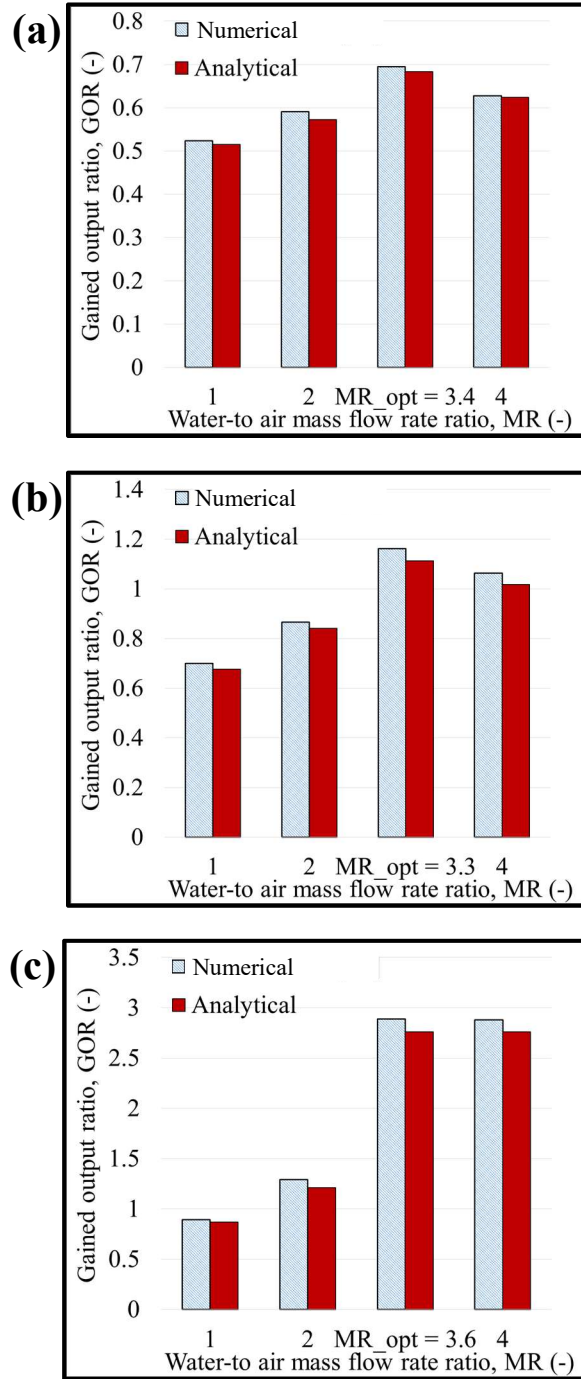


Figure 4.8. Impact of mass flowrate ratio and dehumidifier effectiveness; $T_{w,2} = 80\text{ }^{\circ}\text{C}$; $T_{w,0} = 30\text{ }^{\circ}\text{C}$; $\epsilon_{\text{hum}} = 80\%$; (a) $\epsilon_{\text{deh}} = 60\%$; (b) $\epsilon_{\text{deh}} = 80\%$; (c) $\epsilon_{\text{deh}} = 100\%$.

4.4.3 Impact of water top temperature

Figure 4.9 depicts the impact of seawater heating temperature on the GOR of the cycle. the bottom temperature of seawater ($T_{w,0}$) is set at 30 °C and the energy effectiveness of both dehumidifier and humidifier are set at 0.92. The seawater heating temperature ($T_{w,2}$) is changed using a range of 60 - 80 °C. The optimum ratio of mass flowrate rises as the heating temperature rises. However, this behavior is governed basically by the energy effectiveness of both the dehumidifier and humidifier. At low values of energy effectiveness, the heating temperature has little or no impact on the GOR of the system. This outcome is contrary to intuition. Though, it can be clarified utilizing the concept of HCR. It is also to be mentioned that generated entropy through mass/heat exchangers is minimum when $HCR = 1$ (i.e. optimum mass flowrate ratio). This understanding can be used to comprehend the trends obtained in Fig. 4.9 at various top water temperatures.

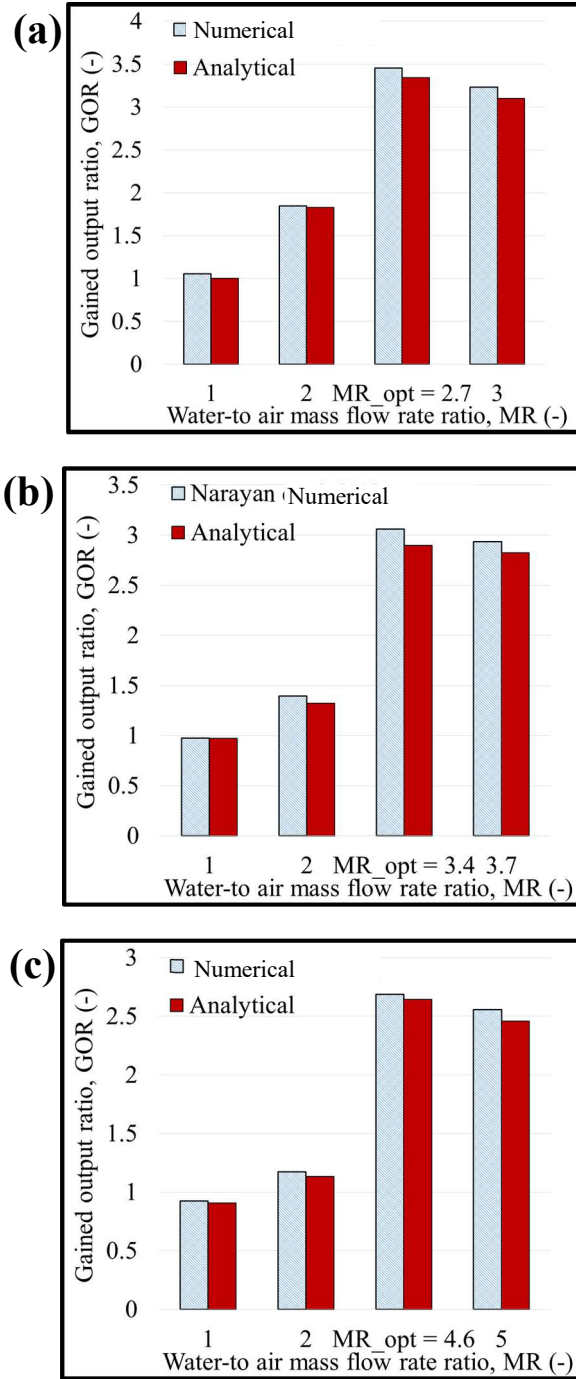


Figure 4.9. Impact of mass flowrate ratio and top water temperature; $T_{w,0} = 30\text{ }^{\circ}\text{C}$; $\varepsilon_{\text{hum}} = \varepsilon_{\text{deh}} = 92\%$; (a) $T_{w,2} = 60\text{ }^{\circ}\text{C}$; (b) $T_{w,2} = 70\text{ }^{\circ}\text{C}$; (c) $T_{w,2} = 80\text{ }^{\circ}\text{C}$.

4.4.4 Impact of water bottom temperature

The temperature at the dehumidifier inlet ($T_{w,0}$) is determined based on the temperature of the ambience of the area from which the seawater is taken. From figure 4.10 (the temperature of heating is 80 °C and humidifier/dehumidifier effectiveness is 0.92. At high values of feed seawater temperature, higher values of GOR are achieved (refer to Fig. 4.10). These findings could be explained using HCR. The level of optimizing the humidifier at the balancing conditions (i.e. optimum water-to-air mass flowrate ratio) for the performance declines as the feed seawater temperature drops. Therefore, the humidifier irreversibilities rise with the drop of feed seawater temperature, and hence, the performance decreases.

4.4.5 Validation of the proposed model

The aforementioned analytical model is tested and validated against the numerical method reported in chapter 3. An assessment of the gained output ratio was evaluated utilizing the proposed analytical model and the numerical method as illustrated in Figs. 4.7 – 4.10. Good agreements are noticed between the analytical and numerical model, the error is around $\pm 5\%$.

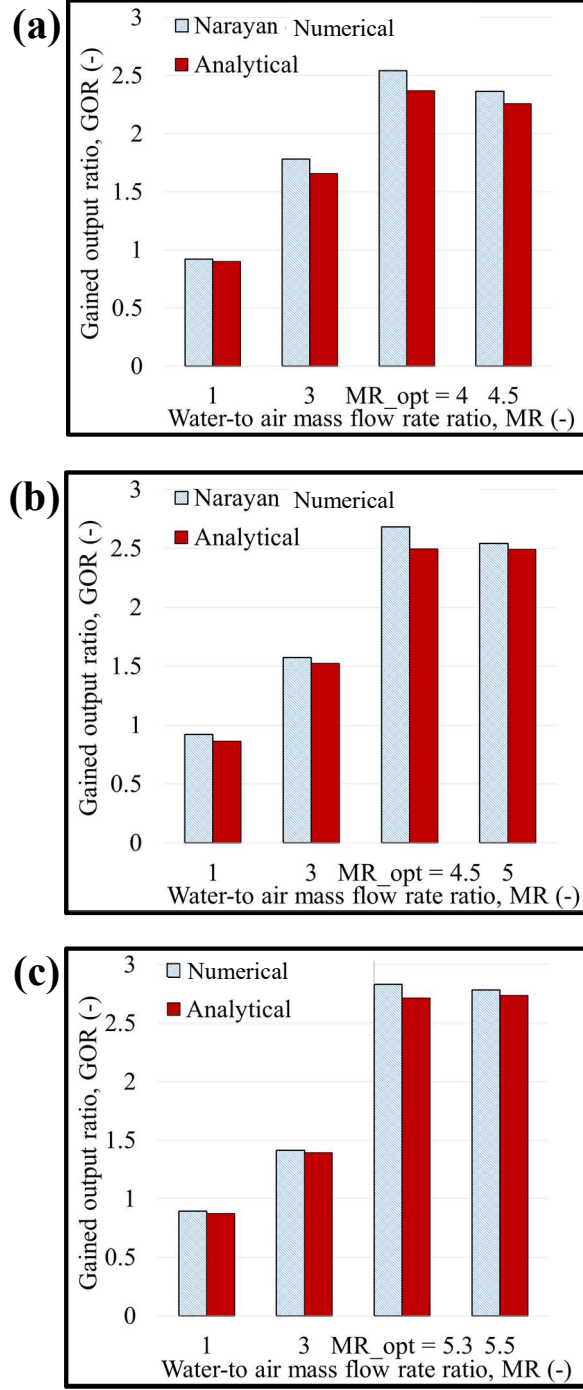


Figure 4.10. Impact of mass flowrate ratio and bottom water temperature; $T_{w,2} = 80\text{ }^{\circ}\text{C}$; $\varepsilon_{\text{hum}} = \varepsilon_{\text{deh}} = 92\%$; (a) $T_{w,0} = 30\text{ }^{\circ}\text{C}$; (b) $T_{w,0} = 35\text{ }^{\circ}\text{C}$; (c) $T_{w,0} = 40\text{ }^{\circ}\text{C}$.

4.5 Concluding remarks

In this chapter, an innovative method based on energy effectiveness as a performance parameter is suggested for modeling the closed-air open-water water-heated humidification-dehumidification desalination system at balanced and unbalanced states. The proposed method target evading the difficulty of using the library of thermos-physical properties. The proposed analytical model is certified with the numerical data reported in chapter 3. A benchmark of the gained output ratio (GOR) evaluated by using the current analytical model is conducted with the chapter 3 model as shown in Figures 4.7 – 4.10. Good agreement with the numerical model is observed with an error of around $\pm 5\%$.

CHAPTER 5

THERMODYNAMIC BALANCING USING THE PROPOSED

PINCH-BASED EFFECTIVENESS DEFINITION

Summary

The roadblock to commercialize the HDH system is that the system performance is relatively modest. In this chapter, thermodynamic balancing in the system under study is proposed by using mass extraction and injection technique. These methods can be regarded as a prospective means to lessen power consumption which dramatically results in higher performance. Thermodynamic balancing reduces the entropy production in both the humidifier and dehumidifier by balancing air-to-water concentration and temperature difference through both towers. The proposed (single extraction) system is a modified (zero extraction) system (Fig. 3.1) with applying mass extraction and injection between the humidifier and the dehumidifier. The mathematical procedure to model the system was drawn to investigate the impact of the injection and extraction technique on the performance of the desalination system. It was concluded that at $\varepsilon = 95\%$; the desalination system performance with one extraction is 83.1 % better than that without extraction.

5.1 Introduction

An optimum design system is one that produces less entropy at the same size (capital cost). The efficiency of such systems is optimized by using finite-time thermodynamics. In this chapter, the concept of thermodynamic balancing is applied to the desalination process of the system under study (dehumidifier and humidifier) to improve the performance of the HDH desalination system. In the HDH system, the carrier air shuttles heat and moisture between the humidifier and dehumidifier. It is found as mentioned before that reduction of the entropy production in the system under study improves the performance (GOR) of the system.

Also, it is found that the injection and extraction of the humid air to the dehumidifier from the humidifier can convey the rate of enthalpies into balancing and lower the generated entropy. In this chapter, extensive analysis is reported to investigate the impact of mass injection and extraction on the HDH cycle. This analysis depends on the idea that at constant dehumidifier and humidifier effectiveness, the HDH system operates optimally for any given operating conditions at sole water-to-air mass flowrate ratio. Design algorithms are developed to balance the desalination process in the HDH cycles.

5.2 CAOW water heated humidification dehumidification arrangement under thermodynamic balancing (extraction)

The water heated CAOW humidification dehumidification cycle in section 3.2 in addition to a consideration of thermodynamical balancing (extraction at one or multiple points) between the cycle humidification and dehumidification sides, as presented in Figure 5.1.

The system comprises a seawater heater, humidifier, and dehumidifier. Significant to mention that the proposed effectiveness method is utilized to estimate the gained-output-ratio (GOR) in the current chapter as an alternate technique to the dehumidifier/humidifier other performance indices.

The key variance concerning the zero-extraction framework and presented one (with extraction) is the multi-extraction. The air to be taken out from the humidification part should be inserted into the dehumidification part to a certain location that has similar specific enthalpy as the point of extraction to evade irreversibilities because of injecting the extracted air into the dehumidifier (mixing). Figure 5.1 expresses extracted air state ' a,ex ' for single points extraction. Where state ' a ' represents recirculated air, ' ex ' denotes extraction.

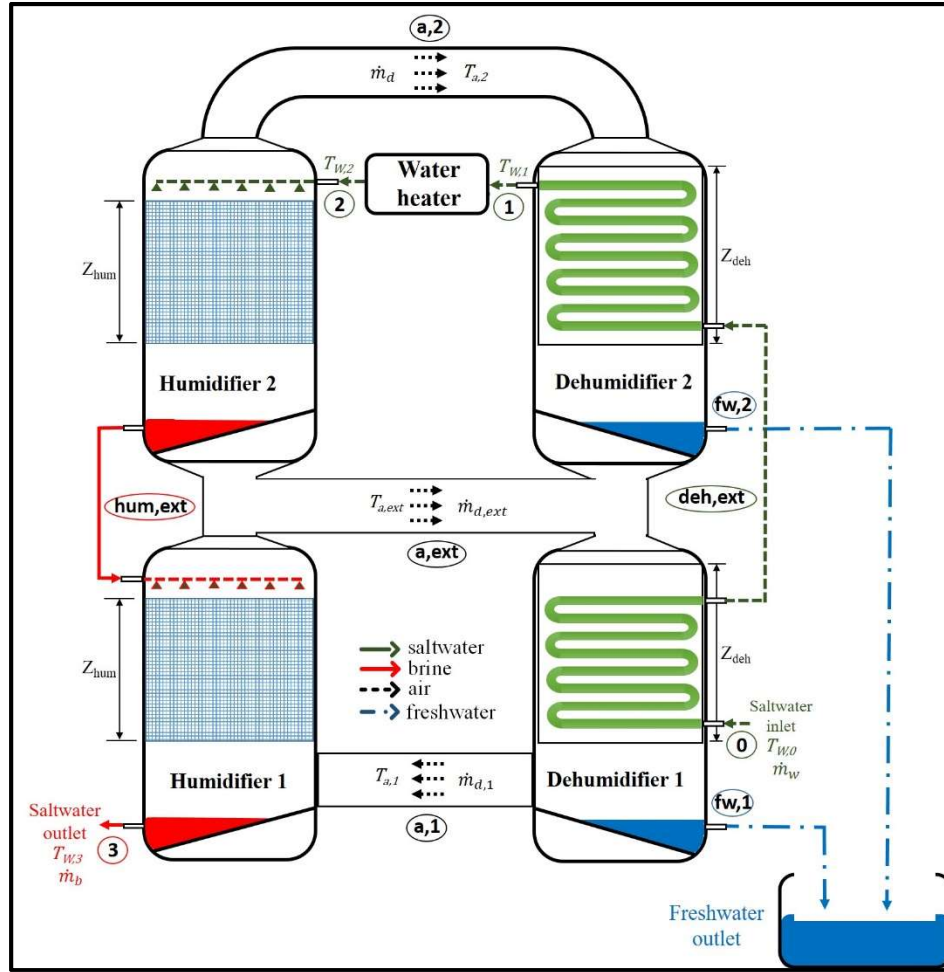


Figure 5.1. Water heated, CAOW-WH humidification dehumidification configuration under thermodynamical balancing (single extraction) [56].

5.3 Thermal balance in the humidifier and dehumidifier

Entropy generation occurs because of the limited area of heat and transfer area in thermal system components. In the CAOW-WH HDH cycle, most of the thermal operations occur in the humidification/dehumidification process. Therefore, a major part of the entropy is generated in the humidifier and dehumidifier. The generation of entropy in the dehumidifier and humidifier has been highlighted to increase HDH cycle performance. The mathematical model that is developed in chapter 3 is reconsidered and prolonged for single balancing by extraction and injection. In this section, for humidifier and dehumidifier modeling, pinch-based energy effectiveness is used as an appropriate alternative to temperature pinch [27], [50], [51] and enthalpy pinch [53]–[55].

5.3.1 Control volume balance – optimum operation (zero extraction)

As stated earlier, a balanced cycle can be regarded as a cycle that has a minimum generated entropy at fixed energy effectiveness when the modified heat capacity ratio = 1. The aforementioned definition can be prolonged to the pinch-based energy effectiveness as an alternative to terminal-based energy effectiveness as in,

$$HCR = \frac{\Delta\dot{H}_{max,w}}{\Delta\dot{H}_{max,air}} \quad (5.1)$$

From Eq. (5.1), for HCR to be equal unity in the dehumidifier, $L_w = L_{air}$ is needed (varying mass flowrate ratio of seawater to air). It should be noted, these finite size losses are located at the inlet and outlet of the dehumidifier, and a single intermediate location in the humidifier.

5.3.2 Mass injection- and extraction-based (single extraction)

From the system discussed in the previous section (control volume balancing), the heat capacity ratio is conveyed to a value of 1 by changing the mass flowrate ratio of seawater to air. For such a situation, HCR is not equal to unity at every location in the mass/heat exchange device. Nevertheless, the method of thermal balancing by operating the system at the optimal operational condition can be lengthened by changing the mass flowrate ratio of seawater to air along both the dehumidifier and humidifier employing extracting and injecting the recirculated air to bring heat capacity ratio to a value of one. However, to achieve this level of thermodynamic balancing, the number of extraction and injection points should incline to infinity which is completely impossible. On the other hand, a distinct number of injection and extraction points is doable. It is to be noted that applying a single extraction and injection technique to enhance HDH system performance is comprehensively examined in this chapter.

Local pinch-based energy effectiveness along the dehumidifier using zero and single injection and extraction is illustrated in Fig. 5.2. Figure 5.2 shows that applying a single injection and extraction conveyed the energy effectiveness to maximum ($HCR = 1$) at the injection location in addition to the terminal points that result in a huge drop in the generated entropy (refer to Fig. 5.3). The main behavior for the system with and without extraction is that the generated entropy rises with the decrease of pinch-based energy effectiveness. The generated entropy of the system with one extraction is less than that of a system without extraction up to a certain level of energy effectiveness after which the

extraction technique has zero impact. The concept behind this trend is elaborated at extent in the results and discussion section.

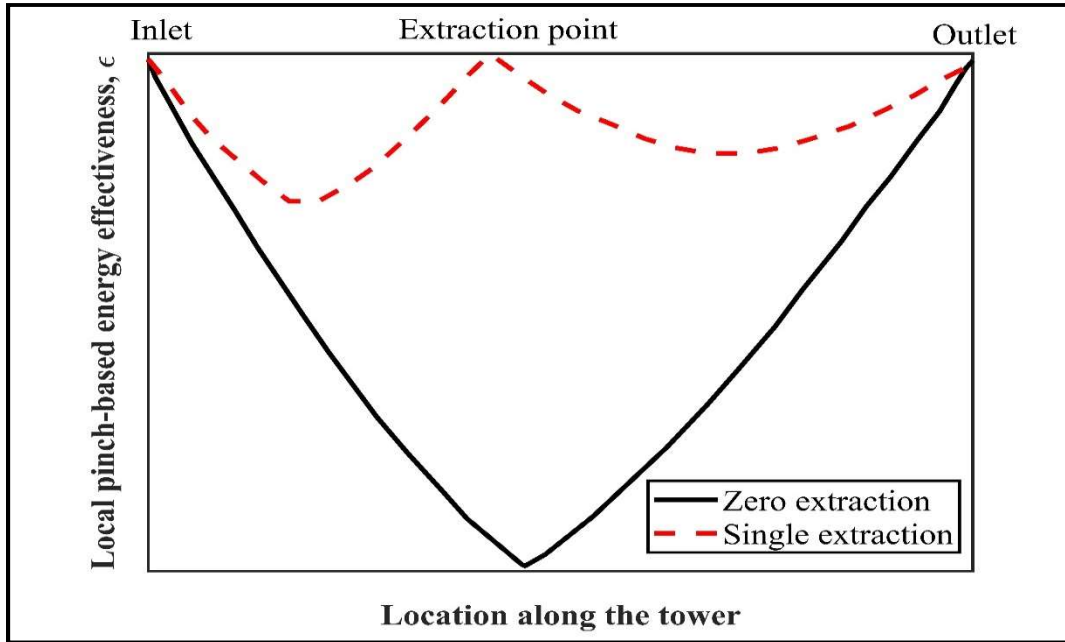


Figure 5.2. Local energy effectiveness in HDH's dehumidifier with and without applying extraction.

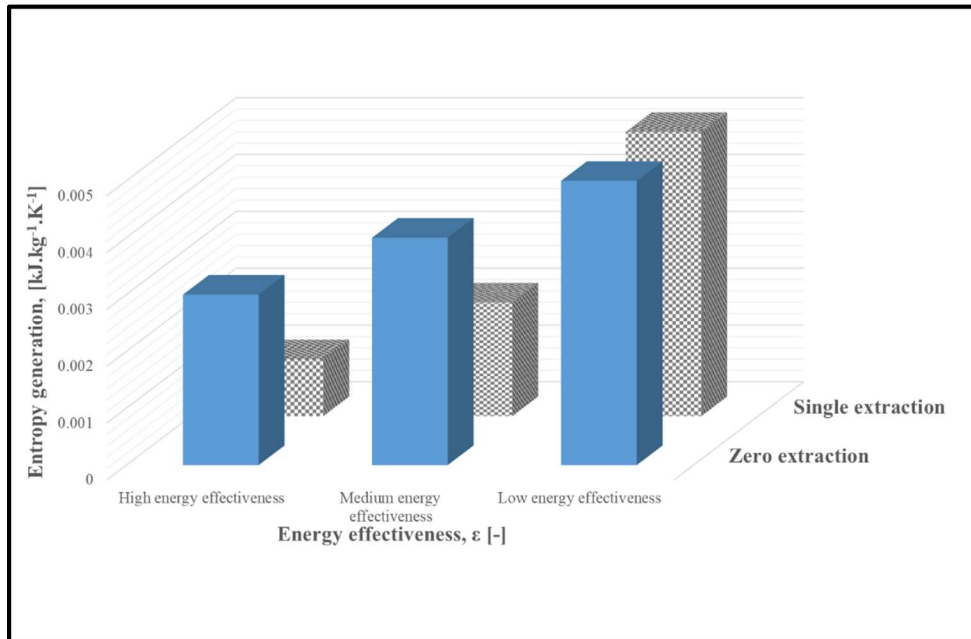


Figure 5.3. Impact of applying extraction in reducing the generated entropy.

5.4 Modeling of the proposed model

The concepts of thermal balance established for mass and heat exchange devices are utilized and in the HDH desalination system using the proposed pinch-based energy effectiveness. The arrangement of the cycle to be investigated is shown in section 5.2 (please refer to Fig. 5.1).

5.4.1 Zero extraction cycle

To evaluate the mass flowrate ratio of seawater to air, this association with the slope of seawater line '0 - 1' and '2 - 3' in the dehumidification and humidification devices, respectively, as in the temperature-enthalpy diagram (Fig. 3.2) can be used. Figure 5.4 represents a CV that comprises a heater, a section from the humidifier, and the corresponding section in the dehumidifier. The balance of energy on this CV that is shown in Fig. 5.4 can be stated as:

$$(\Delta\dot{H})_w - (\Delta\dot{H})_{fw} = (\Delta\dot{H})_{da} \quad (5.2)$$

or,

$$\dot{m}_w(C_p\Delta T)_w - \dot{m}_{fw}(C_p\Delta T)_{fw} = \dot{m}_{da}(\Delta h^*)_{da} \quad (5.3)$$

$$(\dot{m}_w - \dot{m}_{pw})C_{p,ave}\Delta T = \dot{m}_{da}(\Delta h^*)_{da} \quad (5.4)$$

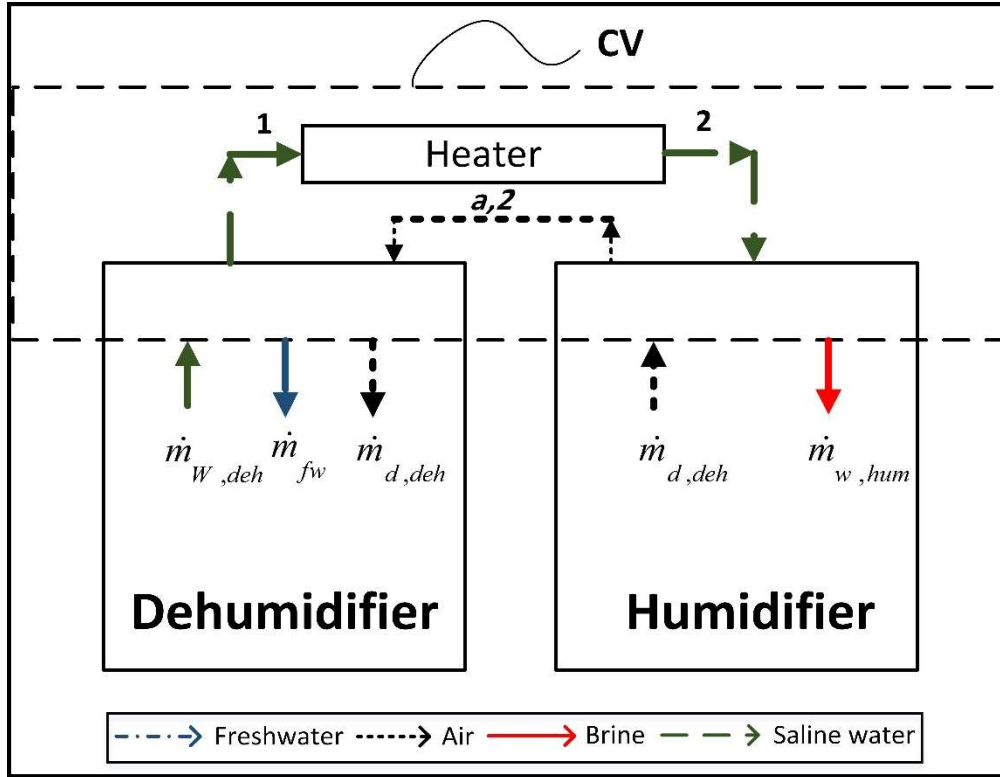


Figure 5.4. A CV comprising the seawater heater, part of the humidifier, and the equivalent part of the dehumidifier [40].

Assuming Δh^* approaches zero, the relation between temperature derivative (slope of seawater process) and mass flowrate ratio of seawater to air can be stated as follows,

$$\frac{dT}{dh^*} = \frac{1}{MRC_p} \quad (5.5)$$

Based on Eq. (5.5), and inclusive algorithm to estimate the humidification dehumidification cycle performance by using the proposed pinch-based energy effectiveness using top seawater temperature, the bottom seawater temperature, and energy effectiveness of both the dehumidifier and humidifier as input data is presented in Fig. 5.5.

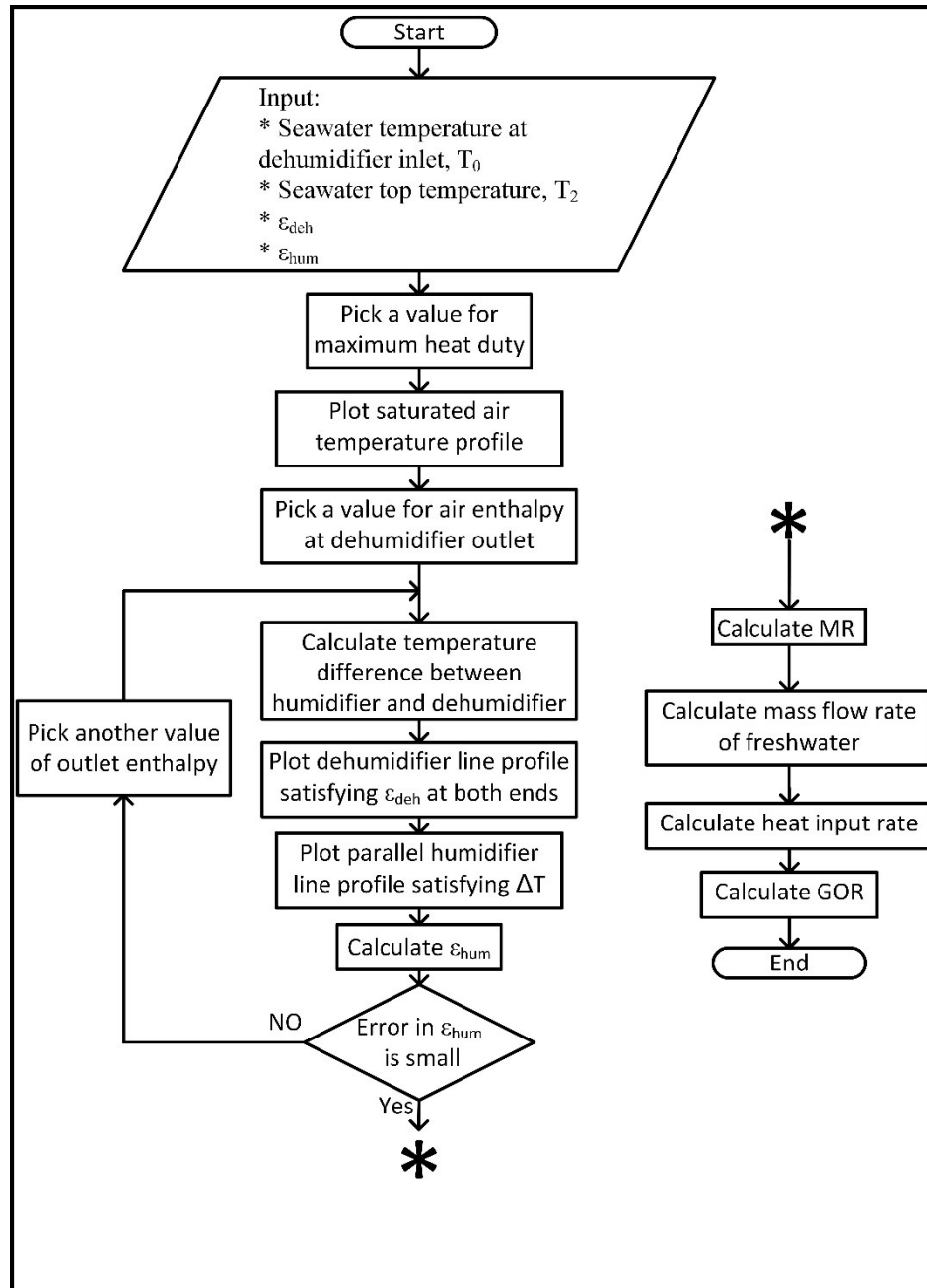


Figure 5.5. Zero extraction algorithm flowchart.

5.4.2 Single extraction cycle

Figure 5.6 illustrates a temperature-enthalpy chart of a system with a single injection and extraction. In this case, the air is extracted from the humidifier at the state 'ex' and is injected to the corresponding location in the dehumidifier with the same state 'ex' to avoid generating additional entropy due to mixing. The foremost dissimilarity between the system with and without extraction is that the thermodynamic model is estimated as double stages. The conception settled in this chapter leads generally to generate Fig. 5.6. as picturing and coding of this cycle depends on Fig. 5.6.

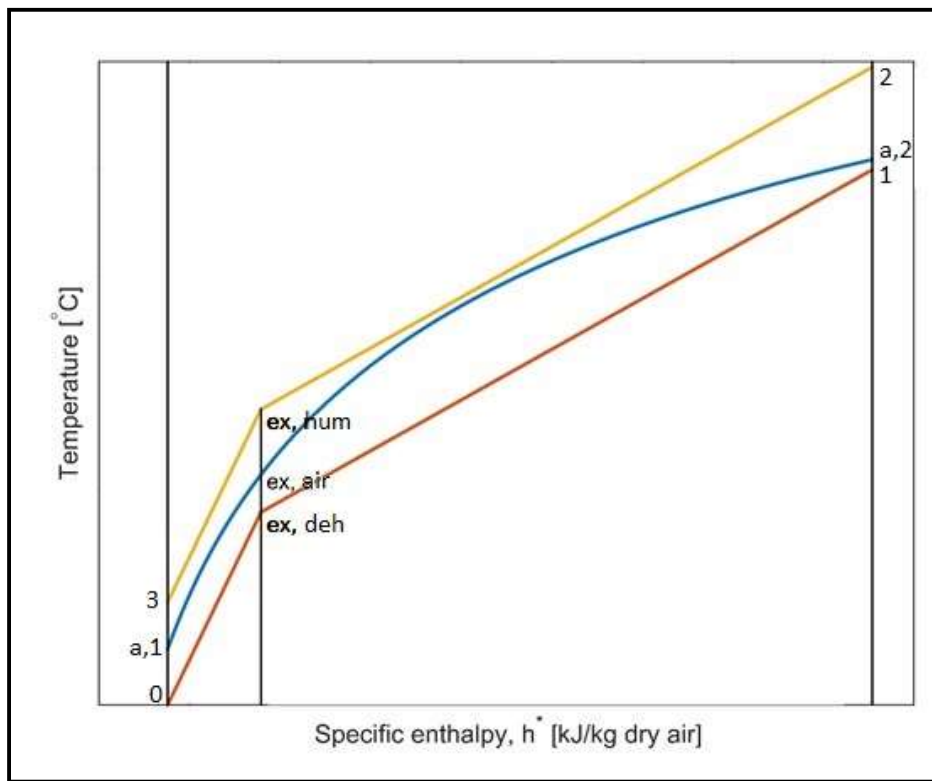


Figure 5.6. T - h chart of CAOW – WH HDH desalination cycle after applying the injection and extraction technique [56].

The concept is translating this figure to a thermodynamic process for estimating the single extraction system performance. The elaborated process for modeling the HDH cycle with one extraction is illustrated in Fig. 5.7.

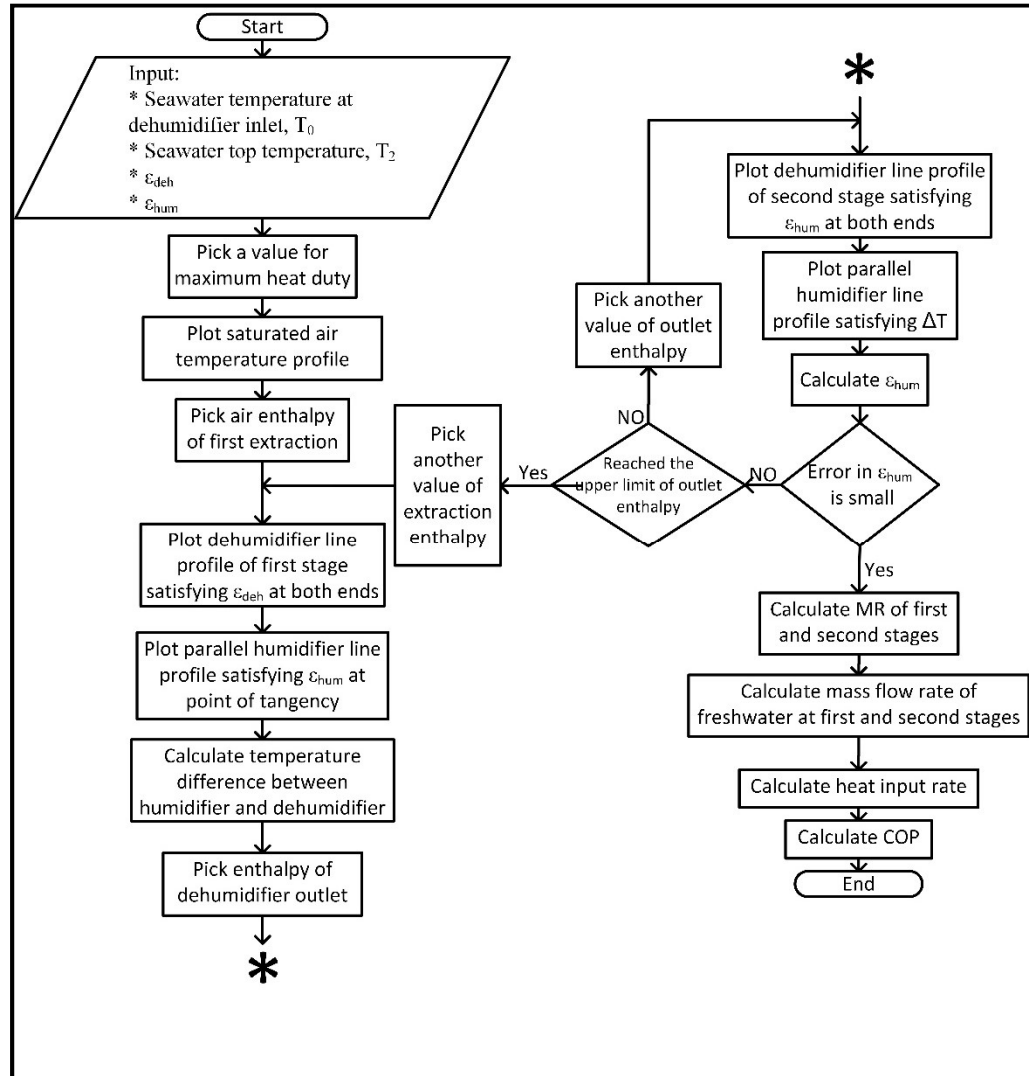


Figure 5.7. Single extraction algorithm flowchart.

5.4.3 Operational variables

The inputs that affect extraction are as follow, the bottom seawater temperature at the dehumidifier inlet, the top seawater temperature, the humidifier, and dehumidifier energy effectiveness. Basic and range values of these variables are presented as shown in Table 4.1.

5.5 Results and discussion

5.5.1 Impact of energy effectiveness

Figure 5.8 represents the impact of energy effectiveness on the CAOW - WH HDH cycle performance by utilizing the proposed energy effectiveness with zero and single extractions. It can be seen from this figure the general trend is that the GOR increases as energy effectiveness increases for the HDH cycle with and without applying extraction.

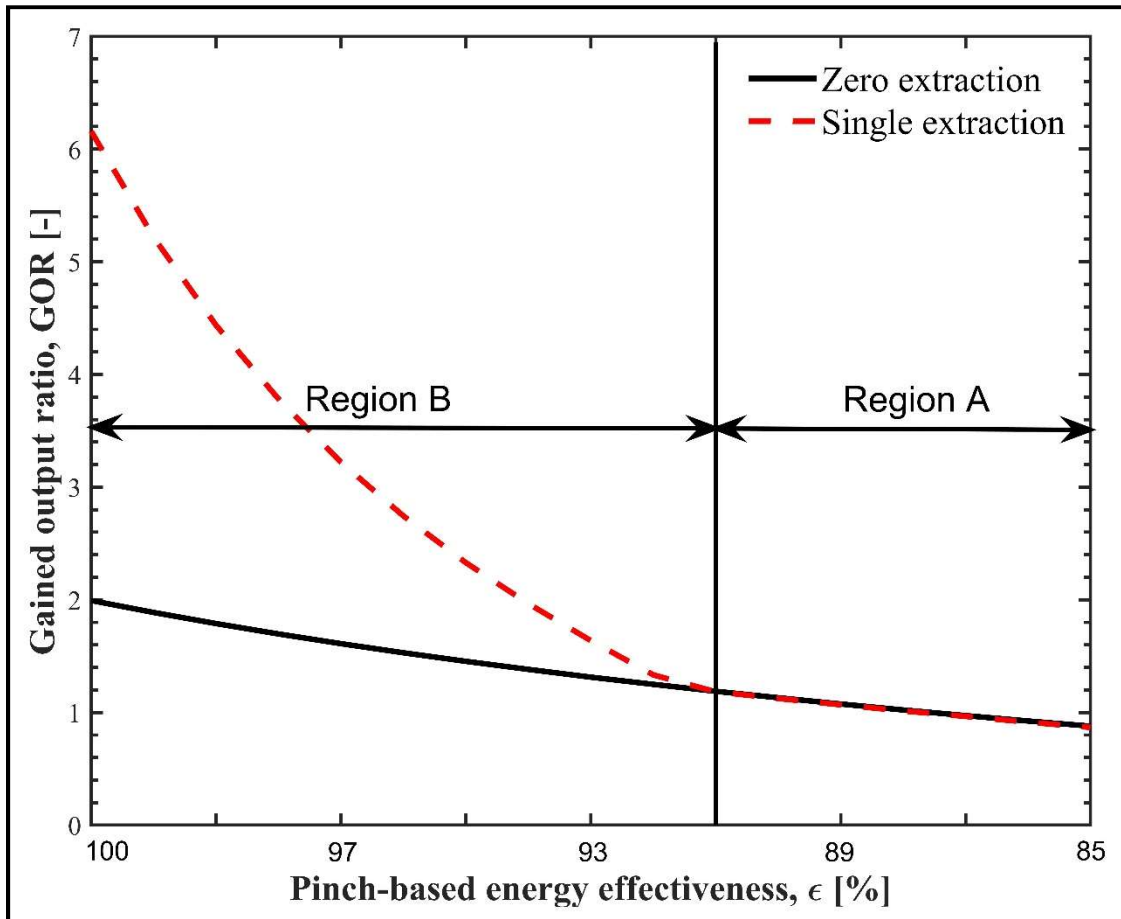


Figure 5.8. Impact of energy effectiveness on GOR.

This is because energy effectiveness is equivalent to performance. Segment “A” can be regarded as the low effectiveness segment (energy effectiveness span is 85 - 91%) for the humidifier and dehumidifier.

The system with one injection and extraction performance is more in region ‘B’ than that without applying injection and extraction for the CAOW – WH HDH desalination cycle. The GOR starts with a value slightly above 1 and then increases to a GOR of 6.2. The decrease of the generated entropy because of applying thermodynamic balancing related to single extraction results in an enormous GOR rise.

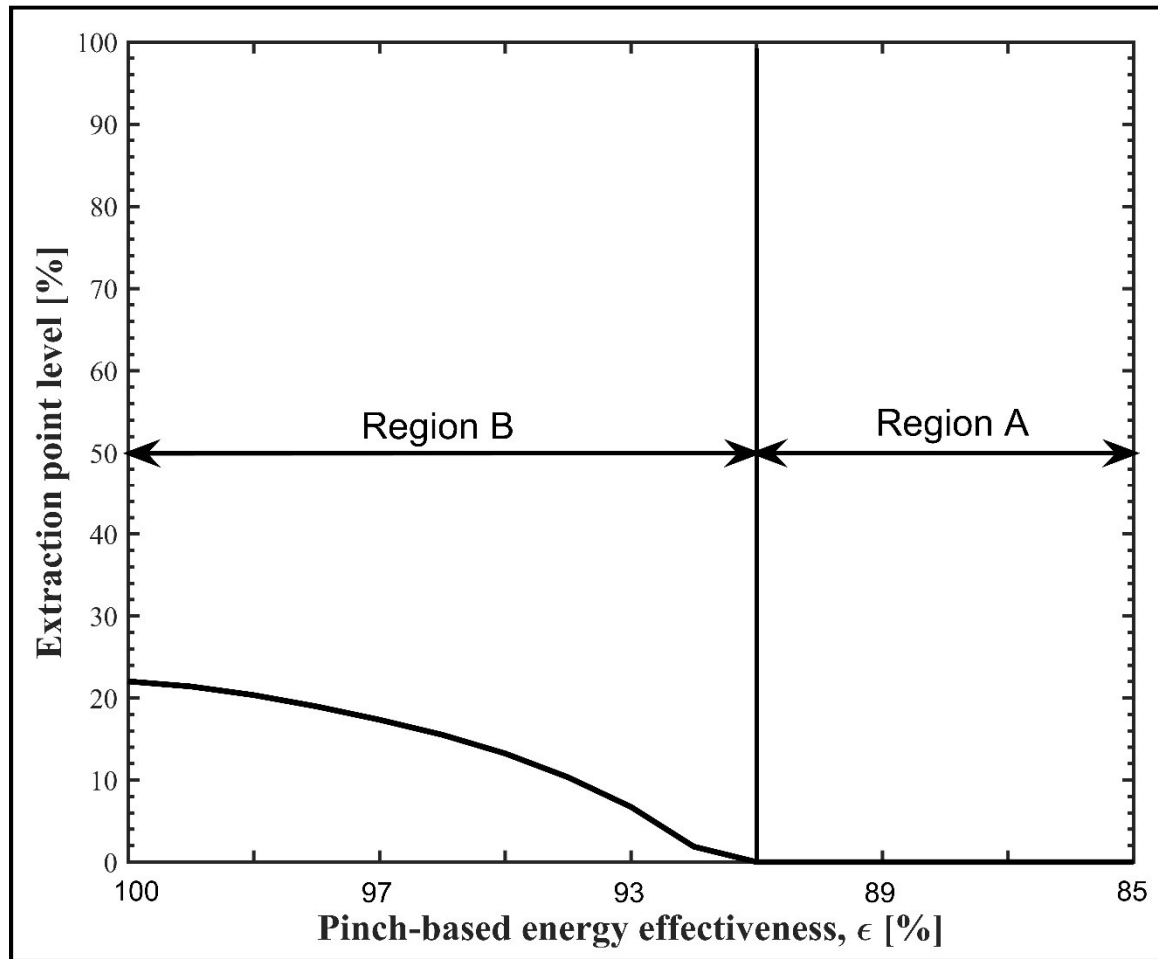


Figure 5.9. Impact of energy effectiveness on extraction level location.

In segment “A” the CAOW – WH HDH cycle performance using zero and single injection and extractions overlaps i.e. the injection and extraction technique has zero impact before the critical level of energy effectiveness at 91%. For this region, the energy effectiveness starts from 91%. The is that for the cycle with one extraction, the air is injected and extracted at a level between the top and bottom of the humidifier; as energy effectiveness decreases, the location of injection and extraction is lowering down until it merges at $\varepsilon = 91\%$ with the air inlet. In other words, the system with one extraction converts to a system without extraction as illustrated in Fig. 5.9. These data are shortened in Table 5.1 to specify the appropriate operating conditions of low and high effectiveness region.

Table 5.1. Suggested arrangement of high and low effectiveness segments.

Region	Effectiveness (%)	Recommended configuration
A	< 91	Zero Extraction
B	100 - 91	Single Extraction

From Fig. 5.8 it is important to emphasize that the thermodynamic balancing technique with single extraction is only feasible at high values of energy effectiveness ($> 91\%$) which is difficult to achieve in the dehumidifier and almost impossible in the humidifier. However, this figure is misleading for two reasons, first, it is generated assuming both the dehumidifier and humidifier are having an equal value of energy effectiveness. Second, these values (for example 91%) are pinch-based energy effectiveness i.e. if it is converted to the terminal (conventional) definition, it is 35%, please refer to Fig. 5.10.

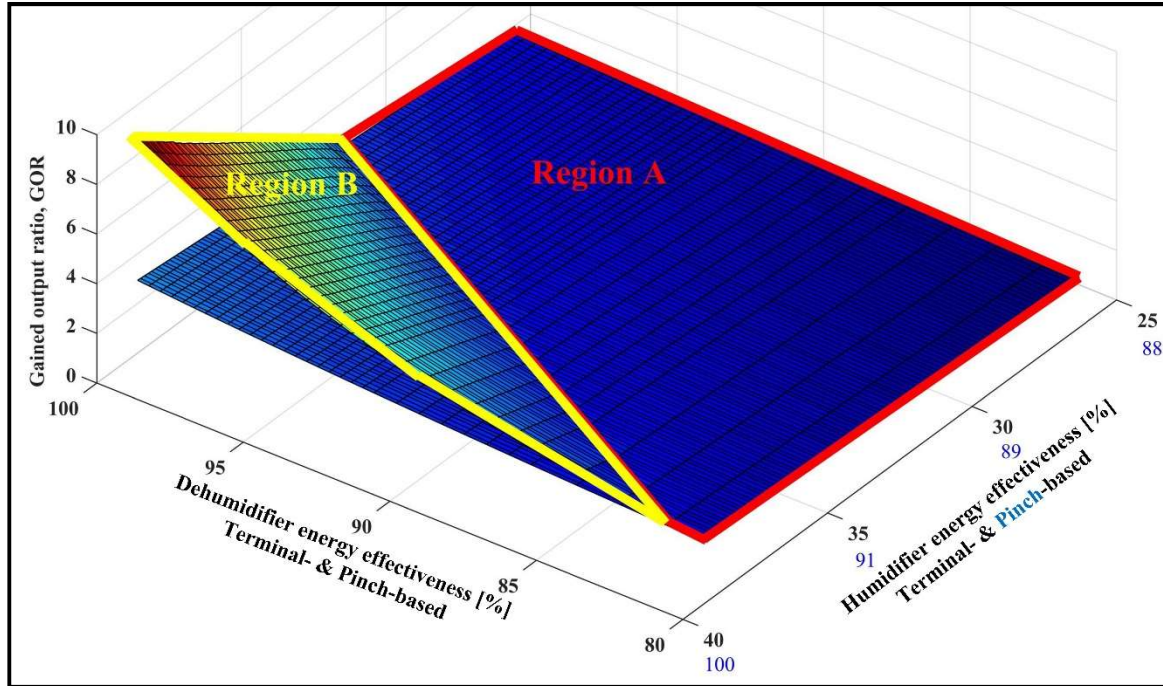


Figure 5.10. Impact of energy effectiveness on GOR.

To solve the aforementioned issues, Fig. 5.8 is regenerated as shown in Fig. 5.10 as a function in both the dehumidifier and humidifier energy effectiveness with terminal- and pinch-based definition. Fortunately, the figure shows that the extraction technique is viable in region ‘B’ where humidifier and dehumidifier terminal-based energy effectiveness ranges between 82% - 100% and 32% - 40%, respectively. These values are achievable by both system components, please refer to Table 5.2.

Table 5.2. Suggested arrangement of high and low Terminal-based effectiveness segments.

Region	ϵ_{deh} (%)	ϵ_{hum} (%)	Recommended configuration
A	< 82	< 32	Zero Extraction
B	100 - 82	40 - 32	Single Extraction

5.5.2 Impact of seawater temperature at inlets of humidifier/dehumidifier

Figure 5.11 depicts the impact of seawater maximum temperature on CAOW - WH cycle using zero and single injection and extractions at 100% energy effectiveness. As top seawater temperature rises, the system performance reduces for both bottom temperatures. As the temperature of the seawater at the dehumidifier inlet is set at 20 and 30 °C. The rise in top seawater temperature has been noted to elevate the required input heat rate. Therefore, the system performance decreases consequently. Additionally, the system performance at 30 °C is lower than that at 20 °C. This can be attributed to the capability of seawater to cool down and condensate extra air in the dehumidifier.

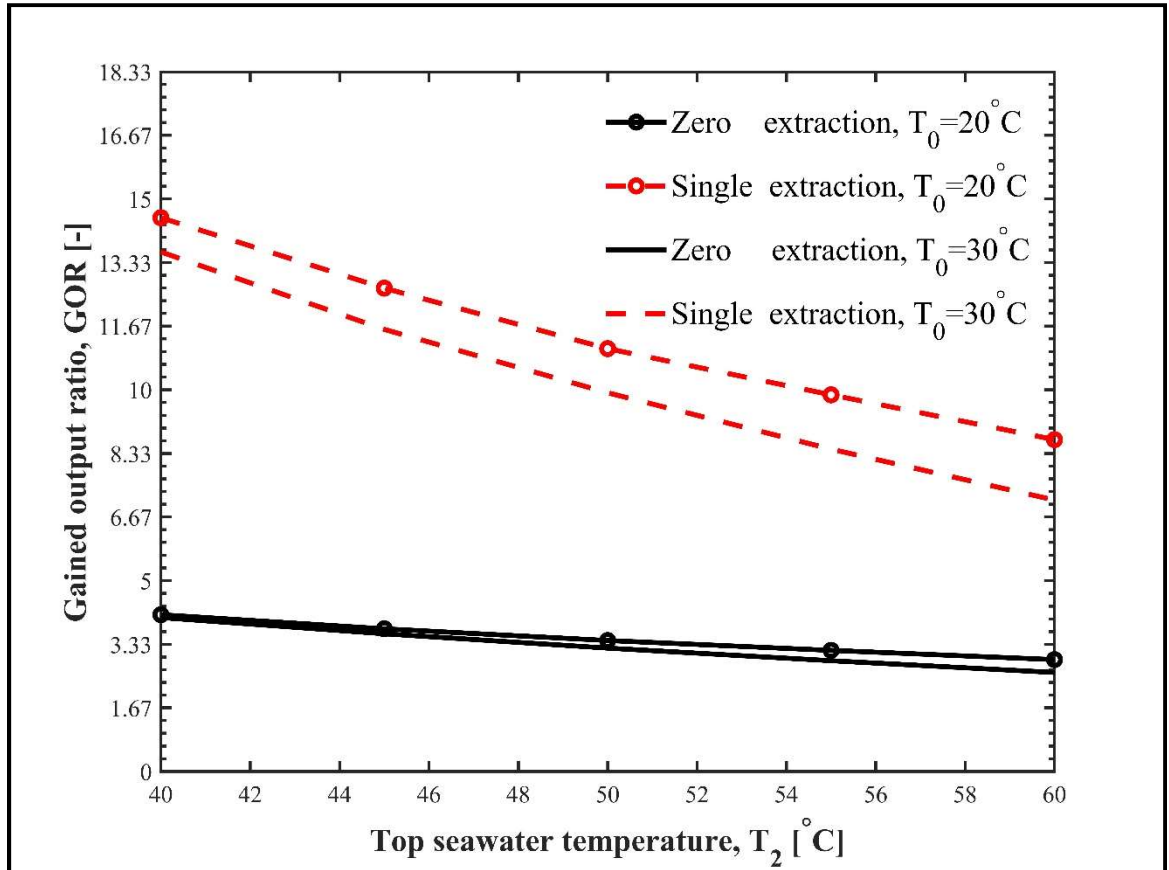


Figure 5.11. Impact of the top seawater temperature; $\epsilon_{\text{deh}} = \epsilon_{\text{hum}} = 100\%$.

The drop in the generated entropy because of the thermodynamic balancing regarded with injection and extraction technique results in an enormous rise in the system performance that approximately multiplied three folds in comparison with the system without injection and extraction at top seawater temperature of 50 °C with a bottom temperature of 20 and 30 °C at dehumidifier inlet.

Figure 5.12 shows the trend of the HDH cycle with similar operation variables except for energy effectiveness (97%). The performance at the seawater bottom temperature of 20 °C rises (with and without injection and extraction) as the seawater temperature at the humidifier inlet rises. This escalation is attributed to the rise of the seawater temperature at the humidifier inlet that results in improved humidifier vaporization, therefore, extra water vapor is liquefied in the dehumidification tower. The system performance with injection and extraction is multiplied twice in comparison with the system without extraction at 50 °C top seawater temperature.

At a seawater temperature of 30 °C at the dehumidifier inlet, the performance of the system without extraction drops as the top seawater temperature rises because of the rise in the input heat rate required. Thermodynamic balancing using one injection and extraction elevates the system performance up to the value of 55 °C top seawater temperature, after that, the performance drops. The performance behavior could be attributed to the rise in the top seawater temperature which means improved vaporization up to a certain level, after that, the rise of input heat rate required impact exceeds the vaporization generation as the top seawater temperature increases, hence, the system performance decreases. Applying the injection and extraction technique, the system

performance is doubled at 55 °C top seawater temperature as a result of the drop in the generated entropy.

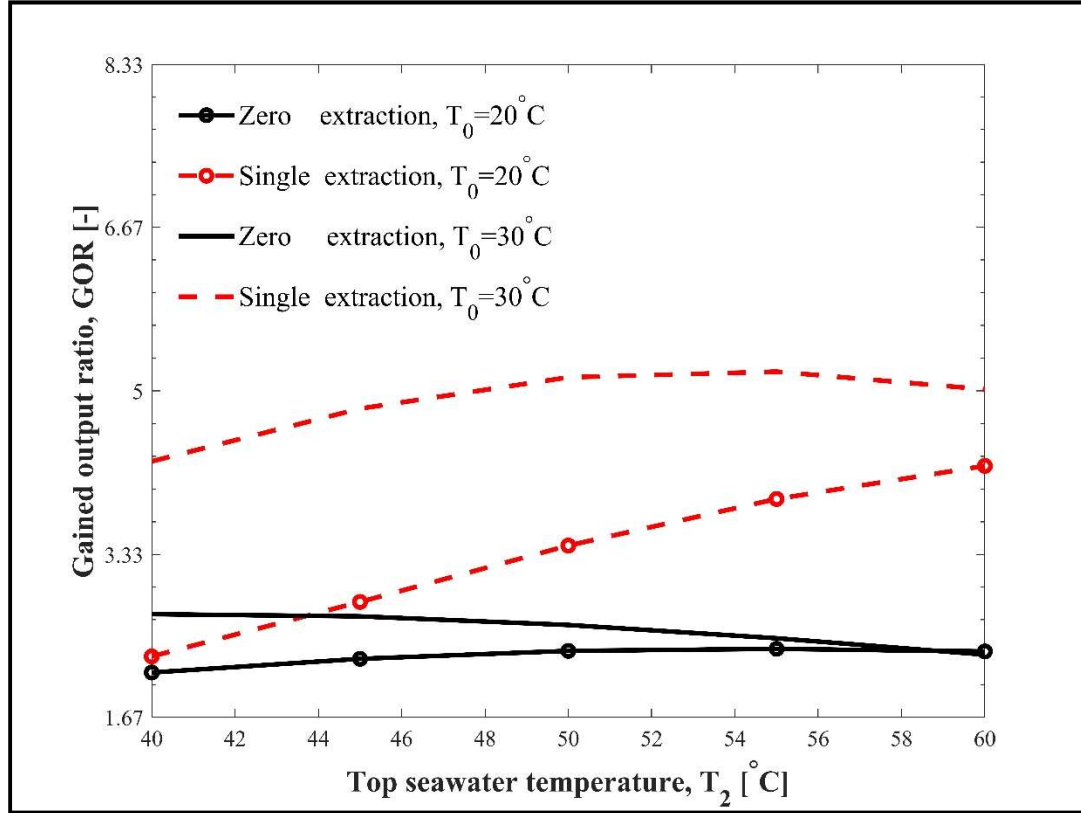


Figure 5.12. Impact of the top seawater temperature; $\epsilon_{\text{deh}} = \epsilon_{\text{hum}} = 97\%$.

The system performance at bottom seawater temperature of 30 °C is more than that at 20 °C for the system with and without injection and extraction that attributed to the lower input heat needed at 30 °C seawater temperature at the dehumidifier inlet.

At a top seawater temperature of 40 °C, the GOR for single extraction at a feed seawater temperature of 30 °C is double in comparison to 20 °C. Then the variance in GOR declines as the top temperature rises to a difference of almost 0.4 at top seawater temperature 60 °C. The reduction in the difference is attributed to the reduction of input heat needed in

comparison with the impact of feed seawater's capability of more moisture condensation in the dehumidifier. Once attaining a 60 °C top seawater temperature, the system performance at the bottom temperature of 20 °C without extraction surpasses the case of 30 °C. At this level, the impact of the feed seawater to dehumidify the saturated air betters the impact of input heat needed.

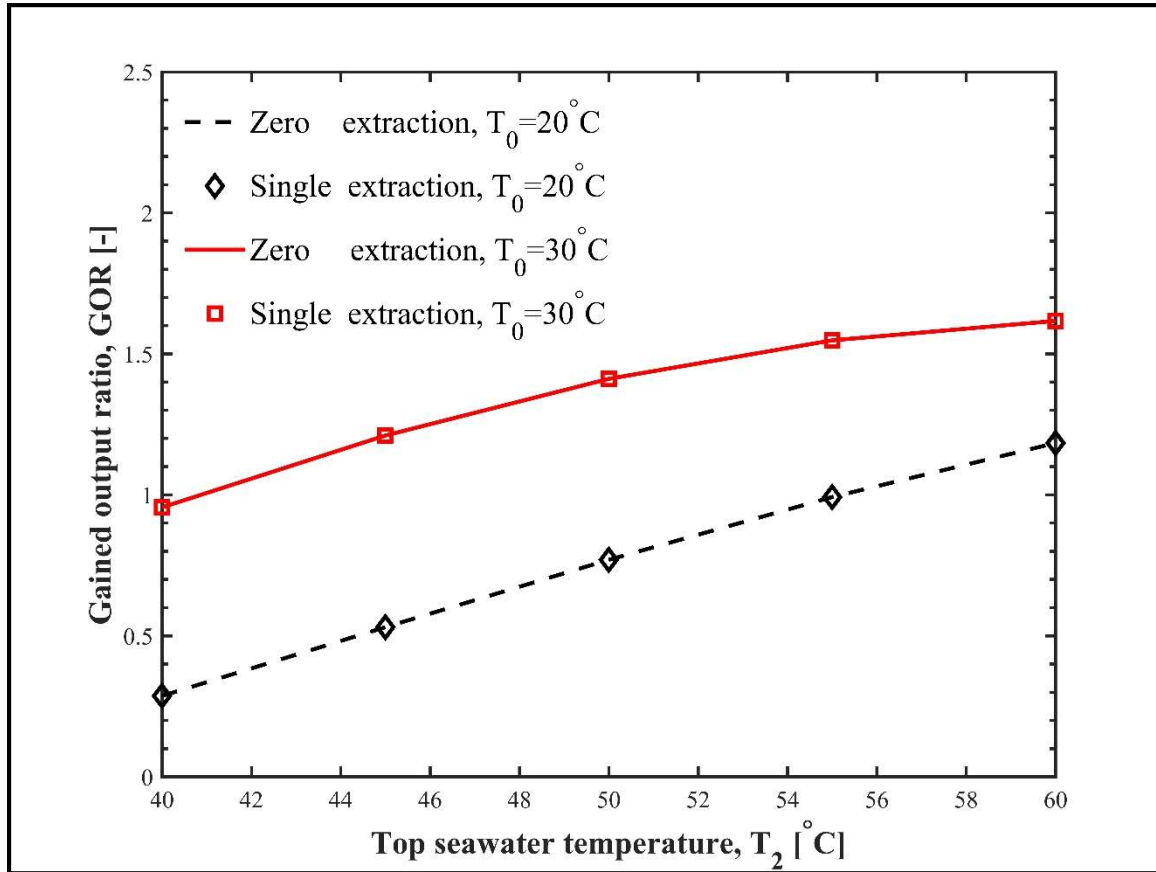


Figure 5.13. Top seawater temperature impact; $\varepsilon_{\text{deh}} = \varepsilon_{\text{hum}} = 91\%$.

Figures 5.13 and 5.14 illustrate the impact of operating the cycle under the same operational variables at the energy effectiveness of 91% and 85%, respectively. The impact of injection and extraction on the cycle GOR is zero with rising the top seawater temperature as the thermodynamic balancing impact works at higher values of energy

effectiveness. Figure 5.13 depicts the rising top temperature of seawater elevates the system performance with and without injection and extraction technique. The rise in system performance can be associated with the humidifier vaporization as a result of the rise in top seawater temperature.

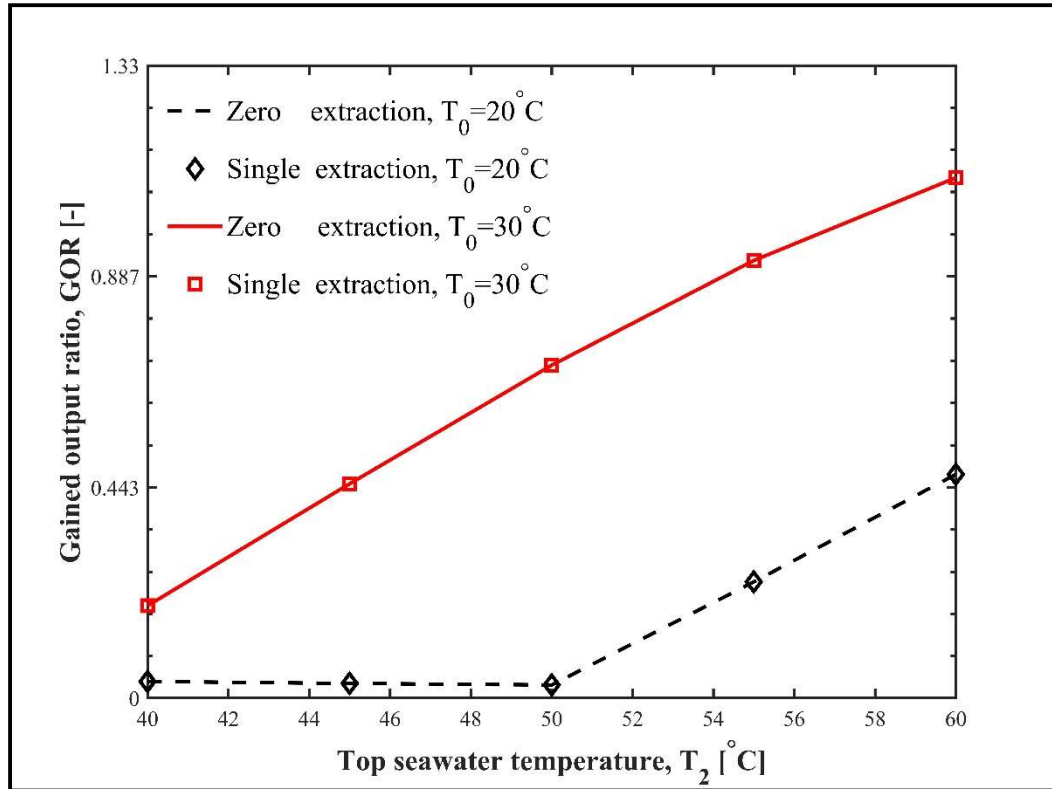


Figure 5.14. Impact of the top seawater temperature; $\epsilon_{\text{deh}} = \epsilon_{\text{hum}} = 85\%$.

Figure 5.14 illustrates the rise in top seawater temperature at $T_2 = 20$ and 30°C . In $T_0 = 20^\circ\text{C}$ case, the GOR is extremely minimal up to a 50°C top seawater temperature. Afterward this level, the system performance rises abruptly from 0.2 to almost one. The system performance is marginal because of the low energy effectiveness in the humidifier and dehumidifier. At higher values than 50°C , the rise in top seawater temperature affects the system GOR as it improves the vaporization meaning additional condensation in the

dehumidification part and that leads to a rise in the system performance. At the bottom seawater temperature of 30 °C, the system performance rises as the top seawater temperature rises. This elevation in the evaporation and condensation processes is regarded as the rise in top seawater temperature.

5.6 Concluding remarks

In this chapter, the closed-air open-water water-heated humidification dehumidification desalination system performance is evaluated using the proposed energy effectiveness with zero and single extractions. The operation condition impact on the GOR of the cycle was examined. Cycle GOR can be utilized to be the criterion for investigating the impact of applying the extraction and injection technique. Based on the analysis presented in this chapter, it is realizable to draw the following conclusions,

1. Pinch-based energy effectiveness is introduced for the analysis of the humidifier and dehumidifier as a replacement to the terminal-based energy effectiveness, enthalpy pinch, and temperature pinch that is usually defined for the humidifier and dehumidifier. Energy effectiveness combines the impact of both concentration (salinity and humidity for seawater and air, respectively) and temperature.
2. Changing the mass flowrate ratio of seawater-to-air by using the mass injection and extraction technique conveys the heat capacity ratio to a value of ($HCR = 1$) at the injection point. Therefore, the local energy effectiveness along the dehumidifier and humidifier is tremendously increased which results in system performance enhancement.
3. The gained output ratio of the dehumidification humidification desalination system with and without extraction increases as energy effectiveness rises.
4. Applying extraction and injection technique improves the system performance after a certain value of energy effectiveness. Before this point, the system performance after applying injection and extraction overlaps with that of zero extraction i.e. injection and extraction technique has zero impact before this point.

5. The gained output ratio of the dehumidification humidification desalination system with and without extraction rises as top seawater temperature rises to a certain level where it is optimum and then it decreases after it.
6. At $\varepsilon = 97\%$ of the humidifier and dehumidifier; the performance of the humidification dehumidification system performance after applying extraction and injection technique is enhanced by 83.1%.

CONCLUSION AND RECOMMENDATIONS

In this thesis, a novel method based on energy effectiveness as a performance index is suggested in the current research in order to investigate and model the water-heated, CAOW humidification dehumidification desalination cycle at balanced and unbalanced states. The proposed method objective is evading the possibility of temperature cross and second law violation as a probable result in the humidifier. It has been found the proposed method is tested and verified against the theoretical data in the reported literature. An assessment of the evaluated results based on the current model and the literature is presented and good agreement is noted between the current model and theoretical data with a minor error of $\pm 2.29\%$. A direct design procedure is proposed in this research in order to size the closed-air open-water water-heated humidification dehumidification desalination system. The city of Dhahran is taken as a case study where the average minimum and maximum temperature of seawater is assumed 25 and 70°C, respectively. The system is designed to produce 5 kg hr⁻¹ of freshwater and the optimal energy effectiveness is found to be 78%. An analytical solution for the numerical proposed model is generated. A benchmark of the gained output ratio evaluated by using the analytical model is conducted where good agreement with the numerical model is observed with an error of around $\pm 5\%$.

As a final point, the closed-air open-water water-heated humidification dehumidification desalination system performance is evaluated using the proposed pinch-based energy effectiveness with zero and single extractions. The operation conditions impact on the cycle performance was examined. Cycle GOR can be utilized to be the criterion for investigating

the impact of applying the extraction and injection technique. Based on the analysis presented in this thesis, it is realizable to draw the following conclusions,

1. Pinch-based energy effectiveness is introduced for the analysis of the humidifier and dehumidifier as a replacement to the terminal-based energy effectiveness, enthalpy pinch, and temperature pinch that is usually defined for the humidifier and dehumidifier. Energy effectiveness combines the impact of both concentration (salinity and humidity for seawater and air, respectively) and temperature.
2. Changing the mass flowrate ratio of seawater-to-air by using the mass injection and extraction technique conveys the heat capacity ratio to a value of ($HCR = 1$) at the injection point. Therefore, the local energy effectiveness along the dehumidifier and humidifier is tremendously increased which results in performance enhancement of the humidification dehumidification system.
3. The gained output ratio of the dehumidification humidification cycle with and without extraction increases as energy effectiveness rises.
4. Applying extraction and injection technique improves the system performance after a certain value of energy effectiveness. Before this point, the system performance after applying injection and extraction overlaps with that of zero extraction i.e. injection and extraction technique has zero impact before this point.
5. The gained output ratio of the dehumidification humidification desalination system with and without extraction rises as top seawater temperature rises to a certain level where it is optimum and then it decreases after it.
6. At $\varepsilon = 97\%$ of the humidifier and dehumidifier; humidification dehumidification cycle performance after applying injection and extraction technique is enhanced by 83.1%.

Recommendations

Using the proposed pinch-based energy effectiveness is promising with plenty of room to improve its range of applications. As mentioned in this thesis, it was applied on a closed-air open-water water-heated humidification dehumidification desalination system. It is to be mentioned that the energy effectiveness can be applied on any humidification dehumidification arrangement as long as the recirculated air is 100% humid along the humidifier. The following recommendations are offered to improve the proposed pinch-based energy effectiveness further as a promising performance index in the field of humidification dehumidification desalination systems or any system that comprise heat/mass exchange device:

1. Create a correction factor, assumption, or an approximation to compensate for the inconsistency of the air humidity (not 100%) along the humidifier. To illustrate more, this enables the proposed pinch-based energy effectiveness to model the traditional closed-air open-water air-heated humidification dehumidification arrangement.
2. Redefine the proposed energy effectiveness to be a function of the relative humidity as well as the air temperature. Also, the change of air and seawater temperature along the humidifier height should be linked to pinch-based energy effectiveness.
3. Finally, the definition of the proposed pinched-based energy effectiveness should be extended to be applied on any heat and mass exchange device where the relative humidity is varying; a strong example for that is cooling towers.

Nomenclature

Symbols

A	area (m^2)
C	minimum to maximum stream heat capacity
C_p	constant pressure heat capacity (kJ kg^{-1})
HCR	heat capacity ratio
$\dot{H}$	rate of total enthalpy (kW)
h	specific enthalpy (kJ kg^{-1})
h^*	specific enthalpy of seawater normalized to air flowrate (kJ kg^{-1}_d)
h_{fg}	vaporization latent heat (kJ kg^{-1})
L	enthalpy losses due to limitation in area of heat and mass transfer (kJ kg^{-1}_d)
$I_{i\text{th}}$	Bessel function of i^{th} -order
$\dot{m}$	mass flowrate (kg s^{-1})
Me	Merkel number
MR	the ratio of seawater to air mass flowrate
NTU	number of transfer units

P	Pressure (kPa)
$\dot{Q}$	rate of input heat (kW)
s	seawater salinity (ppt)
slp	the slope of humidifier/dehumidifier ($\text{kJ}^{-1} \text{kg}_d \text{ } ^\circ\text{C}$)
T	temperature ($^\circ\text{C}$)
U	U-value coefficient ($\text{m}^{-2} \text{K}^{-1} \text{kW}$)
Z	height of the humidifier (m)

Greeks

ε	energy effectiveness
Δ	change
ζ	coefficients of empirical relations
ϕ	relative humidity
Ψ	enthalpy pinch ($\text{kJ kg}^{-1} \text{da}$)
ω	humidity ratio ($\text{kg}_{\text{H}_2\text{O}} \text{ kg}^{-1}_d$)

Subscripts

0 - 3	states of seawater
-------	--------------------

a	circulated saturated air
b	brine
c	cross section / cold
deh	dehumidifier
da	recirculated dry air
fw	freshwater
h	Hot
ht	heater
hum	humidifier
in	inlet
id	ideal
max	maximum
min	minimum
out	outlet
opt	optimum
Pb	pinch-based
Tb	terminal-based

tan	the closest point at the circulated air curve to the humidifier line
v	water vapor
w	seawater

References

- [1] F. R. Rijsberman, “Water scarcity: Fact or fiction?,” *Agric. Water Manag.*, vol. 80, no. 1-3 SPEC. ISS., pp. 5–22, 2006.
- [2] A. K. Venkatesan, S. Ahmad, W. Johnson, and J. R. Batista, “Salinity reduction and energy conservation in direct and indirect potable water reuse,” *Desalination*, vol. 272, no. 1–3, pp. 120–127, 2011.
- [3] W. Wang *et al.*, “Simultaneous production of fresh water and electricity via multistage solar photovoltaic membrane distillation,” *Nat. Commun.*, vol. 10.3012, no. July, 2019.
- [4] C. Fritzmann, J. Löwenberg, T. Wintgens, and T. Melin, “State-of-the-art of reverse osmosis desalination,” *desalination*, vol. 216, no. 1–3, pp. 1–76, 2007.
- [5] G. P. N. and J. L. V H., “Thermal Design of Humidification-Dehumidification Systems for Affordable Small-Scale Desalination,” *IDA J. Desalin. water reuse*, vol. 3, no. March, pp. 24–34, 2012.
- [6] G. P. Narayan, M. H. Sharqawy, E. K. Summers, J. H. Lienhard, S. M. Zubair, and M. A. Antar, “The potential of solar-driven humidification-dehumidification desalination for small-scale decentralized water production,” *Renew. Sustain. Energy Rev.*, vol. 14, no. 4, pp. 1187–1201, 2010.
- [7] J. F. Klausner, R. Mei, Y. Li, and J. Knight, “Innovative Fresh Water Production Process for Fossil Fuel Plants,” 2006.

- [8] G. Al-Enezi, H. Ettouney, and N. Fawzy, “Low temperature humidification dehumidification desalination process,” *Energy Convers. Manag.*, vol. 47, no. 4, pp. 470–484, 2006.
- [9] H. Scarborough, O. Sahin, M. Porter, and R. Stewart, “Long-term water supply planning in an Australian coastal city: Dams or desalination?,” *Desalination*, vol. 358, no. 1, pp. 61–68, 2015.
- [10] H. B. Ghalib, “Groundwater chemistry evaluation for drinking and irrigation utilities in east Wasit province, Central Iraq,” *Appl. Water Sci.*, vol. 7, no. 1, pp. 3447–3467, 2017.
- [11] G. J. Bowden, H. R. Maier, and G. C. Dandy, “Input determination for neural network models in water resources applications,” *J. Hydrol.*, vol. 301, no. 1, pp. 93–107, 2005.
- [12] A. Cilluffo and N. G. Ruiz, “World’s population is projected to nearly stop growing by the end of the century,” 2019. [Online]. Available: www.pewresearch.org/fact-tank/2019/06/17/.
- [13] A. Alkaisi, R. Mossad, and A. Sharifian-Barforoush, “A review of the water desalination systems integrated with renewable energy,” *Energy Procedia*, vol. 110, no. 1, pp. 268–274, 2017.
- [14] J. A. Allan, “Hydro-Peace in the Middle East: Why no Water Wars? A Case Study of the Jordan River Basin,” *SAIS Rev.*, vol. 22, no. 1, pp. 255–272, 2002.
- [15] Y. Ghalavand, M. S. Hatamipour, and A. Rahimi, “A review on energy consumption

- of desalination processes,” *Desalin. Water Treat.*, vol. 54, no. 1, pp. 37–41, 2014.
- [16] M. Al-sahali and H. Ettouney, “Developments in thermal desalination processes: Design, energy, and costing aspects,” *Desalination*, vol. 214, no. 1, pp. 227–240, 2007.
- [17] A. R. Hoffman, “Desalination: An Important Part of Our Water Future,” 2013. [Online]. Available: www.lapsedphysicist.org/2013/12/24/.
- [18] L. F. Greenlee, D. F. Lawler, B. D. Freeman, B. Marrot, and P. Moulin, “Reverse osmosis desalination: Water sources, technology, and today’s challenges,” *Water Res.*, vol. 43, no. 1, pp. 2317–2348, 2009.
- [19] R. Borsani and S. Rebagliati, “Fundamentals and costing of MSF desalination plants and comparison with other technologies,” *Desalination*, vol. 182, no. 1, pp. 29–37, 2005.
- [20] H. Fath, A. Sadik, and T. Mezher, “Present and Future Trend in the Production and Energy Consumption of Desalinated Water in GCC Countries,” *Int. J. Therm. Environ. Eng.*, vol. 5, no. 1, pp. 155–165, 2013.
- [21] S. Lattemann, M. D. Kennedy, J. C. Schippers, and G. Amy, “Global Desalination Situation,” in *Sustainability Science and Engineering*, vol. 2, no. 09, Elsevier, 2010, pp. 7–39.
- [22] M. A. Al-weshahi, A. Anderson, and G. Tian, “Exergy efficiency enhancement of MSF desalination by heat recovery from hot distillate water stages,” *Appl. Therm. Eng.*, vol. 53, no. 2, pp. 226–233, 2013.

- [23] M. A. Darwish, F. Al-juwayhel, and H. K. Abdulraheim, "Multi-effect boiling systems from an energy viewpoint," *Desalination*, vol. 194, no. 1, pp. 22–39, 2006.
- [24] K. H. Mistry, A. Mitsos, and J. H. Lienhard V, "Optimal operating conditions and configurations for humidification-dehumidification desalination cycles," *Int. J. Therm. Sci.*, vol. 50, no. 1, pp. 779–789, 2011.
- [25] G. P. Narayan, M. H. Sharqawy, J. H. Lienhard V., and S. M. Zubair, "Thermodynamic analysis of humidification dehumidification desalination cycles," *Desalin. Water Treat.*, vol. 16, no. 1, pp. 339–353, 2010.
- [26] G. P. Narayan, K. H. Mistry, M. H. Sharqawy, S. M. Zubair, and J. H. Lienhard V, "ENERGY EFFECTIVENESS OF SIMULTANEOUS HEAT AND MASS EXCHANGE DEVICES," *Front. Heat Mass Transf.*, vol. 023001, no. 1, pp. 1–13, 2010.
- [27] R. K. McGovern, G. P. Thiel, G. P. Narayan, S. M. Zubair, and J. H. Lienhard V, "Performance limits of zero and single extraction humidification-dehumidification desalination systems," *Appl. Energy*, vol. 102, no. 1, pp. 1081–1090, 2013.
- [28] G. P. Narayan, K. M. Chehayeb, R. K. McGovern, G. P. Thiel, S. M. Zubair, and J. H. Lienhard V, "Thermodynamic balancing of the humidification dehumidification desalination system by mass extraction and injection," *Int. J. Heat Mass Transf.*, vol. 57, no. 2, pp. 756–770, 2013.
- [29] G. P. Narayan, J. H. Lienhard V, and S. M. Zubair, "Entropy generation minimization of combined heat and mass transfer devices," *Int. J. Therm. Sci.*, vol.

- 49, no. 1, pp. 2057–2066, 2010.
- [30] M. A. Ahmed, N. A. A. Qasem, S. M. Zubair, P. Gandhidasan, and H. M. Bahaidarah, “Thermodynamic balancing of the regeneration process in a novel liquid desiccant cooling/desalination system,” *Energy Convers. Manag.*, vol. 176, pp. 86–98, 2018.
- [31] G. Wu, H. Zheng, X. Ma, C. Kutlu, and Y. Su, “Experimental investigation of a multi-stage humidification-dehumidification desalination system heated directly by a cylindrical Fresnel lens solar concentrator,” *Energy Convers. Manag.*, vol. 143, no. 1, pp. 241–251, 2017.
- [32] W. F. He, L. N. Xu, D. Han, L. Gao, C. Yue, and W. H. Pu, “Thermodynamic investigation of waste heat driven desalination unit based on humidification dehumidification (HDH) processes,” *Appl. Therm. Eng.*, vol. 100, no. 1, pp. 315–324, 2016.
- [33] N. A. A. Qasem and S. M. Zubair, “Performance evaluation of a novel hybrid humidification-dehumidification (air-heated) system with an adsorption desalination system,” *Desalination*, vol. 461, no. 1, pp. 37–54, 2019.
- [34] D. Lawal, M. Antar, A. Khalifa, S. Zubair, and F. Al-Sulaiman, “Humidification-dehumidification desalination system operated by a heat pump,” *Energy Convers. Manag.*, vol. 161, no. January, pp. 128–140, 2018.
- [35] S. M. Zubair, M. A. Antar, S. M. Elmutasim, and D. U. Lawal, “Performance evaluation of humidification-dehumidification (HDH) desalination systems with

and without heat recovery options: An experimental and theoretical investigation,” *Desalination*, vol. 436, no. 1, pp. 161–175, 2018.

- [36] H. Rostamzadeh, A. S. Namin, H. Ghaebi, and M. Amidpour, “Performance assessment and optimization of a humidification dehumidification (HDH) system driven by absorption-compression heat pump cycle,” *Desalination*, vol. 447, no. 1, pp. 84–101, 2018.
- [37] K. M. Chehayeb, G. P. Narayan, S. M. Zubair, and J. H. Lienhard V, “Use of multiple extractions and injections to thermodynamically balance the humidification dehumidification desalination system,” *Int. J. Heat Mass Transf.*, vol. 68, no. 1, pp. 422–434, 2014.
- [38] K. M. Chehayeb, G. P. Narayan, S. M. Zubair, and J. H. Lienhard V, “Thermodynamic balancing of a fixed-size two-stage humidification dehumidification desalination system,” *Desalination*, vol. 369, no. 1, pp. 125–139, 2015.
- [39] N. A. A. Qasem, M. A. Ahmed, and S. M. Zubair, “The impact of thermodynamic balancing on performance of a desiccant-based humidification-dehumidification system to harvest freshwater from atmospheric air,” *Energy Convers. Manag.*, vol. 199, p. 112011, Nov. 2019.
- [40] M. A. Ahmed, N. A. A. Qasem, and S. M. Zubair, “Analytical and numerical schemes for thermodynamically balanced humidification-dehumidification desalination systems,” *Energy Convers. Manag.*, vol. 200, no. June, pp. 1–17, 2019.

- [41] M. Faegh, P. Behnam, and M. B. Shafii, "A review on recent advances in humidification-dehumidification (HDH) desalination systems integrated with refrigeration, power and desalination technologies," *Energy Convers. Manag.*, vol. 196, no. June, pp. 1002–1036, 2019.
- [42] H. R. Abbasi and H. Pourrahmani, "Multi-criteria optimization of a renewable hydrogen and freshwater production system using HDH desalination unit and thermoelectric generator," *Energy Convers. Manag.*, vol. 214, no. May, p. 112903, 2020.
- [43] H. Xu, X. Y. Sun, and Y. J. Dai, "Thermodynamic study on an enhanced humidification-dehumidification solar desalination system with weakly compressed air and internal heat recovery," *Energy Convers. Manag.*, vol. 181, no. November, pp. 68–79, 2019.
- [44] D. U. Lawal, S. M. Zubair, and M. A. Antar, "Exergo-economic analysis of humidification-dehumidification (HDH) desalination systems driven by heat pump (HP)," *Desalination*, vol. 443, no. March, pp. 11–25, 2018.
- [45] D. U. Lawal, M. A. Antar, A. Khalifa, S. M. Zubair, and F. Al-Sulaiman, "Experimental investigation of heat pump driven humidification-dehumidification desalination system for water desalination and space conditioning," *Desalination*, vol. 475, no. September, p. 114199, 2020.
- [46] I. Ghofrani and A. Moosavi, "Energy, exergy, exergoeconomics, and exergoenvironmental assessment of three brine recycle humidification-dehumidification desalination systems applicable for industrial wastewater

- treatment,” *Energy Convers. Manag.*, vol. 205, no. July 2019, p. 112349, 2020.
- [47] K. R. Ranjan and S. C. Kaushik, “Energy , exergy and thermo-economic analysis of solar distillation systems: A review,” *Renew. Sustain. Energy Rev.*, vol. 27, no. 1, pp. 709–723, 2013.
- [48] M. Faegh, P. Behnam, and M. B. Shafii, “A review on recent advances in humidification-dehumidification (HDH) desalination systems integrated with refrigeration, power and desalination technologies,” *Energy Convers. Manag.*, vol. 196, no. 1, pp. 1002–1036, 2019.
- [49] M. A. Ahmed, S. M. Zubair, M. A. Abido, and H. M. Bahaidarah, “An innovative closed-air closed-desiccant HDH system to extract water from the air: A case for zero-brine discharge system,” *Desalination*, 2018.
- [50] M. Zamen, S. M. Soufari, and M. Amidpour, “Improvement of solar humidification-dehumidification desalination using multi-stage process,” *Chem. Eng. Trans.*, vol. 25, no. 1, pp. 1091–1096, 2011.
- [51] S. Hou, “Two-stage solar multi-effect humidification dehumidification desalination process plotted from pinch analysis,” *desalination*, vol. 222, no. 1, pp. 572–578, 2008.
- [52] R. K. McGovern, G. P. Thiel, G. P. Narayan, S. M. Zubair, and J. H. Lienhard V, “Performance limits of zero and single extraction humidification-dehumidification desalination systems,” *Appl. Energy*, vol. 102, no. 1, pp. 1081–1090, 2013.
- [53] G. P. Narayan, R. K. McGovern, S. M. Zubair, and J. H. L. V, “High-temperature-

- steam-driven, varied-pressure, humidification-dehumidification system coupled with reverse osmosis for energy-efficient seawater desalination,” *Energy*, vol. 37, no. 1, pp. 482–493, 2012.
- [54] G. Prakash Narayan, K. H. Mistry, M. H. Sharqawy, S. M. Zubair, and J. H. Lienhard, “Energy effectiveness of simultaneous heat and mass exchange devices,” *Front. Heat Mass Transf.*, vol. 023001, no. 1, pp. 1–13, 2010.
- [55] G. P. Narayan, R. K. McGovern, J. H. Lienhard, and S. M. Zubair, “Variable Pressure Humidification Dehumidification Desalination System,” in *ASME/JSME 8th Thermal Engineering Joint conference*, 2011.
- [56] M. S. Elzayed, M. A. M. Ahmed, N. A. A. Qasem, M. A. Antar, and S. M. Zubair, “A design procedure to size thermodynamically-balanced humidification-dehumidification desalination systems,” *Energy Convers. Manag.*, vol. 224, no. March, p. 113357, 2020.
- [57] W. F. He, D. Han, L. N. Xu, C. Yue, and W. H. Pu, “Performance investigation of a novel water–power cogeneration plant (WPCP) based on humidification dehumidification (HDH) method,” *Energy Convers. Manag.*, vol. 110, no. 1, pp. 184–191, 2016.
- [58] X. Huang, W. Liu, X. Yu, T. Ke, X. Ling, and Y. Li, “Thermodynamic analysis of a two-stage humidification-dehumidification system with reflux configuration,” *Energy Convers. Manag.*, vol. 197, no. 1, pp. 1–13, 2019.
- [59] X. Huang, X. Ling, Y. Li, W. Liu, and T. Ke, “A graphical method for the

- determination of optimum operating parameters in a humidification-dehumidification desalination system,” *Desalination*, vol. 455, no. 1, pp. 19–33, 2019.
- [60] Y. Zhao, H. Zheng, S. Liang, N. Zhang, and X. long Ma, “Experimental research on four-stage cross flow humidification dehumidification (HDH) solar desalination system with direct contact dehumidifiers,” *Desalination*, vol. 467, no. 1, pp. 147–157, 2019.
- [61] M. S. Elzayed, M. A. M. Ahmed, M. A. Antar, M. H. Sharqawy, and S. M. Zubair, “The impact of thermodynamic balancing on the performance of a humidification dehumidification desalination system,” *Therm. Sci. Eng. Prog.*, vol. 21, no. October, p. 100794, 2020.
- [62] G. P. Thiel and J. H. Lienhard V, “Entropy generation in condensation in the presence of high concentrations of noncondensable gases,” *Int. J. Heat Mass Transf.*, vol. 55, no. 1, pp. 5133–5147, 2012.
- [63] R. W. Hyland and A. Wexler, “Formulations for the thermodynamic properties of the saturated phases of H₂O from 173.15 K to 473.15 K,” in *ASHRAE Transaction*, 1983, vol. 89, no. 2, pp. 500–519.
- [64] M. H. Sharqawy, M. A. Antar, S. M. Zubair, and A. M. Elbashir, “Optimum thermal design of humidification dehumidification desalination systems,” *desalination*, vol. 349, no. 1, pp. 10–21, 2014.
- [65] M. H. Sharqawy, J. H. Lienhard V, and S. M. Zubair, “On Thermal Performance of

- Seawater Cooling Towers,” *Journa Eng. Gas Turbines Power*, vol. 133, no. 4, pp. 1–7, 2011.
- [66] B. S. Baclic, “A simplified formula for cross-flow heat exchanger effectiveness,” *ASME J. Heat Transf.*, vol. 100, no. 4, pp. 746–747, 1978.
- [67] T. Kuppan, *Heat exchanger design handbook*. New York: Marcel Dekker Inc., 2000.
- [68] S. M. Elmutasim, M. A. Ahmed, M. A. Antar, P. Gandhidasan, and S. M. Zubair, “Design strategies of conventional and modified closed-air open-water humidification dehumidification systems,” *Desalination*, vol. 435, no. 1, pp. 114–127, 2018.
- [69] “National Centers for Environmental Information.” [Online]. Available: <https://www.ncdc.noaa.gov/>. [Accessed: 12-Nov-2020].
- [70] IPCC *et al.*, *Climate Change 2013: The Physical Science Basis. Contribution of Working Group I to the Fifth Assessment Report of the Intergovernmental Panel on Climate Change*. Cambridge, United Kingdom and New York, NY, USA: Cambridge University Press, 2013.
- [71] International Energy Agency, “CO₂ emissions from fuel combustion,” Paris, 2016.
- [72] Electricity & Cogeneration Regulatory Authority-Saudi Arabia, “ECRA-Reports,” 2015.
- [73] Y. Wada, L. P. H. Van Beek, D. Viviroli, H. H. Drr, R. Weingartner, and M. F. P. Bierkens, “Global monthly water stress: 2. Water demand and severity of water stress,” *Water Resour. Res.*, vol. 47, no. 7, pp. 1–17, 2011.

- [74] Y. Wada, L. P. H. van Beek, and M. F. P. Bierkens, “Modelling global water stress of the recent past: on the relative importance of trends in water demand and climate variability,” *Hydrol. Earth Syst. Sci.*, vol. 15, no. 12, pp. 3785–3808, 2011.
- [75] M. M. Mekonnen and Y. A. Hoekstra, “Four Billion People Experience Water Scarcity,” *Sci. Adv.*, vol. 2, no. February, pp. 1–7, 2016.
- [76] H. El-Dessouky and H. Ettouney, *Fundamentals of salt water desalination*. Elsevier, 2002.
- [77] A. E. Kabeel, M. H. Hamed, Z. M. Omara, and S. W. Sharshir, “Water Desalination Using a Humidification-Dehumidification Technique—A Detailed Review,” *Nat. Resour.*, vol. 04, no. 03, pp. 286–305, 2013.
- [78] F. A. Al-Sulaiman, M. I. Zubair, M. Atif, P. Gandhidasan, S. A. Al-Dini, and M. A. Antar, “Humidification dehumidification desalination system using parabolic trough solar air collector,” *Appl. Therm. Eng.*, vol. 75, pp. 809–816, 2015.
- [79] J. Monteith and M. Unsworth, *Principles of environmental physics*. Amsterdam: Academic Press, 2007.

Vitae

Name : ABDULAZIZ ABDULLAH ALSAIF

Nationality : SAUDI

Date of Birth : 10/4/1967

Email : AALSAIF1991@gmail.com

Address : Al-Jubail 35815 – 4105
Eastern Province, Saudi Arabia

Academic Background : B.Sc. Honor in Mechanical Engineering
King Saud University, Riyadh, Saudi Arabia

Research Interests : Heat and Mass Transfer Exchangers, Thermal Systems,
Thermodynamic Balancing, Air Conditioning Systems,
Remnant Irreversibilities, Hybrid Systems, and Water
Desalination

Publications

- **Abdulaziz A. AlSaif**, Mohamed S. Elzayed, M.A.M. Ahmed, Mohamed A. Antar, S.M. Zubair, A robust definition of energy effectiveness for humidification dehumidification desalination systems, Desalination – under review.