Thermal Design of Humidification
Dehumidification Systems for Affordable and
Small-scale Desalination

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Submitted to the Department of Mechanical Engineering in partial fulfillment of the requirements for the degree of Doctor of Philosophy in Mechanical Engineering at the

MASSACHUSETTS INSTITUTE OF TECHNOLOGY
September 2012
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Abstract

The humidification dehumidification (HDH) technology is a carrier-gas-based thermal desalination technique ideal for application in a small-scale system but, currently, has a high cost of water production (about 30 $/m^3 of pure water produced). The present thesis describes fundamental contributions to the thermal design of HDH systems that have made them affordable (< 5 $/m^3). These contributions include: (1) the development of thermal design algorithms for thermodynamic balancing via mass extractions and injections; (2) design of a bubble column dehumidifier for high heat and mass transfer rates even in the presence of a large percentage of non-condensable gas; and (3) optimization of system design with pressure as a parameter.

Definition of a novel non-dimensional parameter known as the ‘modified heat capacity rate ratio’ (HCR) has enabled designs that minimize the imbalance in local driving temperature and concentration differences. The design algorithm has been validated by experimental data from a pilot-scale HDH unit constructed as part of the thesis work. The energy consumption of HDH was reduced by 55% by this technique.

Bubble column (BC) heat exchangers can have high heat and mass transfer rates by condensing the vapor-gas mixture in a column of liquid rather than on a cold surface. New physical understanding of heat transfer in BCs has led to low pressure drop designs (< 1 kPa) and the concept of multistaging the uniform temperature column in several temperature steps has led to high effectiveness designs (about 90%). A prototype with an order of magnitude higher heat rate compared to existing dehumidifiers operating in the film condensation regime was developed to validate the physical models.

Overall cost of water production in HDH can be further reduced by treating pressure as a variable parameter. Systems operating under varied pressure consume half the energy as existing HDH systems.

Thesis Supervisor: John H. Lienhard V
Title: Samuel C. Collins Professor of Mechanical Engineering
I would like to dedicate this doctoral dissertation to my mother and father - Malathy and Govindan. There is no doubt in my mind that without their continued support I could not have gotten this far.
Acknowledgments

It has been a terrific four years at MIT for me and I have several people to thank for that. There could not have been a better thesis supervisor for me than Professor John Lienhard. I dedicate all accolades I received during my PhD work to him\(^1\). I also thank him for providing me with a transformative amount of self-confidence.

Professor Syed Zubair has been a great pillar of support and strength. I could not have asked for a better mentor to collaborate with. I would like to thank Professor Bora Mikic for giving me advise throughout my PhD. I especially thank him for teaching 2.55 in the unique and awesome way he does and separately, for incepting the idea of direct contact condensation in me. I am indebted to Mr. Leon Awerbuch for being a great advisor for me. I consider myself lucky for having the foremost expert on desalination as a mentor. I am grateful to Professor Alexander Mitsos for giving me advise whenever I needed it. I also thank him for being my co-author in the optimization paper presented at IIT Madras.

Professor Sarit Das, my Master’s thesis advisor and mentor, visited MIT in 2011 (when my research was going through a bit of a slump) and he energized me. I owe him a great deal for all that he is done for me during that period and beyond. I am thankful to Professor Peter Griffith for spending time with me from time-to-time and sharing with me the decades of wisdom he has on design of thermal systems.

Dr. Mostafa was the post-doc in the group when I first joined MIT and was extremely supportive and taught me several things. I owe him my gratitude. I am especially thankful for the help he later provided in design of the bubble column experiment.

I have had the oppurtunity to work with some terrific MIT undergrads including Yoshio Perez, Steven Lam, Maximuss St. John, Victor Nevarez and Jeff Huang. I appreciate their efforts and hard work for the project and thank them for working with me. I m sure they will go on to become very successful in their careers. I am

\(^1\)Including the best paper award at the IDA world water congress 2011, MIT Legatum Fellowship for Development and Entrepreneurship, the deFlorez prize for graduate design 2012, all the papers and patent disclosures
excited by the opportunity to further work with Steven and Max during the coming year. Many thanks to Karim Chehayeb for working with me over the last year from AUB and at MIT as an exchange student on the balancing project. I wish him all the best with his graduate studies at MIT!

I would like to express my whole hearted thanks to all co-authors in all of my papers including Ronan McGovern, Karan Mistry, Greg Thiel, Jacob Miller, Fahad Al-Sulaiman and Ed Summers. Thanks for working with me and greatly contributing to the project! My thanks to Professor Antar for being a great collaborator and co-author. I would like to thank Professor Amro Al Qutub for reviving the idea of varied pressure HDH systems. I am indebted to Shanon Liburd, Anand Palapally, Leo Banchik, Moe Mirhi, Martin Sievers, Jacob Miller, Anurag Bajpayee and all previously mentioned members of the Lienhard group for being so patient with me and for being such a fun group.

Thanks to my friends at MIT - Jongho, Chintoo, Sourabh, Amith, Sam Davis, Meggy, Gaurav - for keeping me sane for four years. Special thanks to Sourabh for helping me with the title for the thesis and to Meggy for proof reading my thesis for language errors. Many thanks to the awesome administrative staff at MIT - Leslie, Una, Joan, Angela Mickunas, Kate, Laura, Meghan Moore, Christine Gervais and Christine Smaldone.

I would like to thank the wonderful couple at Yoga 24x7 - Aravind and Krishna Priya - for being my spiritual mentors.

I would like to thank my brother - Prasad - for always being there for me! I would like to thank my soon-to-be-wife, Aishwarya for being the best partner anybody can ever have. Thank you all!

Last but not least, I would like to thank the King Fahd University of Petroleum and Minerals for funding the research reported in this thesis through the Center for Clean Water and Clean Energy at MIT and KFUPM (project # R4-CW-08).
# Contents

1 Introduction

<table>
<thead>
<tr>
<th>Section</th>
<th>Page</th>
</tr>
</thead>
<tbody>
<tr>
<td>Global water problem</td>
<td>31</td>
</tr>
<tr>
<td>1.1 Conventional desalination technologies</td>
<td>33</td>
</tr>
<tr>
<td>1.2 Humidification Dehumidification (HDH)</td>
<td>36</td>
</tr>
<tr>
<td>1.3 Organization of the thesis</td>
<td>38</td>
</tr>
</tbody>
</table>

2 Technology review

<table>
<thead>
<tr>
<th>Section</th>
<th>Page</th>
</tr>
</thead>
<tbody>
<tr>
<td>2.1 Review of systems in literature</td>
<td>42</td>
</tr>
<tr>
<td>2.1.1 Closed-air open-water (CAOW) water heated systems</td>
<td>43</td>
</tr>
<tr>
<td>2.1.2 Closed-water open-air (CWOA) water heated systems</td>
<td>47</td>
</tr>
<tr>
<td>2.1.3 Closed-air open-water (CAOW) air heated systems</td>
<td>50</td>
</tr>
<tr>
<td>2.1.4 HDH systems with mass extractions and injections</td>
<td>54</td>
</tr>
<tr>
<td>2.2 Alternate cycles resembling the HDH process</td>
<td>57</td>
</tr>
<tr>
<td>2.2.1 Dew-vaporation technique</td>
<td>57</td>
</tr>
<tr>
<td>2.2.2 Diffusion-driven desalination technique</td>
<td>58</td>
</tr>
<tr>
<td>2.2.3 Atmospheric water vapor processers</td>
<td>59</td>
</tr>
<tr>
<td>2.3 Review of components in the HDH system</td>
<td>60</td>
</tr>
<tr>
<td>2.3.1 Humidifiers</td>
<td>60</td>
</tr>
<tr>
<td>2.3.2 Dehumidifiers</td>
<td>62</td>
</tr>
<tr>
<td>2.4 Performance benchmarking of HDH systems</td>
<td>64</td>
</tr>
<tr>
<td>2.5 Un answered questions</td>
<td>65</td>
</tr>
</tbody>
</table>
3 Thermal design of simultaneous heat and mass exchange (HME) devices

3.1 Control volume models ............................................. 67
   3.1.1 New heat and mass exchanger (HME) terminology .......... 70
   3.1.2 Equations and modeling details .......................... 76
   3.1.3 Solution technique .................................. 80
3.2 Control volume based second law based design of HME devices ... 81
   3.2.1 Expressions for entropy generation ..................... 82
   3.2.2 Condition for minimum entropy generation ............. 87
   3.2.3 Section summary ................................ 92
3.3 Applicability of energy effectiveness .......................... 93
   3.3.1 Heat exchangers .................................. 93
   3.3.2 Direct contact heat and mass exchangers ............... 94
   3.3.3 Indirect contact heat and mass exchangers ............. 99
3.4 Chapter conclusions ........................................ 100

4 Theoretical control-volume based analysis of existing embodiments
of the HDH system ............................................ 103

4.1 Water heated HDH cycle ....................................... 105
   4.1.1 Effect of relative humidity of the air entering and exiting the
         humidifier ($\phi_{a,1}$, $\phi_{a,2}$). .......................... 105
   4.1.2 Effect of component effectiveness ($\epsilon_h$, $\epsilon_d$). .......... 108
   4.1.3 Effect of top temperature ($T_{w,2}$) ...................... 108
   4.1.4 Effect of bottom temperature ($T_{w,0}$) ................. 111
4.2 Single and multi-stage air heated cycle ...................... 113
4.3 Chapter conclusions ........................................ 114

5 Thermodynamic balancing of HME devices and the HDH system by
mass extraction and injection .................................... 117

5.1 Thermal balancing in simultaneous heat and mass transfer devices 118
   5.1.1 'Control volume' balancing ............................ 118
5.1.2 Enthalpy pinch: novel parameter to define performance of HME device .............................................. 120
5.1.3 Mass extractions or injections based balancing ................................................................. 124
5.1.4 Functional form for continuous thermodynamic balancing ..................................................... 127

5.2 Modeling of HDH systems ........................................................................................................... 129
5.2.1 System without extractions ...................................................................................................... 129
5.2.2 System with infinite extractions and injections ......................................................................... 131
5.2.3 System with a single extraction and injection .......................................................................... 131
5.2.4 Property packages .................................................................................................................. 133

5.3 Results and discussion ................................................................................................................. 133
5.3.1 Continuous extractions with “infinitely large” HME devices: the upper limit on HDH performance ................................................................. 135
5.3.2 Effect of finite system size ....................................................................................................... 137
5.3.3 Uncertainty associated with ‘saturated air’ approximation ...................................................... 137
5.3.4 Comparison of dehumidifier balanced and humidifier balanced systems ................................... 139
5.3.5 Effect of number of extractions .............................................................................................. 141

5.4 Chapter conclusions ..................................................................................................................... 144

6 Experimental investigation of thermal design of HME devices and HDH systems ........................ 149

6.1 Control volume balancing of HME devices ............................................................................... 152
6.1.1 Experimental details ................................................................................................................ 152
6.1.2 Results ................................................................................................................................... 154
6.1.3 Section conclusions ................................................................................................................ 156

6.2 HDH system experiments ........................................................................................................... 157
6.2.1 Effect of mass flow rate ratio .................................................................................................. 160
6.2.2 Effect of top brine temperature .............................................................................................. 160
6.2.3 Effect of feed water temperature ........................................................................................... 162
6.2.4 Effect of mass extraction and injection .................................................................................. 162
6.2.5 Peak Performance of the pilot HDH system 167
6.3 Chapter conclusions 167

7 Mechanical compression driven varied pressure HDH system 169
7.1 Variable pressure HDH cycle 170
7.2 Terminology used 172
7.2.1 Isentropic efficiency 172
7.2.2 System and performance parameters 173
7.3 Equations and modeling details 174
7.3.1 Compressor 174
7.3.2 Expander 175
7.4 Results and discussions 175
7.4.1 Parametric study 175
7.4.2 Selection of expansion device 181
7.5 Comparison with other HDH cycles 183
7.6 Chapter conclusions 184

8 Thermal compression driven hybrid HDH-RO system 187
8.1 Reversible entrainment efficiency of the TVC 193
8.1.1 System and performance parameters 193
8.2 Equations and modeling details 195
8.3 A typical embodiment of the novel cycle 197
8.4 Results and discussions 198
8.4.1 Effect of using an expander and a RO unit for additional water production 198
8.4.2 Effect of heating steam conditions 199
8.4.3 Effect of pressure ratio 204
8.4.4 Effect of operating pressures 204
8.4.5 Effect of air side pressure drop in heat and mass exchangers 207
8.5 Comparison with existing desalination techniques 209
8.6 Chapter conclusions 211
9 Multi-stage bubble columns for high heat rate dehumidification

9.1 Predictive model for combined heat and mass transfer

9.1.1 Thermal resistance between the liquid in the column and the coil surface

9.1.2 Thermal resistance between the liquid in the column and the bubbles

9.1.3 Evaluation of total heat flux from the resistance model

9.2 Experimental details

9.3 Results and discussion

9.3.1 Effect of superficial velocity

9.3.2 Effect of bubble diameter

9.3.3 Effect of inlet mole fraction

9.3.4 Effect of liquid column height

9.3.5 Comparison of model and experiments

9.3.6 Effect of bubble-on-coil impact

9.4 Effect of multi-staging

9.4.1 Prototype

9.4.2 Comparison with existing devices

9.5 Chapter conclusions

10 Conclusions

10.1 Component design

10.2 HDH system design

A Calculation of reversible GOR for HDH

B Entropy generation in a counterflow cooling tower

C Algorithms for modeling HDH systems with and without thermo-dynamic balancing

D Cost of water production in HDH system proposed in this thesis
## E Helium as carrier gas in HDH systems

<table>
<thead>
<tr>
<th>Section</th>
<th>Page</th>
</tr>
</thead>
<tbody>
<tr>
<td>E.1 Rationale for selecting helium</td>
<td>277</td>
</tr>
<tr>
<td>E.1.1 Thermophysical properties</td>
<td>278</td>
</tr>
<tr>
<td>E.1.2 Psychrometric properties</td>
<td>278</td>
</tr>
<tr>
<td>E.1.3 Estimate of gas side heat transfer coefficient in the dehumidifier</td>
<td>280</td>
</tr>
<tr>
<td>E.1.4 Gas side pressure drop in the dehumidifier</td>
<td>282</td>
</tr>
<tr>
<td>E.2 Thermodynamic cycle for HDH desalination</td>
<td>282</td>
</tr>
<tr>
<td>E.3 Relative thermodynamic performance of helium based cycles</td>
<td>283</td>
</tr>
<tr>
<td>E.4 Chapter conclusions</td>
<td>285</td>
</tr>
</tbody>
</table>
List of Figures

1-1 World map showing areas of physical and economic water scarcity [1]. 32
1-2 Illustration of various membrane technologies and the various contaminants they remove (Figure by MIT OCW; source: A. Twort et al. [2]). 34
1-3 Simplest embodiment of HDH process. 37

2-1 Classification of HDH systems based on cycle configurations. 42
2-2 Water heated CAOW HDH process on psychrometric chart. 44
2-3 Water heated CWOA HDH process on psychrometric chart. 48
2-4 Single stage air heated CAOW HDH process on psychrometric chart. 51
2-5 Three stage air heated CAOW HDH process on psychrometric chart. 52
2-6 Schematic diagram of a water-heated, closed-air, open-water humidification-dehumidification desalination system with mass extraction and injection of the moist air stream. 55
2-7 Dew-vaporation process. 58
2-8 Performance of HDH systems in literature. 65

3-1 Psychrometric chart illustrating an example of the Second Law limits on a counterflow cooling tower operation. 71
3-2 Maximum value of effectiveness versus exit relative humidity for a cooling tower: $T_{w,i} = 55^\circ C; T_{a,i} = 34^\circ C; \phi_i = 100; p = 100$ kPa. 75
3-3 Control volume for counterflow cooling tower. 77
3-4 Control volume for counterflow dehumidifier. 79
3-5 Control volume for a heat exchanger. 82
3-6 Non-dimensional entropy generation versus heat capacity rate ratio for countercflow heat exchangers; $\frac{T_{2,\text{h}}}{T_{1,\text{c}}} = 0.5$ .......................... 84

3-7 Mass flowrate ratio versus non-dimensional entropy generation; $T_{a,i} = 34^\circ C$; $\varepsilon = 0.8$; $\phi_o = 0.9$; $\phi_i = 0.6$; $P = 100$ kPa. .................... 88

3-8 Effect of water inlet temperature on entropy generation; $T_{a,i} = 34^\circ C$; $\varepsilon = 0.8$; $\phi_o = 0.9$; $\phi_i = 0.6$; $\dot{m}_{w,in}/\dot{m}_{da} = 1.2 - 29$; $P = 100$ kPa. .... 90

3-9 Effect of air inlet temperature on entropy generation; $T_{w,i} = 55^\circ C$; $\varepsilon = 0.8$; $\phi_o = 0.6$; $\phi_i = 0.8$; $\dot{m}_{w,in}/\dot{m}_{da} = 1.2 - 29$; $P = 100$ kPa. .... 90

3-10 Effect of outlet air relative humidity on entropy generation in a cooling tower; $T_{w,i} = 50^\circ C$; $T_{a,i} = 34^\circ C$; $\varepsilon = 0.7$; $\phi_i = 0.5$; $\dot{m}_{w,in}/\dot{m}_{da} = 1.2 - 29$; $P = 100$ kPa. ..................... 91

3-11 Effect of inlet air relative humidity on entropy generation in a cooling tower; $T_{w,i} = 50^\circ C$; $T_{a,i} = 34^\circ C$; $\varepsilon = 0.7$; $\phi_o = 0.9$; $\dot{m}_{w,in}/\dot{m}_{da} = 1.2 - 29$; $P = 100$ kPa. ..................... 91

3-12 Effect of air side pressure drop on entropy generation in a cooling tower; $T_{w,i} = 50^\circ C$; $T_{a,i} = 34^\circ C$; $\varepsilon = 0.7$; $\phi_o = 0.9$; $\phi_i = 0.6$; $\dot{m}_{w,in}/\dot{m}_{da} = 1.2 - 29$; $P = 100$ kPa. ..................... 92

3-13 Comparison of different effectiveness definitions in a counterflow humidifier with saturated air at the inlet and when moist air is the minimum heat capacity stream (HCR < 1). .................. 95

3-14 Comparison of different effectiveness definitions in a counterflow humidifier when water is the minimum heat capacity stream (HCR > 1). 96

3-15 Comparison of different effectiveness definitions in a counterflow humidifier at thermally balanced condition (HCR = 1). ................ 97

3-16 Comparison of different effectiveness definitions at thermally balanced condition (HCR = 1) for counterflow humidifier with air entering hotter than water. .................. 98

3-17 Comparison of different effectiveness definitions at thermally balanced condition (HCR = 1) for counterflow dehumidifier. ............. 99
4-1 Schematic diagram of water heated closed air open water HDH cycle. 106
4-2 Effect of relative humidity on performance of the WH-CAOW HDH cycle. 107
4-3 Effect of component effectiveness of humidifier on performance of the WH-CAOW HDH cycle. 107
4-4 Effect of component effectiveness of dehumidifier on performance of the WH-CAOW HDH cycle. 109
4-5 Effect of top brine temperature on performance of the WH-CAOW HDH cycle. 110
4-6 HCR of dehumidifier versus GOR at various top brine temperatures. 110
4-7 HCR of humidifier versus GOR at various top brine temperatures. 111
4-8 Effect of feedwater temperature on performance of the WH-CAOW HDH cycle. 112
4-9 HCR of dehumidifier versus GOR at various feedwater temperatures. 112
4-10 HCR of humidifier versus GOR at various feedwater temperatures. 113
4-11 Effect of number of stages on performance of air heated CAOW HDH system. 115

5-1 Temperature versus enthalpy diagram representing the dehumidification process highlighting the maximum change in enthalpy rates (per kg of dry air) that can be achieved by each of the fluid streams ($\Delta h_{max,c}$ and $\Delta h_{max,h}$) and the terminal enthalpy pinches ($\Psi_c$ and $\Psi_h$). 120
5-2 Temperature versus enthalpy diagram for the dehumidification process highlighting ‘loss in ideal enthalpy’ or enthalpy pinch at any given location ($\Psi_{local}$) as a measure of local effectiveness in HME devices. 123
5-3 Temperature versus enthalpy diagram representing the humidification process highlighting the ‘pinch point’ occurring at an intermediate location rather than at a terminal one. 124
5-4 A plot of local enthalpy pinch values ($\Psi_{local}$) relative to the overall enthalpy pinch ($\Psi$) to illustrate the effect of extractions in a dehumidifier with the control volume balanced case. ........................................... 126

5-5 Effect of extraction on the irreversibility in the dehumidifier evaluated at $T_a = 20^\circ C$; $T_c = 70^\circ C$; $\Psi_{deh} = 20$ kJ/kg dry air; HCR =1. ........ 126

5-6 An illustration of (a) temperature and (b) humidity ratio profiles in an dehumidifier with complete thermodynamic balancing by continuous injection. ....................................................... 128

5-7 Temperature profile representing the HDH system without extractions or injections. Boundary conditions: $T_a = 20^\circ C$; $T_c = 80^\circ C$; $\Psi_{deh} = \Psi_{hum} = 20$ kJ/kg dry air. ................................. 130

5-8 Temperature profiles representing the HDH system with continuous extractions to completely balance (a) dehumidifier and (b) humidifier. Boundary conditions: $T_a = 20^\circ C$; $T_c = 80^\circ C$; $\Psi_{deh} = \Psi_{hum} = 20$ kJ/kg dry air. ................................. 132

5-9 Temperature profile representing the HDH system with a single extraction. Boundary conditions: $T_a = 20^\circ C$; $T_c = 80^\circ C$; $\Psi_{deh} = \Psi_{hum} = 20$ kJ/kg dry air. ................................. 134

5-10 Mass flow rate ratio and HCR profile for complete thermodynamic balancing in a HDH system with 100% effective humidifier and dehumidifier. Boundary conditions: $T_a = 20^\circ C$; $S = 35$ g/kg; $T_c = 80^\circ C$; $\Psi_{deh} = \Psi_{hum} = 0$ kJ/kg dry air; $N = \infty$; System performance: GOR = 109.7; RR=7.6%. ......................... 136

5-11 Effect of having finite-size HME devices on the performance of the HDH system with infinite extractions highlighting the maximum possible uncertainty associated with using the saturation line as the air process path. Boundary conditions: $T_a = 20^\circ C$; $S = 35$ g/kg; $T_c = 80^\circ C$; $N = \infty$; HCR$_{deh}$=1. ................................. 138
5-12 Comparison of performance of the HDH system with infinite extractions for complete thermodynamic balancing of humidifier with that for complete thermodynamic balancing of the dehumidifier. Boundary conditions: $T_a = 20^\circ C; S = 35$ g/kg; $T_c = 80^\circ C; N = \infty; HCR_{deh} = 1.$

5-13 Reduction in total system irreversibility with complete thermodynamic balancing of either the humidifier or the dehumidifier in HDH. Boundary conditions: $T_a = 20^\circ C; S = 35$ g/kg; $T_c = 80^\circ C; \Psi_{deh} = \Psi_{hum} = 20$ kJ/kg dry air; $HCR_{deh} = 1$ or $HCR_{hum} = 1.$

5-14 Effect of number of extractions (for thermodynamic balancing) on the performance of the HDH system with finite and infinite size HME devices. Boundary conditions: $T_a = 20^\circ C; S = 35$ g/kg; $T_c = 80^\circ C; HCR_{deh} = 1.$

5-15 Effect of extraction on total system irreversibilities. Boundary conditions: $T_a = 20^\circ C; S = 35$ g/kg; $T_c = 80^\circ C; \Psi_{deh} = \Psi_{hum} = 20$ kJ/kg dry air; $HCR_{deh} = 1.$

6-1 Schematic diagram of a water-heated, closed-air, open-water humidification-dehumidification desalination system with mass extraction and injection of the moist air stream.

6-2 Three dimensional rendering of the experimental humidifier unit.

6-3 Effectiveness and heat capacity rate ratio versus water inlet temperature in the humidifier. Boundary conditions: $T_e = 32^\circ C; m_r = 2.85; T_{ub,e} = 20^\circ C; P = 101.3$ kPa; $V_h = 0.27$ m$^3$.

6-4 Effect of mass flow rate ratio on non-dimensional entropy generation in the humidifier. Boundary conditions: $T_e = 32^\circ C; T_c = 60^\circ C; T_{ub,e} = 20^\circ C; P = 101.3$ kPa; $V_h = 0.27$ m$^3$.

6-5 Effect of packing volume on the performance of the humidification device. Boundary conditions: $T_e = 32^\circ C; T_c = 60^\circ C; T_{ub,e} = 20^\circ C; P = 101.3$ kPa; $V_{block} = 0.07$ m$^3; m_r = 2.8$. 

17
Photograph of the 700 liter/day water-heated, closed-air, open-water humidification-dehumidification desalination system with mass extraction and injection of the moist air stream described in this chapter.

A state-of-the-art dehumidifier procured from George Fischer LLC

Effect of mass flow rate ratio on the performance of the HDH system without mass extractions. Boundary conditions: $T_a = 20^\circ C; T_c = 80^\circ C; N = 0; V_h = 0.27 \text{ m}^3; A_d = 8 \text{ m}^2$

Effect of top brine temperature on the performance of the HDH system without mass extractions. Boundary conditions: $T_a = 25^\circ C; \text{HCR}_d = 1; N = 0; V_h = 0.27 \text{ m}^3; A_d = 8 \text{ m}^2$

Effect of feed water temperature on the performance of the HDH system without mass extractions. Boundary conditions: $T_c = 90^\circ C; m_r = 2.4; N = 0; V_h = 0.27 \text{ m}^3; A_d = 8 \text{ m}^2$

Effect of mass flow rate of air extracted on the performance of the HDH system. Boundary conditions: $T_a = 25^\circ C; T_c = 90^\circ C; N = 1; V_h = 0.27 \text{ m}^3; A_d = 8 \text{ m}^2$

Effect of top brine temperature on the performance of the HDH system with a single air extraction from the humidifier to the dehumidifier. Boundary conditions: $T_a = 25^\circ C; \text{HCR} = 1; N = 1; V_h = 0.27 \text{ m}^3; A_d = 8 \text{ m}^2$

Effect of pressure on humidity ratio of saturated moist air.

Schematic diagram of mechanical compression driven HDH system.

Mechanical compression driven HDH cycle represented in psychrometric coordinates.

Effect of modified heat capacity ratio of humidifier on specific work and specific entropy generation. $T_{sw,in} = 30^\circ C; \varepsilon_H = \varepsilon_D = 80\%; \eta_{com} = \eta_e = 100\%; P_H = 40 \text{ kPa}; P_D = 48 \text{ kPa}$

Effect of component efficiency or effectiveness on cycle performance for $T_{sw,in} = 30^\circ C; P_H = 33.33 \text{ kPa}; P_D = 40 \text{ kPa}$
7-6 Effect of pressure ratio and dehumidifier pressure on cycle performance
for $T_{sw,in} = 30^\circ C; \varepsilon_H = \varepsilon_D = 80\%; \eta_{com} = \eta_e = 100\%$. ........................ 179

7-7 The effect of pressure ratio on specific net work and vapor productivity
ratio to explain the trend in Fig. 7-6. ................................. 179

7-8 The effect of dehumidifier pressure on specific net work and vapor
productivity ratio to explain the trend in Fig. 7-6. ........................ 180

7-9 Effect of air-side pressure drop in the humidifier on cycle performance
for $T_{sw,in} = 30^\circ C; \varepsilon_H = \varepsilon_D = 80\%; \eta_{com} = \eta_e = 100\%; P_D = 40$ kPa. 180

7-10 Effect of air-side pressure drop in dehumidifier on cycle performance
for $T_{sw,in} = 30^\circ C; \varepsilon_H = \varepsilon_D = 80\%; \eta_{com} = \eta_e = 100\%; P_{D,i} = 40$ kPa. 181

7-11 Effect of using a throttle versus using an air expander in the two pressure
cycle for $T_{sw,in} = 30^\circ C; \varepsilon_H = \varepsilon_D = 80\%; \eta_{com} = \eta_e = 100\%; \eta_e = 0$ or $100\%; P_D = 40$ kPa. ................................. 182

7-12 Entropy generation in the throttle and the air expander cycles for
$T_{sw,in} = 30^\circ C; \varepsilon_H = \varepsilon_D = \eta_{com} = \eta_e = 90\%; P_H = 40$ kPa; $P_D = 50$ kPa. 182

7-13 Thermal energy driven varied pressure HDH system ..................... 184

8-1 Black box model for steam driven thermal desalination system........... 188

8-2 Effect of the heating steam temperature (and correspondingly the total
entropy rate of steam) entering a thermal desalination system (shown
in Fig. 8-1) on the least thermal energy required to drive the system.
$T_0 = 30^\circ C; x_{st,in} = 1; T_{st,out} = 30^\circ C; \dot{m}_3 = 1$ kg/s; $S_1 = 35,000$ ppm; RR =
50\%. ................................................................. 190

8-3 Schematic diagram of thermal vapor compression driven HDH-RO sys-
tem. ................................................................. 192

8-4 Psychrometeric representation of thermal vapor compression driven
HDH system. ................................................................. 192

8-5 A typical embodiment of the TVC driven HDH-RO cycle. .................. 197
8-6  The effect of expander efficiency on performance of thermal vapor compression driven HDH-RO system and the specific entropy generated in the expander. $T_{sw,in} = 30^\circ C; \varepsilon_H = 60\%; \varepsilon_D = 70\%; \eta_{vc} = 30\%; P_{st} = 5\text{ MPa}; P_H = 86.96\text{ kPa}; P_D = 100\text{ kPa}; x_{st,in} = 1; \text{HCR}_H = 1.$

8-7  The effect of expander efficiency on relative importance of the HDH process to the thermal vapor compression driven HDH-RO system. $T_{sw,in} = 30^\circ C; \varepsilon_H = 60\%; \varepsilon_D = 70\%; \eta_{vc} = 30\%; P_{st} = 5\text{ MPa}; P_H = 86.96\text{ kPa}; P_D = 100\text{ kPa}; x_{st,in} = 1; \text{HCR}_H = 1.$

8-8  The effect of steam pressure on performance of thermal vapor compression driven HDH-RO system. $T_{sw,in} = 30^\circ C; \varepsilon_H = 60\%; \varepsilon_D = 70\%; \eta_{vc} = 30\%; \eta_e = 50\%; P_H = 86.96\text{ kPa}; P_D = 100\text{ kPa}; x_{st,in} = 1; \text{HCR}_H = 1.$

8-9  The effect of steam pressure on total entropy rate of steam entering and leaving the system, to explain the trends in Fig. 8-8.

8-10 The effect of steam pressure on vapor entrainment ratio and vapor productivity ratio, to explain the trends in Fig. 8-8.

8-11 Effect of pressure ratio on performance of thermal vapor compression driven HDH-RO system. $T_{sw,in} = 30^\circ C; \varepsilon_H = 60\%; \varepsilon_D = 70\%; \eta_{vc} = 30\%; \eta_e = 50\%; P_D = 100\text{ kPa}; P_{st} = 5\text{ MPa}; x_{st,in} = 1; \text{HCR}_H = 1.$

8-12 The effect of pressure ratio on vapor entrainment ratio and vapor productivity ratio, to explain the trends in Fig. 8-11.

8-13 Effect of humidifier pressure on performance of thermal vapor compression driven HDH-RO system. $T_{sw,in} = 30^\circ C; \varepsilon_H = 60\%; \varepsilon_D = 70\%; \eta_{vc} = 30\%; \eta_e = 50\%; P_D/P_H = 1.15; P_{st} = 5\text{ MPa}; x_{st,in} = 1; \text{HCR}_H = 1.$

8-14 The effect of humidifier pressure on vapor entrainment ratio and vapor productivity ratio, to explain the trends in Fig. 8-13.
8-15 Effect of air-side pressure drop in dehumidifier on performance of thermal vapor compression driven HDH-RO system. $T_{sw,in} = 30^\circ$C; $e_H = 60\%$; $e_D = 70\%$; $\eta_{vwc} = 30\%$; $\eta_e = 50\%$; $P_D = 100$ kPa; $P_{st} = 5$ MPa; $x_{st,in} = 1$; HCR$_H = 1$. ........................................ 208

8-16 Effect of air-side pressure drop in humidifier on performance of thermal vapor compression driven HDH-RO system. $T_{sw,in} = 30^\circ$C; $e_H = 60\%$; $e_D = 70\%$; $\eta_{vwc} = 30\%$; $\eta_e = 50\%$; $P_D = 100$ kPa; $P_{st} = 5$ MPa; $x_{st,in} = 1$; HCR$_H = 1$. ........................................ 208

8-17 Benchmarking of new HDH techniques against existing desalination systems. ........................................ 210

9-1 Schematic diagram of the bubble column dehumidifier. ................. 215

9-2 A thermal resistance model for the bubble column dehumidifier. .... 215

9-3 A mass transfer resistance model between the liquid in the column and the bubbles ........................................ 222

9-4 Schematic diagram of test apparatus; (1, 11, 12) valves, (2, 6, 13) rotameter, (3, 8) sparger, (4) humidifier column, (5) submerged electric heater, (7) pressure gauge, (9) dehumidifier column, (10) water coil, (14) inline water heater, (15) cooling water tank, (16) chilled water coil, (T1 – T8) thermocouples ........................................ 227

9-5 Photographs showing design of sparger and coil for (a) non-impact and (b) impact cases ........................................ 229

9-6 Effect of superficial velocity on the total heat flux in the bubble column measured and evaluated at $D_b = 4$ mm; $\chi_{in} = 21\%$; $H = 254$ mm. .... 231

9-7 Effect of bubble diameter on the total heat flux in the bubble column measured and evaluated at $V_g = 3.8$ cm/s; $\chi_{in} = 21\%$; $H = 254$ mm. .... 233

9-8 Effect of inlet mole fraction of the vapor on the total heat flux in the bubble column measured and evaluated at $V_g = 3.8$ cm/s; $D_b = 4$ mm; $H = 254$ mm. ........................................ 233

21
9-9 Effect of liquid height on the total heat flux in the bubble column measured and evaluated at \( V_g = 6.52 \, \text{cm/s}; D_b = 4 \, \text{mm}; \chi_{in} = 21\% \).

9-10 Parity plot of heat flux values evaluated by the model and that measured by experiments for various boundary conditions.

9-11 Bubble rise path in circular (left) and serpentine (right) coils.

9-12 Effect of bubble-on-coil impact.

9-13 Schematic diagram of multi-stage bubble column dehumidifier.

9-14 An illustration of the temperature profile in the bubble columns for (a) single stage and (b) multi-stage.

9-15 Prototype of multi-stage bubble column.

9-16 Effect of multistaging the bubble column on energy effectiveness of the device.

A-1 Schematic diagram for calculating Carnot GOR of HDH.

C-1 Flowchart of the overall HDH system design for the no extractions case.

C-2 Flowchart of the overall system design for the continuous air extractions case.

C-3 Flowchart of the overall system design for the single air extraction case.

D-1 Three-dimensional model of a trailer-mounted, sub-atmospheric-pressure, natural-gas-fired, single air extraction HDH system with a 12 foot tall packed bed humidifier and a four-stage bubble column dehumidifier.

E-1 Psychrometric chart for helium water vapor mixture and moist air.

E-2 Relative performance of water heated HDH cycle with helium or air as carrier gas. \( T_{sw,in} = 30^\circ\text{C}; T_{cg,H,\text{out}} = 90^\circ\text{C}; \varepsilon_H = 60\%; \varepsilon_D = 90\%; P = 100 \, \text{kPa} \).
List of Tables

2.1 Features and summary of results from various previous works on water heated CAOW HDH cycle. ......................................................... 45
2.2 Features and summary of results from various previous works on water heated CWOA HDH cycle. ......................................................... 49
2.3 Features and summary of results from various previous works on air heated CAOW HDH cycle. ......................................................... 52

3.1 Various definitions of effectiveness for simultaneous heat and mass exchange components. ................................................................. 68
3.2 Examples of maximum effectiveness for a counterflow cooling tower with following boundary condition: $T_{w,i} = 70^\circ C; p_{w,i} = 1 \text{ atm}, T_{a,i} = 30^\circ C; p_{a,i} = 1 \text{ atm}, \phi_i = 1.0.$ ................................................................. 97
3.3 Examples of maximum effectiveness for a counterflow cooling tower with following boundary condition: $T_{w,i} = 70^\circ C; p_{w,i} = 1 \text{ atm}, T_{a,i} = 30^\circ C; p_{a,i} = 1 \text{ atm}, \phi_i = 0.5.$ ................................................................. 98
3.4 Examples of maximum effectiveness for a counterflow air dehumidifier with following boundary condition: $T_{w,i} = 30^\circ C; p_{w,i} = 1 \text{ atm}, T_{a,i} = 70^\circ C; p_{a,i} = 1 \text{ atm}, \phi_i = 1.0.$ ................................................................. 100
3.5 Applicable range of various definitions of effectiveness of simultaneous heat and mass exchange devices. ......................................................... 102
4.1 Comparison of GOR of HDH cycles under representative boundary conditions ................................................................. 116
<table>
<thead>
<tr>
<th>Section</th>
<th>Title</th>
<th>Page</th>
</tr>
</thead>
<tbody>
<tr>
<td>6.1</td>
<td>Peak Performance of the pilot HDH system</td>
<td>167</td>
</tr>
<tr>
<td>7.1</td>
<td>Comparison of mechanical compression HDH with other HDH desalination technologies</td>
<td>183</td>
</tr>
<tr>
<td>9.1</td>
<td>Sparger design</td>
<td>228</td>
</tr>
<tr>
<td>9.2</td>
<td>Coil design for multi-stage prototype</td>
<td>241</td>
</tr>
<tr>
<td>D.1</td>
<td>Various components of capital expenditure (CAPEX) for a 10 m³ per day HDH system</td>
<td>275</td>
</tr>
<tr>
<td>E.1</td>
<td>Thermophysical properties of different carrier gases at STP and of dry saturated steam at atmospheric pressure</td>
<td>278</td>
</tr>
<tr>
<td>E.2</td>
<td>Estimated improvement in gas side dehumidification heat transfer coefficients when helium is used as the carrier gas.</td>
<td>282</td>
</tr>
<tr>
<td>E.3</td>
<td>Various system parameters and temperatures for cases shown in Fig. E-2284</td>
<td></td>
</tr>
</tbody>
</table>
Nomenclature

Acronyms
ED    Electrodialysis
GOR   Gained Output Ratio
HDH   Humidification Dehumidification
HME   Heat and Mass Exchanger
MSF   Multi-stage Flash
MED   Multi-effect Distillation
MVC   Mechanical Vapor Compression Distillation
RO    Reverse Osmosis
TBT   Top Brine Temperature
TTD   Terminal Temperature Difference
TVC   Thermal Vapor Compressor
RR    Recovery Ratio

Symbols
\(a_s\)  specific interfacial area of the bubble column (m\(^2\)/m\(^3\))
\(c\)    total molar concentration (mol/m\(^3\))
\(C\)    heat capacity rate (W/K)
\(c_p\)  specific heat capacity at constant pressure (J/kg·K)
\(c_{p,g}\) average specific heat capacity at constant pressure of the vapor-air mixture (J/kg·K)
\(D_b\)  bubble diameter (m)
\(d_o\)  sparger hole diameter (m)
\( D' \) \hspace{1em} \text{thermal diffusion coefficient (m}^2/\text{s-K)}

\( D_{AB} \) \hspace{1em} \text{diffusion coefficient (m}^2/\text{s)}

\( \text{ER}_{\text{vap}} \) \hspace{1em} \text{vapor entrainment ratio (-)}

\( \dot{E}_c \) \hspace{1em} \text{equivalent electricity consumption (kWh/m}^3\)

\( g \) \hspace{1em} \text{specific Gibbs energy (J/kg) or gravitational acceleration (m/s}^2\)

\( H \) \hspace{1em} \text{liquid height in the column (m)}

\( \dot{H} \) \hspace{1em} \text{total enthalpy rate (W)}

\( h \) \hspace{1em} \text{specific enthalpy (J/kg)}

\( h^* \) \hspace{1em} \text{Water and moist air enthalpy rate normalized by mass flow rate of dry air (J/kg dry air)}

\( h_a \) \hspace{1em} \text{specific enthalpy of moist air (J/kg of dry air)}

\( h_{fg} \) \hspace{1em} \text{enthalpy of evaporation (J/kg)}

\( \text{HCR} \) \hspace{1em} \text{heat capacity rate ratio (-)}

\( h_t \) \hspace{1em} \text{heat transfer coefficient for the sensible heat exchanged between bubble and liquid column (W/m}^2\cdot\text{K)}

\( I \) \hspace{1em} \text{specific entropy generation in the expander (J/kg.K)}

\( j \) \hspace{1em} \text{mass flux (kg/m}^2\cdot\text{s)}

\( k \) \hspace{1em} \text{thermal conductivity (W/m-K)}

\( k_l \) \hspace{1em} \text{mass transfer coefficient for the latent heat exchanged between bubble and liquid column (m/s)}

\( l \) \hspace{1em} \text{characteristic length (m)}

\( l_{\text{int}} \) \hspace{1em} \text{integral length for turbulence (m)}

\( m \) \hspace{1em} \text{mass flow rate (kg/s)}

\( m_r \) \hspace{1em} \text{water-to-air mass flow rate ratio (-)}

\( n \) \hspace{1em} \text{exponent (-)}

\( N \) \hspace{1em} \text{number of extractions (-)}

\( P \) \hspace{1em} \text{absolute pressure (N/m}^2\)

\( \dot{Q} \) \hspace{1em} \text{heat rate (W)}

\( R \) \hspace{1em} \text{gas constant (J/kg-K)}

\( \tilde{R} \) \hspace{1em} \text{universal gas constant (J/mol-K)}

26
The symbols and their meanings:

- $q$: heat flux (W/m$^2$)
- $q_{lt}$: heat flux due to condensation of vapor from the bubble in the liquid column (W/m$^2$)
- $q_{lt,impact}$: heat flux due to direct condensation of vapor from the bubble on the coil surface (W/m$^2$)
- $q_{sensible}$: heat flux due to the sensible heat exchange between bubble and liquid column (W/m$^2$)
- $R$: thermal resistance (K·m$^2$/W)
- $R_{bc}$: thermal resistance between the liquid column and the coil surface (K·m$^2$/W)
- $R_{coil}$: thermal resistance due to coolant flow inside the coil (K·m$^2$/W)
- $R_{bc}$: thermal resistance between the liquid column and the coil surface (K·m$^2$/W)
- $R_{coil}$: thermal resistance due to coolant flow inside the coil (K·m$^2$/W)
- $R_{sensible}$: thermal resistance for the sensible heat exchange between bubble and liquid column (K·m$^2$/W)
- $s$: specific entropy (J/kg·K)
- $S$: salinity or total dissolved solids (ppm)
- $\dot{S}$: total entropy rate of steam (W/K)
- $s_{fg}$: specific entropy change of evaporation (J/kg·K)
- $\dot{S}_{gen}$: entropy generation rate (W/K)
- $\dot{S}_{gen}$: entropy generation rate per unit volume (W/m$^3$·K)
- $t$: surface renewal time (s)
- $t_f$: average residence time of the bubble in the liquid (s)
- $T$: temperature (°C)
- $T_{air}$: local energy-averaged temperature of the air-vapor bubble (°C)
- $T_{coil}$: local temperature of the coil surface (°C)
- $T_{column}$: local energy-averaged temperature of the liquid in the column (°C)
- $T_{cooiant}$: local energy-averaged temperature of the coolant in the coil (°C)
- $u$: velocity in the x direction (m/s)
- $V$: velocity (m/s)
- $V_b$: bubble velocity (m/s)
\(V_c\) circulation velocity (m/s)
\(V_g\) superficial velocity (m/s)
\(V_r\) radial velocity in the liquid column (m/s)
\(vol\) volume of the bubble column (m³)
VPR vapor productivity ratio (-)
\(x\) quality of steam (-) or mass concentration (kg/kg)

Greek
\(\alpha\) thermal diffusivity (m²/s)
\(\Delta\) Difference or change
\(\epsilon\) volumetric gas holdup (-)
\(\epsilon\) component effectiveness (-)
\(\eta_{TVC}\) reversible entrainment efficiency for a TVC (-)
\(\eta_e\) isentropic efficiency for an expander (-)
\(\theta\) log mean temperature difference (°C)
\(\nu\) kinematic viscosity (m²/s)
\(\rho\) Density (kg/m³)
\(\sigma\) surface tension (N/m)
\(\phi\) Relative humidity (-)
\(\Psi\) enthalpy pinch (J/kg dry air)
\(\Psi_{TD}\) terminal enthalpy pinch (J/kg dry air)
\(\Psi_{local}\) local average enthalpy pinch (J/kg dry air)
\(\omega\) absolute humidity (kg/kg of dry air)

Subscripts
\(a\) humid air
\(c\) cold stream
\(D\) dehumidifier
\(da\) dry air
\(e\) expander
\(g\) gas
\( h \)  
hot stream

\( H \)  
humidifier

\( i \)  
inlet

\( in \)  
entering

\( max \)  
maximum

\( least \)  
least

\( l \)  
liquid

\( o \)  
outlet

\( out \)  
leaving

\( pw \)  
product water

\( rev \)  
reversible

\( RO \)  
reverse osmosis

\( s \)  
isentropic

\( sat \)  
saturated

\( st \)  
steam

\( tur \)  
steam turbine

\( tvc \)  
thermal vapor compressor

\( w \)  
water

**Superscripts**

\( ideal \)  
ideal condition

\( mod \)  
modified

\( rev \)  
reversible

**Non-dimensional Numbers**

\( Fr \)  
Froude Number, \( V_g^2/(g \cdot D_b) \)

\( Le \)  
Lewis Number, \( \alpha/D_{AB} \)

\( Le_t \)  
Lewis Factor, \( h_i/(\rho c_p g k_t) \)

\( Pr \)  
Prandtl Number, \( \nu/\alpha \)

\( Re \)  
Reynolds Number, \( (V_g \cdot D_b)/\nu \)

\( St \)  
Stanton Number, \( h/(\rho c_v V_g) \)
**Thermodynamic states**

a. Seawater entering the dehumidifier

b. Preheated seawater leaving the dehumidifier

c. Seawater entering the humidifier from the brine heater

d. Brine reject leaving the humidifier

e. Moist air entering the dehumidifier

ex. Moist air state at which mass extraction and injection is carried out in single extraction cases

f. Relatively dry air entering the humidifier

g. Air at an arbitrary intermediate location in the dehumidifier

i. Seawater at an arbitrary intermediate location in the dehumidifier
Chapter 1

Introduction

More than a billion people lack access to safe drinking water worldwide [3]. A large majority of these people live in low income communities. The United Nations acknowledges this fact in its millennium development goals [4] by highlighting the critical need for impoverished and developing regions of the world to achieve self-sustenance in potable water supply. Fig. 1-1 further illustrates how intense water scarcity exists mainly in the developing parts of the world\(^1\). For example, in India alone, 200,000 villages (and several peri-urban communities) lack access to safe potable water [5]. There is a clear need to help create a sustainable solution to the rural water problem in order to solve the global water crisis.

Most of the villages lacking safe drinking water are small communities with a population between 1,000 and 10,000 people. Thus, the water needs (for drinking and cooking) for each one of those communities is between 10 and 100 cubic meter of pure water per day (at a consumption rate of 10 liters per person per day). Systems that produce such amounts of pure water are relatively small-scale compared to conventional water treatment systems (for example, most existing state-of-the-art desalination systems are of the order of 100,000 to 1 million cubic meter per day [6]).

\(^1\)Often, lack of a potable water supply to the general population (or water scarcity) is misunderstood as absence of freshwater in a community. This is only one of the forms of water scarcity known as physical water scarcity. There is also scarcity in areas where there is plenty of rainfall and/or freshwater. This is primarily because of the lack of infrastructure to purify and transport the fresh water from aquifers or water bodies like lakes and rivers to the people who need it. This is termed as economic water scarcity.
Any potential small-scale solution to the problem needs to be both implementable and scalable. For the solution to be implementable, it has to be cost effective and resource-frugal. Currently, the price of safe drinking water (in the rare case it is available) in these low income communities is very high relative to the cost of tapped municipal drinking water in nearby “developed” regions (for example, in some parts of rural India the cost of water is up to 10 $/m³ which is roughly 40 times the cost of municipal drinking water available a few miles away in a nearby city [7]). Furthermore, in many villages, resources including skilled labor, a continuous energy supply, and raw materials are not readily available. The solution should, hence, be implementable within these constraints too.

An implementable solution is truly worthwhile only if it is scalable and can reach a large number of people (say, a million or more). For such scalability, the solution should be able to handle an array of contaminants in the water to be treated. In India alone, the contaminants range from high fluoride content to bacterial contamination to water being very brackish. Sixty six million people have been reported to be
consuming water with elevated levels of fluoride in India [8]. Most of these people live in the states of Rajasthan and Gujarat where fluoride contents reach up to 11 mg/L. Some districts in Assam, Orissa have very high iron content in water (1 to 10 mg/L - red water) and some in Rajasthan, Uttar Pradesh and Bihar have yellow water (>1 mg/L of iron) [9]. Certain places in Haryana, Gujarat, and Andhra Pradesh were also found to have dangerously high levels of mercury. The problems associated with high levels of Arsenic in ground water (in West Bengal) are well documented [10]. At least 300,000 people are affected by drinking water with arsenic above the permissible limit of 0.05 mg/L in this region. In parts of coastal Tamil Nadu, because of seawater intrusion, there is the problem of high salinity in drinking water (as high as 10,000 ppm in some cases) [11]. These problems are almost exclusively limited to rural and peri-urban communities. In all, almost one in three of the 600,000 Indian villages face problems of brackish or contaminated water and scarcity of fresh water. The India example is typical in that most developing and under-developed nations face similar water problems.

Desalination technologies are known to remove all contaminants including dissolved ions, micro organisms and so on. For example, as illustrated in Fig. 1-2, reverse osmosis removes even the smallest contaminants (albeit at a higher cost and complexity compared to other water treatment techniques). Furthermore, all thermal desalination technologies (MSF, MED, HDH and so on) are commonly known to remove all contaminants (producing what is in principle de-ionized water). The challenges in implementing these technologies are, however, to make them low cost (< 5 $/m³) at a community-scale (10-100 m³/day) and relatively maintenance-free (or maintained by non-technical laborers). The goal of the present thesis work is to develop a small-scale desalination technology which can meet these challenges.

1.1 Conventional desalination technologies

Desalination is generally performed by either of two main processes: by evaporation of water vapor or by use of a semi-permeable, non-porous membrane to separate fresh
water from a concentrate.

Figure 1-2: Illustration of various membrane technologies and the various contaminants they remove (Figure by MIT OCW; source: A. Twort et al. [2]).

The most important of these technologies are reviewed in this section. In thermal processes, the distillation of seawater is achieved by utilizing a heat source. The heat source may be obtained from a conventional fossil-fuel, nuclear energy or from a non-conventional source like solar energy or geothermal energy. In the membrane processes, electricity is used either for driving high-pressure pumps to overcome the osmotic pressure difference or for establishing electric fields to selectively migrate the ions.

The most important commercial desalination processes based on thermal energy
are multi-stage flash (MSF) distillation, multiple effect distillation (MED) and mechanical vapor compression (MVC). The MSF and MED processes consist of many serial stages at successively decreasing temperature and pressure. The MSF process is based on the generation of vapor from seawater or brine due to a sudden pressure reduction (flashing) and subsequent condensation of the vapor to produce pure water. The process is repeated stage-by-stage at successively decreasing pressures. This process requires an external steam supply as energy input, normally at a temperature of around 110°C. The maximum operating temperature is limited by scale formation, and thus the thermodynamic performance of the process is also limited. For the MED system, water vapor is generated by heating the seawater at a given pressure in each of a series of cascading chambers. Condensation of steam generated in one stage (or “effect”) is used to heat the brine in the next stage, which is at a lower pressure. The thermal performance of these systems is proportional to the number of stages, with capital cost and practical considerations like leakage and carbon-dioxide outgassing limiting the number of stages to be used.

In MVC systems, the energy for desalination is provided as a work transfer in a steam compressor as a work transfer which maintains the condensation at a higher pressure than the evaporation. The condensation of pure vapor provides the heat of evaporation in a single heat exchange where there is phase change in either stream exchanging energy. When an external source of steam is available, the compression process might be powered by it in a thermal vapor compressor (or TVC which is a steam-to-steam ejector) instead of providing a work transfer in a mechanical compressor.

The second important class of desalination processes, reverse osmosis (RO) and electrodialysis (ED), use a membrane to desalinate water. RO requires power to drive a pump that increases the pressure of the feed water to a desired value. The required pressure depends on the salt concentration of the feed water and the associated osmotic pressure. The ED process uses an electric potential to produce selective migration of ions through suitable ion-exchange membranes [12]. Both RO and ED are widely used for brackish water desalination; however, RO is also competitive with
MSF distillation processes for seawater desalination.

MSF plants typically have capacities ranging from 100,000 to almost 1,000,000 m³/day [13]. MED plants tend to be of a similar size and MVC plants are typically small-scale (about 1,000 m³ per day). The largest operational RO plant currently is the Ashkelon plant, at 330,000 m³/day [6]. RO is also applicable on a smaller scale at around 1,000 m³/day [14]. ED plants are of the order of 1,000 to 10,000 m³/day. Other approaches to desalination include processes like the ion-exchange process, liquid-liquid extraction, and the gas hydrate process. Most of these approaches are not generally used unless when there is a requirement to produce high purity (total dissolved solids < 10 ppm) water for specialized applications.

The MSF process represents more than 90% of the thermal desalination processes installed worldwide, while the RO process represents more than 80% of membrane processes for water production. Together MSF and RO currently account for around 70 to 80% of the total installed desalination capacity in the world. Most of this capacity is in large scale plants in the economically developed regions of the world (including the Middle East, Australia and USA).

1.2 Humidification Dehumidification (HDH)

Nature uses air as a carrier gas to desalinate seawater by means of the rain cycle. In the rain cycle, seawater gets heated (by solar irradiation) and evaporates into the air above to humidify it. Then the humidified air rises and forms clouds. Eventually, the clouds 'dehumidify' as rain over land and can be collected for human consumption. The man-made version of this cycle is called the humidification-dehumidification desalination (HDH) cycle. The simplest form of the HDH cycle is illustrated in Figure 1-3. The cycle constitutes of three subsystems: (a) the air and/or the brine heater (only a brine heater is shown in the figure), which can use various sources of heat like solar, thermal, geothermal or combinations of these; (b) the humidifier or the evaporator; and (c) the dehumidifier or the condenser.

The HDH cycle has received some attention in recent years and many researchers
have investigated the intricacies of this technology (see Chapter 2). It should be noted here that the predecessor of the HDH cycle is the simple solar still. Several researchers [15–17] have reviewed the numerous works on the solar still. It is important to understand the demerits of the solar still concept.

The most prohibitive drawback of a solar still is its low efficiency (Gained-output-ratio, or GOR\(^2\), less than 0.5) which is primarily the result of the immediate loss of the latent heat of condensation through the glass cover of the still. Some designs recover and reuse the heat of condensation, increasing the efficiency of the still. These designs (called multi-effect stills) achieve some increase in the efficiency of the still but the overall performance is still relatively low. The main drawback of the solar still is that the various functional processes (solar absorption, evaporation, condensation, and heat recovery) all occur within a single component. By separating these

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\(^2\)see Sec. 2.1 for definition of GOR
functions into distinct components, thermal inefficiencies may be reduced and overall performance improved. This separation of functions is the essential characteristic of the HDH system. For example, the recovery of the latent heat of condensation, in the HDH process, is affected in a separate heat exchanger (the dehumidifier) wherein the seawater, for example, can be preheated. The module for heat input (like a solar collector) can be optimized almost independently of the humidification or dehumidification component. The HDH process, thus, promises higher productivity due to the separation of the basic processes.

HDH systems are ideal for application in small-scale systems. They have no parts which require extensive maintenance like membranes or high temperature steam lines. There is also no bottleneck in applying HDH for varied and tough feedwater qualities.

The present thesis describes fundamental contributions to thermal design of HDH systems which make it affordable for application in small-scale desalination.

1.3 Organization of the thesis

Chapter 2 details a literature review of the HDH technology providing the specific objectives and scope of the present thesis work.\(^3\)

Chapter 3 describes methodology to model simultaneous heat and mass exchange devices (HME) including the humidifier and the dehumidifier and a new second law design algorithm for the same.

Chapter 4 uses the theory developed for HME devices in Chapter 3 and extends it to on-design modeling of existing HDH cycles. Chapters 5 to 8 describe novel system designs for HDH. These include HDH systems with mass extraction and injection (Chapter 5) and HDH systems powered by a mechanical compressor (Chapter 7) and a thermal vapor compressor (Chapter 8). Chapter 6 details experimental work on a pilot scale HDH unit.

\(^3\)All of the work reported in the present document has been written as 8 international journal papers (5 of them already published [18-22] and 3 of them are under review [23-25]), 3 conference papers [26-28] and 5 patent disclosures [29-33] (2 of them issued and 3 pending). All of these documents I am the first author or the lead inventor.
Chapter 9 describes the invention of a novel dehumidifier to reduce capital cost.
Chapter 2

Technology review

HDH systems are classified under three broad categories. One is based on the form of energy used such as solar, thermal, geothermal, or hybrid systems. This classification brings out a promising merit of the HDH concept: the promise of water production by use of low grade energy, especially from sources of industrial waste heat or from renewable resources like solar energy.

The second classification of HDH processes is based on the cycle configuration (Figure 2-1). As the name suggests, a closed-water open-air (CWOA) cycle is one in which ambient air is taken into the humidifier where it is heated and humidified and sent to the dehumidifier where it is partially dehumidified and let out in an open cycle as opposed to a closed air cycle wherein the air is circulated in a closed loop between the humidifier and the dehumidifier. In this cycle, the brine is recirculated until a desirable recovery is attained. The air in these systems can be circulated by either natural convection or mechanical blowers and feedwater is typically circulated by a pump. It is of pivotal importance to understand the relative technical advantages of each of these cycles and choose the one that is best in terms of efficiency and cost of water production.

The third classification of the HDH systems is based on the type of heating used - water or air heating systems. The performance of the system depends greatly on which fluid is heated.
2.1 Review of systems in literature

As a first step for understanding different works in literature the following performance parameters are defined.

1. Gained-Output-Ratio (GOR): is the ratio of the latent heat of evaporation of the water produced to the net heat input to the cycle.

\[
\text{GOR} \equiv \frac{\dot{m}_{pw} \cdot h_{fg}}{\dot{Q}_{in}} \quad (2.1)
\]

This parameter is, essentially, the effectiveness of water production, which is an index of the amount of the heat recovery affected in the system. This is the primary performance parameter of interest in HDH (and in thermal desalination, in general) and is very similar to the performance ratio (PR) defined for MED and MSF systems. For steam driven desalination systems (like in most state-of-the-art MSF and MED systems), PR is approximately equal to GOR.
GOR = \frac{\dot{m}_{pw} \cdot h_{fg}}{\dot{m}_s \cdot \Delta h_s} \quad (2.2)
\approx \frac{\dot{m}_{pw}}{\dot{m}_s} \quad (2.3)

It is worthwhile to note that GOR is also defined as the ratio of the latent heat \( h_{fg} \) to the specific thermal energy consumption. The latent heat in the equations above is calculated at the average partial pressure of water vapor (in the moist air mixture) in the dehumidifier.

2. Recovery ratio (RR): is the ratio of the amount of water produced per kg of feed. This parameter is also called the extraction efficiency [34]. This is, generally, found to be around 10% for the HDH system in single pass and can be increased to higher values (up to 90%) by brine recirculation.

\[ RR \equiv \frac{\dot{m}_{pw}}{\dot{m}_w} \quad (2.4) \]

2.1.1 Closed-air open-water (CAOW) water heated systems

A typical CAOW system is shown in Fig. 1-3. The humidifier is irrigated with hot water and the air stream is heated and humidified using the energy from the hot water stream. This process on the psychometric chart is represented by the line 1-2 (Fig. 2-2). The humidified air is then fed to the dehumidifier and is cooled in a compact heat exchanger using seawater as the coolant. The seawater gets preheated in the process and is further heated in a solar collector before it irrigates the humidifier. The dehumidified air stream from the dehumidifier is then circulated back to the humidifier. This process on the psychometric chart is represented by the line 2-1 (Fig. 2-2).

There are several works in literature on this type of cycle. The important features of the system studied and the main observations from these studies are tabulated in
Table 2.1.

Some common conclusions can be drawn from this table. Almost all the investigators have observed that the performance is maximized at a particular value of the water flow rate. There also is an almost unanimous consensus that natural circulation of air yields better efficiency than forced circulation of air for the closed air water heated cycle. However, it is not possible to ascertain the exact advantage in performance (for natural circulation) from the data available in literature.
Table 2.1: Features and summary of results from various previous works on water heated CAOW HDH cycle.

<table>
<thead>
<tr>
<th>Reference</th>
<th>Unit Features</th>
<th>Main observations</th>
</tr>
</thead>
</table>
| Al-Hallaj et al. [35] | - Solar collector (tubeless flat plate type of 2 m² area) has been used to heat the water to 50-70°C and air is circulated by both natural and forced convection to compare the performance of both these modes.  
- Humidifier, a cooling tower with wooden surface, had a surface area of 87 m²/m³ for the bench unit and 14 m²/m³ for the pilot unit.  
- Condenser area 0.6 m² for bench unit and 8 m² for the pilot unit. | - The authors noted that results show that the water flow rate has an optimum value at which the performance of the plant peaks.  
- They found that at low top temperatures forced circulation of air was advantageous and at higher top temperatures natural circulation gives better performance. |
| Ben Bacha et al. [36] | - Solar collector used for heating water (6 m² area)  
- There is a storage tank which runs with a minimum temperature constraint.  
- Cooling water provided using brackish water from a well.  
- The packed bed type - Thorn trees  
- Dehumidifier made of polypropylene plates. | - A daily water production of 19 litres was reported.  
- Without thermal storage 16% more solar collector area was reported to be required to produce the same amount of distillate.  
- The authors also stated that the water temperature at inlet of humidifier, the air and water flow rate along with the humidifier packing material play a vital role in the performance of the plant. |
| Farid et al. [37] | - 1.9 m² solar collector to heat the water.  
- Air was in forced circulation.  
- Wooden shaving packing used for the humidifier. | - 12 l/m² production achieved.  
- The authors report the effect of air velocity on the production is complicated and can not be stated simply.  
- The water flow rate was observed to have an optimum value. |
| Garg et al. [38] | • Their system has a thermal storage of 5 litre capacity and hence has longer hours of operations.  
  • Solar collector area (used to heat water) is about 2 m².  
  • Air moves around due to natural convection only.  
  • The latent is recovered partially.  
  • Indigenous structure claimed to be used for the packing in the humidifier. | • The authors conclude that the water temperature at the inlet of the humidifier is very important to the performance of the cycle.  
• They also observe that the heat loss from the distillation column (containing both the humidifier and the dehumidifier) is important in assessing the performance accurately. |
|---|---|---|
| Nafey et al. [39] | • This system is unique in the sense that it uses a dual heating scheme wherein there are separate heaters for both air and water.  
  • Humidifier is a packed bed type with canvas as the packing material.  
  • Air cooled dehumidifier is used and hence there is no latent heat recovery in this system. | • The authors reported a maximum production of 1.2 litre/hr and about 9 liters per day.  
• Higher air mass flow gave less productivity because increasing air flow reduced the inlet temperature to humidifier. |
| Nawayseh et al. [40] | • Three units constructed in Jordan and Malaysia. Different configurations of condenser and humidifier were studied.  
  • Solar collector heats up the water to 70-80 °C.  
  • Air circulated by both natural and forced draft.  
  • Humidifier with vertical/inclined wooden slates packing. | • The authors observed that the water flow rate has a major effect on the wetting area of the packing.  
• They also note that natural circulation yields better results than forced circulation.  
• The heat/mass transfer coefficient calculated were used to simulate performance and the authors report that the water production was up to 5 kg/hr. |
A unique HDH cycle with a direct contact packed bed dehumidifier was used in this study. The system uses waste heat to heat water to 60°C. It uses a part of the water produced in the dehumidifier as coolant and recovers the heat from this coolant in a separate heat exchanger.

The authors demonstrated that this process can yield a fresh water production efficiency of 8% with an energy consumption of 0.05 kWh per kilogram of fresh water production based on a feed water temperature of only 60°C. It should be noted that the efficiency is the same as the recovery ratio defined in the present thesis. Also the energy consumption does not include the solar energy consumed.

2.1.2 Closed-water open-air (CWOA) water heated systems

In a typical CWOA system the air is heated and humidified in the humidifier using the hot water from the solar collector and then is dehumidified using outlet water from the humidifier. The water, after being pre-heated in the dehumidifier, enters the solar collector, thus working in a closed loop. The dehumidified air is released to ambient.

The humidification process is shown in the psychometric chart (Fig. 2-3) by line 1-3-2. Air entering at ambient conditions is saturated to a point 2 and then the saturated air follows the line 2-3. The dehumidification process is shown by line 3-4. The air is dehumidified along the saturation line. A relatively small number of works in literature consider this type of cycle. The important features of the system studied and main observations from these studies are shown in Table 2.2.

One disadvantage of the CWOA is that when the humidification process does not cool the water sufficiently the coolant water temperature to the inlet of the dehumidifier goes up. This limits the dehumidification of the humid air resulting in a reduced water production compared to the open water cycle. However, when efficient humidifiers at optimal operating conditions are used, the water may be potentially
cooled to temperatures below the ambient temperature (up to the limit of the ambient wet-bulb temperature). Under those conditions, the closed water system is more productive than the open water system. Closed water (brine recirculation) is also desired in all HDH systems which aim for higher recovery ratios.

Figure 2-3: Water heated CWOA HDH process on psychometric chart.
Table 2.2: Features and summary of results from various previous works on water heated CWOA HDH cycle.

<table>
<thead>
<tr>
<th>Reference</th>
<th>Unit Features</th>
<th>Brief summary of the paper</th>
</tr>
</thead>
</table>
| Khedr [41]     | • The system comprises of a seawater heater, dehumidifier is regarded as a tower packed with 50mm ceramic Raschig rings.  
• The first system to use direct contact condenser in a HDH technology.  
• The performance parameters are calculated numerically.                                  | • The authors report the GOR for their system as 0.8 which shows that heat recovery is rather poor in their systems.  
• Based on an economic analysis, they conclude that the HDH Process has significant potential for small capacity desalination plants as small as 10 m$^3$/day. |
| Al-Enzi et al. [42] | • Solar collector designed to heat air to 90°C  
• Forced circulation of air.  
• Cooling water circuit for the condenser.  
• Heater for preheating water to 35-45°C.                                               | • The authors have studied the variation of production in kg/ day and heat and mass transfer coefficients with respect to variation in cooling water temperature, hot water supply temperature, air flow rate and water flow rate.  
• They conclude that the highest production rates are obtained at high hot water temperature, low cooling water temperature, high air flow rate and low hot water flow rate.  
• The variation in parameters the authors have considered is very limited and hence these conclusions are true only in that range. |
Honeycomb paper used as humidifier packing material.

Forced convention for the air circulation.

This works in a closer-water open-air cycle.

Condenser is fin tube type heat exchanger which also helps recover the latent heat by pre-heating seawater.

It was found that the performance of the system was strongly dependent on the temperature of inlet salt water to the humidifier, the mass flow rate of salt water, and the mass flow rate of the process air.

The authors report that there is an optimal air velocity for a given top temperature of water.

The top water temperature has a strong effect on the production of fresh water.

2.1.3 Closed-air open-water (CAOW) air heated systems

Another class of HDH systems which has attracted much interest is the air heated system. These systems are of two types - single and multi-stage systems. The air is heated in a solar collector to a temperature of 80 to 90°C and sent to a humidifier. This heating process is represented by the constant humidity line 1-2 in the psychometric chart (Fig. 2-4). In the humidifier, the air is cooled and saturated. This process is represented by the line 2-3. It is then dehumidified and cooled in the process 3-4 represented on the saturation line. A major disadvantage of this cycle is that the absolute humidity of air that can be achieved at these temperatures is very low (<6% by weight). This impedes the water productivity of the cycle.

Chafik [44] reported a method to address this problem. He used a multistage heating and humidification cycle (Fig 2-5). The air after getting heated in the solar collector (line 1-2) and humidified in the evaporator (line 2-3) is fed to another solar collector for further heating (line 3-4) and then to another humidifier (line 4-5) to attain a higher value of absolute humidity. Many such stages can be arranged to attain absolute humidity values of 15% and beyond. Point 2' in the figure represents the high temperature that has to be reached in a single stage cycle to attain the same
Figure 2-4: Single stage air heated CAOW HDH process on psychometric chart.

humidity as a 3 stage cycle. This higher temperature has substantial disadvantages for the solar collectors. However, from an energy efficiency point of view, there is not much of an advantage to multi-staging, as the higher water production comes with a higher energy input as compared to single stage systems.

Also, from the various studies in literature, we observe that the air-heated systems have higher energy consumption than water heated systems. This is because air heats up the water in the humidifier and this energy is not subsequently recovered from the water, unlike in the water-heated cycle in which the water stream is cooled in the humidifier.
Figure 2-5: Three stage air heated CAOW HDH process on psychometric chart.

Table 2.3: Features and summary of results from various previous works on air heated CAOW HDH cycle.

<table>
<thead>
<tr>
<th>Reference</th>
<th>Unit Features</th>
<th>Brief summary of the paper</th>
</tr>
</thead>
<tbody>
<tr>
<td>E. Chafik [44]</td>
<td>- Solar collectors (four-fold-web-plate (FFWP) design) of 2.08 m² area heats air to 50-80°C.</td>
<td>- The author reported that the built system is too costly and the solar air heaters constitute 40% of the total cost.</td>
</tr>
<tr>
<td></td>
<td>- Multi-stage system that breaks up the humidification and heating in multiple stages.</td>
<td>- Also he observed that the system can be further improved by minimizing the pressure drop through the evaporator and the dehumidifiers.</td>
</tr>
<tr>
<td></td>
<td>- Pad humidifier with corrugated cellulose material.</td>
<td></td>
</tr>
<tr>
<td></td>
<td>- 3 separate heat recovery stages.</td>
<td></td>
</tr>
<tr>
<td></td>
<td>Forced circulation of air.</td>
<td></td>
</tr>
</tbody>
</table>
| Ben-Amara et al. [45] | • FFPW collectors (with top air temperature of 90°C) were studied.  
• Polycarbonate plates and the blackened aluminium strips make up the solar collector.  
• Aluminium foil and polyurethane for insulation.  
| • Variation of performance with respect to variation in wind velocity, inlet air temperature and humidity, solar irradiation and air mass flow rate was studied.  
• Endurance test of the polycarbonate material showed it could not withstand the peak temperatures of summer and it melted. Hence a blower is necessary.  
• But minimum wind velocity gave maximum collector efficiency. |  |
| Orfi et al. [46] | • The experimental setup used in this work uses a solar heater for both air and water (has 2 m² collector surface area).  
• There is a heat recovery unit to preheat seawater.  
• The authors have used an evaporator with the heated water wetting the horizontal surface and the capillaries wetting the vertical plates and air moving in from different directions and spongy material used as the packing.  
| • The authors claim that there is an optimum mass flow rate of air to mass flow rate of water that gives the maximum humidification.  
• This ratio varies for different ambient conditions. |  |
| Yamali et al. [47] | • A single stage double pass flat plate solar collector heats the water.  
• A pad humidifier is used and the dehumidifier used is a tube-fin heat exchanger.  
• Also a tubular solar water heater was used for some cases.  
• The authors also used a 0.5 m³ water storage tank.  
• No heat recovery.  
| • The plant produced 4 kg/day maximum.  
• Increase in air flow rate had no affect on performance.  
• An increase in mass flow rate of water increased the productivity.  
• When the solar water heater was turned on the production went up to 10 kg/day maximum primarily because of the ability to operate it for more time. |  |
2.1.4 HDH systems with mass extractions and injections

A schematic diagram of an embodiment of the HDH system with mass extractions and injections is shown in Fig. 2-6. The system shown here is a water-heated, closed-air, open-water system with three air extractions from the dehumidifier into the humidifier. States a to d are used to represent various states of the seawater stream and states e and f represent that of moist air before and after dehumidification.

Even though there has been no clear conceptual understanding of how the thermal design of HDH systems with mass extraction/injection should be carried out, a small number of studies in literature discuss limited performance characteristics of these systems. Müller-Holst pioneered the thermal balancing of HDH systems by proposing to balance the stream-to-stream temperature difference in the HME devices by ‘continuous’ variation of the water-to-air mass flow rate ratio [49, 50]. The moist...
air in the proposed system was circulated using natural convection and the mass flow rate of this stream was varied by strategically placed extraction and injection ports in the humidifier and the dehumidifiers respectively. An optimized GOR of these systems was reported to be between 3 and 4.5.

Zamen et al. [51, 52] reported a novel ‘multi-stage process’ which was designed for varying the water-to-air mass flow rate ratio. This was achieved by having multiple stages of humidification and dehumidification in series with separate air flow for each stage and a common brine flow. A similar design was also reported by Schlickum [53] & Hou [54]. Zamen et al. [51] used a temperature pinch (defined as the minimum
stream-to-stream temperature difference in the HME device) between the water and the air streams to define the performance of the system. For a four stage system with component pinch of 4°C, at a feed water temperature of 20°C and a top brine temperature of 70°C, Zamen et al. reported an energy consumption of slightly less than 800 kJ/kg (corresponding to a GOR of 3).

Brendel [55, 56] invented a novel forced convection driven HDH system in which water was extracted from several locations in between the humidifier and sent to corresponding locations in the dehumidifier. This was done such that the temperature profiles were balanced (as was the case with Zamen et al. [51]). In chapter 5 of the present thesis we have shown that in order to attain the thermodynamic optimum in HME devices we have to consider both temperature, and concentration profiles and that the optimum lies closer to a balanced humidity profile than a balanced temperature profile.

Younis et al. [57] studied air extraction and injection in forced convection driven HDH systems. They found that having two extractions of air from the dehumidifier to the humidifier decreased the energy consumption of the system to 800 kJ/kg. Like in several other publications [49, 51, 55, 58], they used enthalpy-temperature diagrams to demonstrate the effect of extraction on HDH system design. McGovern et al. [58] pinoored the use of the graphical technique and highlighted the important approximations that need to be made to use it for HDH system design.

Bourouni [59] in a review of the HDH technology reported that a few other authors [60] studied air extractions in HDH systems and reported performance enhancements as a result of such a design. However, these studies are dated and no longer available in open literature.
2.2 Alternate cycles resembling the HDH process

2.2.1 Dew-vaporation technique

Beckmann and coworkers have invented [61] and investigated [62, 63] a desalination technology that works on the humidification dehumidification principle. They call it the ‘Dew-vaporation’ technique (figure 2-7). Unlike the HDH process, it uses a common heat transfer wall between the humidifier (which they call the evaporation chamber) and the dehumidifier (which they call the dew formation chamber). The latent heat of condensation is directly recovered through this wall for the humidification process. It is reported that the use of this common heat transfer wall makes the process energy efficient.

In this process the saline water, after being preheated using the exit distillate water stream, wets the heat transfer wall and is heated by means of the latent heat of condensation from the dew-formation chamber. It then evaporates into the air stream, humidifying it. The humidified air stream is then heated using an external source and is fed to the dehumidifier at a temperature higher than the temperature of air leaving the humidifier. While, heat is directly recovered from the dew-formation tower, it should be noted that the condensation process itself is relatively ineffective. The dehumidified air exits the tower at a high temperature of around 50°C (compared to 30-35°C in a HDH cycle). Also, the coupling of the humidification and dehumidification processes sacrifices the modularity of the HDH system and the related opportunities to optimize subsystem design and performance separately. Since a single wall is used for both dehumidification and humidification there is hardly any optimization possible to the heat exchange surfaces which can improve the transfer rates (for example, the ‘equivalent’ heat transfer coefficient in the dehumidification process was in some cases is as low as 1 W/m²·K). However, Hamieh et al. [62] has reported that the energy efficiency of this system is higher than regular HDH systems (a GOR of about 5).
2.2.2 Diffusion-driven desalination technique

Investigators at University of Florida have patented [34] an alternate desalination process that works on the HDH principle. They call it the 'diffusion driven desalination' (DDD) process. The system is similar to the closed-air open-water HDH cycle, but it uses a direct contact dehumidifier in place of the non-contact heat exchanger normally used for condensation in the HDH systems. The dehumidification process uses a portion of the distilled water produced from the cycle as a coolant. A chiller is used to provide the distilled water at a low temperature. In a similar system, Khedr [41] had earlier proposed an HDH system with a direct contact dehumidifier having
ceramic Raschig rings as the packing material. The specific energy demand of the DDD process (GOR is about 1.2) is higher for this cycle than for a normal HDH cycle. This is because the latent heat of condensation is captured in two steps (from the moist air stream to the pure water stream in the direct contact device and then from the pure water stream to the brine in a liquid-to-liquid heat exchanger).

2.2.3 Atmospheric water vapor processers

Wahlgren [64] reviewed various processes that extract the humidity from ambient air. These processes are called dew collection processes and the system is sometimes called an atmospheric vapor processor. Three different methods have been applied in these systems: (1) surface cooling using heat pumps or radiative cooling devices; (2) using of solid/liquid desiccants to concentrate the moisture in atmospheric air before condensing it out; and (3) convection-induced dehumidification in a tower structure.

While it may seem promising to take advantage of air that is already humidified and a cycle which consists of only dehumidification (which is by itself exothermic), some major drawbacks accompany this concept of water extraction. The absolute humidity in ambient air found in most places around the world is low, and hence to produce a reasonable amount of water a large amount of air needs to circulate through the process equipment. Also, even though the dehumidification process is exothermic the possibility of extracting any thermodynamic advantage from it exists only when a lower temperature sink is available. On the contrary, the energy consumption of these devices has been reported to be much higher than HDH systems. The theoretical minimum energy consumption for these systems is 681 kWh/m³. The actual energy consumption is bound to be much higher than the limit. Even this reversible limit is higher than the energy consumption of existing HDH systems.
2.3 Review of components in the HDH system

2.3.1 Humidifiers

Many devices are used for air humidification including spray towers, wetted-wall towers and packed bed towers [65]. The principle of operation for all of these devices is the same. When water is brought into contact with air that is not saturated with water vapor, water evaporates into air and raises the humidity of the air. The driving forces for this humidification process are the concentration and temperature differences between the water-air interface and the water vapor in air. The concentration difference depends on the vapor pressure at the gas-liquid interface and the partial pressure of water vapor in the air.

Any of the above mentioned devices can be used as a humidifier in the HDH system. A spray tower for instance consists essentially of a cylindrical vessel in which water is sprayed at the top of the vessel and moves downward by gravity dispersed in droplets within a continuous air stream flowing upward. These towers are simple in design and have minimal pressure drop on the gas side. It is generally known that this device has high capacity but low efficiency. The low efficiency is as a result of the low water holdup due to the loose packing flow [66]. The diameter-to-length ratio is a very important parameter in spray tower design. For a large ratio air will be thoroughly mixed with the spray. Small diameter-to-length ratio will let the spray quickly reach the tower walls, forming a film becoming ineffective as a spray. Design of spray towers requires knowledge of heat and mass transfer coefficients as well as the contact surface area of the water droplets. Many empirical correlations and design procedures are given in Kreith and Boehm [66].

Younis et al. [57] and Ben-Amara et al. [45] used a spray tower as the humidifier in their HDH systems. Ben-Amara et. al tested the spray tower humidifier by varying the ratio of water-to-dry air mass flow rate and keeping the inlet water temperature and absolute humidity constant. The inlet air temperature (80°C) was higher than the water spray temperature (60°C). They found that increasing the amount of water sprayed increased the absolute outlet humidity. However, further increase in the water
quantity resulted in air cooling and this condensed some of the water vapor content in the air. This means a decrease in the absolute humidity, although the outlet air is always saturated. Therefore, for air heated HDH cycles there is an optimum value of the mass flow ratio which gives maximum air humidity. This fact promotes the use of multi-stage air heater and humidifier combinations to increase the fresh water production.

Wetted-wall towers have been used as humidifier in HDH systems by Müller-Hölst [49] and Orfi et al. [46]. In a wetted-wall tower, a thin film of water is formed running downward inside a vertical pipe, with air flowing either co-currently or counter-currently. Water is loaded into the top of the tower and a weir distributes the flow of water around the inner perimeter of the tube that wets the inner surface of the tube down its length. Such devices have been used for theoretical studies of mass transfer, since the contact area can be calculated, accurately. In Müller-Hölst’s system, heated water was distributed onto vertically hanging fleeces made of polypropylene and trickled downward. The air move in countercurrent flow to the brine through the humidifier and becomes saturated at the outlet. On the other hand, Orfi et al. [46] used a different design for their wetted-wall humidifier. To improve the heat and mass exchange process, they covered the wooden vertical wetted-walls with a cotton wick to reduce the water flowing velocity and use the capillary effect to keep the vertical walls always wetted. Their design shows higher performance with about 100% humidification efficiency.

To increase the humidification efficiency, packing is typically used. This helps by increasing the dispersion of water droplets, the contact area and contact time. Devices that contain packing material are known as packed bed towers and special types that are used to cool water are called cooling towers. These are vertical columns filled with packing materials with water sprayed at the top and air flows in counter or cross flow arrangement. Packed bed towers have been used by many researchers as a humidifier device in HDH desalination systems because of the higher effectiveness. Different packing materials have been used as well. The factors influencing the choice of a packing are its heat and mass transfer performance, the quality of water, pressure
drop, cost and durability.

In general, all of these humidification devices are relatively low cost at reasonable performance unlike the dehumidifiers which are known to have low heat and mass transfer rates due to the presence of large concentrations of non-condensable gas (equivalent heat transfer coefficients as low as 1 W/m²-K) in what is typically a film condensation process. These dehumidifiers are reviewed in the following paragraphs.

2.3.2 Dehumidifiers

The types of heat exchangers used as dehumidifiers for HDH applications vary. For example, flat-plate heat exchangers were used by Müller-Hölst et al. [49]. Others used finned tube heat exchangers [44, 46]. A long tube with longitudinal fins was used in one study [37], while a stack of plates with copper tubes mounted on them in another study [40]. Bourouni et al. [67] used a horizontal falling film-type condenser.

A flat plate heat exchanger made of double webbed slabs of propylene was used by Müller-Hölst [49] in his HDH system. The distillate runs down the plates trickling into the collecting basin. Heat recovery is achieved by transferring heat to the cold seawater flowing inside the flat plate heat exchanger. The temperature of seawater in the condenser increases from 40°C to 75°C. In a similar study, Chafik [44] used seawater as a coolant wherein the water is heated by the humid air before it is pumped to the humidifiers. Three heat exchangers were used in three different condensation stages. An additional heat exchanger is added at the intake of seawater (low temperature level) for further dehumidification of air. The heat exchangers (or dehumidifiers) are finned tube type air coolers. It is important to note that to withstand the corrosive nature of seawater; stainless steel is used for frames, collecting plates, while the fins are made of aluminum. In addition, special attention was exercised to avoid leakage of distillate water.

Different designs of condensers in a HDH cycle were used by Farid et al. [37]. In a pilot plant built in Malaysia, the dehumidifier was made of a long copper-galvanized steel tube (3 m length, 170 mm diameter) with 10 longitudinal fins of 50 mm height on the outer tube surface and 9 fins on the inner side. In another location, they used
a simplified stack of flat condenser made of 2 x 1 m² galvanized steel plates with long copper tubes mounted on each side of the plate to provide a large surface area. The condenser size was made large, particularly to overcome the small heat transfer coefficients both on the air- and water-sides due to relatively low air velocity, as well as low water flow rates.

In another design, the dehumidifier was made of 27 m long copper pipe having a 10 mm OD, mechanically bent to form a 4 m long helical coil fixed in the PVC pipe. The preheated feed water was further heated in a flat plate collector. The hot water leaving the collector was uniformly distributed over a wooden shaving packing in a 2 m long humidifier. It is important to note that the condenser or dehumidifier was made of hard PVC pipes connected to form a loop with the blower fixed at the bottom. The condenser was made of a copper pipe mechanically bent to form a helical coil fixed in the PVC pipe.

Bourouni et al. [67] used a condenser made of polypropylene which was designed to work at low temperatures (70-90°C) for a HDH system. It is similar to a horizontal falling film-type condenser. At the top of the dehumidifier, the hot humid air is forced down where the distilled water is recovered. It is important to note that heat recovery in an HDH system requires a larger heat transfer area for improving the overall system performance. For this reason, 2000 m of tubes are used in the evaporator, while 3000 m of tubes in the condenser.

The system Orfi et al. [46] used had two solar heaters, one for heating water and the other for heating air. The condenser, that uses seawater for cooling, consists of a chamber with a rectangular cross section. It contains two rows of long cylinders made of copper in which the feed water flows. Longitudinal fins were soldered to the outer surface of the cylinders. The condenser is characterized by heat-transfer surface area of 1.5 m² having 28 m as a total length of the coil.
2.4 Performance benchmarking of HDH systems

A detailed literature review has been conducted in this chapter. Based on this review, we try to benchmark the key performance metrics of existing HDH systems: (1) the cost of water production; (2) the heat and mass transfer rates in the dehumidifier; and (3) the system energy efficiency (GOR).

The total cost of water production in HDH systems is mostly a sum of the energy cost (captured by the GOR of the system) and the capital cost. A large fraction of the capital investment in typical HDH systems is the dehumidifier cost. This is driven by the low heat and mass transfer rates in the device. It has been reported that the ‘equivalent’ heat transfer coefficient in the dehumidifier is between 1 and 100 W/m²-K [42, 62]. This is two orders of magnitude lower than for pure vapor condensers.

Using the data given in various papers, GOR for the reported systems was calculated. It was found that the maximum GOR among existing HDH systems was about 3. Figure 2-8 illustrates the GOR of a few of the studies. The GOR varied between 1.2 to 3. These values of GOR translate into energy consumption rates from 215 kWh/m³ to 550 kWh/m³. The low value of GOR achieved by Ben Bacha et al. [36] was because they did not recover the latent heat of condensation. Instead, they used separate cooling water from a well to dehumidify the air. Lack of a systematic understanding of the thermal design of HDH systems which can help optimize performance is the reason behind such inefficient designs. The higher value of GOR achieved by Müller-Hölst et al. [49] was because of high heat recovery (using the concept of thermodynamic balancing explained in Chapter 5). These results tell us the importance of maximizing heat recovery in minimizing the energy consumption and the operating and capital cost of HDH systems. It is also to be noted that the GOR varied from 3 to 4.5 in Müller-Hölst’s system because of the inability of that system to independently control the air flow by natural convection. It is, hence, desirable to develop forced convection based systems which have a sustainable peak performance.

---

1The system is almost maintenance-free.
Based on a simple thermodynamic calculation, the GOR of a thermodynamically reversible HDH system can be evaluated to be 122.5 for typical boundary conditions (see Appendix A). When compared to a GOR of 3 for existing systems, the reversible GOR of 122.5 shows that there is significant potential for improvement to existing HDH systems in terms of reducing thermodynamic losses, and this gives ample motivation to study the thermal design of these systems in detail.

![Graph showing performance of HDH systems in literature.](image)

**Figure 2-8: Performance of HDH systems in literature.**

A few studies in literature actually report the overall cost of water production in a HDH system [44, 48, 49]. This cost is found to be about $30 per cubic meter of water produced which is very high. The central goal of the science developed in this thesis is to reduce this cost to affordable levels (<$5 per cubic meter).

### 2.5 Unanswered questions

Based on the literature review presented in the current chapter, it is clear that to reduce the overall cost of water production to affordable levels in HDH systems, several unanswered issues need to be addressed. These include:

1. Development of systematic methodologies for modeling the humidifier and the dehumidifier in the context of the overall HDH system design and analysis;
2. Development of thermal design algorithms to minimize thermodynamic losses in the humidifier and the dehumidifier;

3. Understanding the thermodynamic relationships that govern the efficiency of various embodiments of HDH;

4. Applying the aforementioned understanding to developing energy efficient versions of the HDH cycle;

5. Developing detailed methodology for design of forced convection based HDH systems with mass extractions and injections; and

6. Developing methods to increase the heat and mass transfer rates in the dehumidifier.

In the following chapters, all of these issues are addressed.
Chapter 3

Thermal design of simultaneous heat and mass exchange (HME) devices

A simultaneous heat and mass exchanger (HME) is a device that is used to transfer heat and mass between two fluid streams at different temperatures and concentrations. Thermal contact between the fluid streams will occur through direct contact of the streams if mass is transferred between them or through indirect contact via a heat transfer surface if the mass transfer is associated with phase change in just one stream with no mass transfer between the two streams. Accordingly, they are classified as direct contact devices (e.g., packed bed humidifiers and cooling towers) and indirect contact devices (e.g., shell-and-tube dehumidifiers and cooling coils). Understanding the thermodynamics of these devices is crucial to optimizing the thermal performance of HDH systems. This also has application in other fields of thermal design (such as cooling towers for thermal power plants and cooling coils in air conditioning systems).

3.1 Control volume models

As mentioned above, HME devices are widely used in power generation, desalination, air conditioning, and refrigeration systems, and their performance is vital to the
Table 3.1: Various definitions of effectiveness for simultaneous heat and mass exchange components.

<table>
<thead>
<tr>
<th>Reference</th>
<th>Effectiveness defined</th>
<th>Defined for</th>
</tr>
</thead>
<tbody>
<tr>
<td>Mandi et al. [68]; Cheremisinoff and Cheremisinoff [71]</td>
<td>$\varepsilon_T = \frac{\Delta T}{\Delta T_{ideal}}$</td>
<td>Cooling towers</td>
</tr>
<tr>
<td>Nellis and Klien [72]</td>
<td>$\varepsilon_w = \frac{\Delta \omega}{\Delta \omega_{ideal}}$</td>
<td>Dehumidifier</td>
</tr>
<tr>
<td>Nellis and Klien [72]</td>
<td>$\varepsilon_h = \frac{\Delta h_a}{\Delta h_{a,ideal}}$</td>
<td>Dehumidifier</td>
</tr>
<tr>
<td>Jaber and Webb [73]</td>
<td>$\varepsilon_J = \frac{\dot{Q}<em>{act}}{\dot{m}</em>{min} \cdot \Delta h_{a,ideal}}$</td>
<td>Cooling towers</td>
</tr>
</tbody>
</table>

overall system performance. For example, the thermal performance of cooling towers is critical to the overall performance of steam or combined cycle power plants in which they are used [68, 69]. For analysis and optimization of cycles containing these components, defining an effectiveness to characterize their performance has considerable advantages [18, 19, 70]. It is, therefore, important to critically examine various definitions of effectiveness that are used in the literature.

In HME literature, several definitions for effectiveness are in use, but they are specific to certain configurations and boundary conditions. The effectiveness definitions that are used for cooling towers, humidifiers, and cooling coils are summarized in Table 3.1.

In cooling tower literature [68, 71], an effectiveness is commonly defined based on the temperature change of one of either the air or the water streams, typically, the change in water temperature. This definition can also be written using terminology
commonly used in cooling towers: (a) **Range**, the change in water temperature between the inlet and the outlet; and (b) **Approach**, the difference between the water exit temperature and the inlet air wet-bulb temperature:

\[ \varepsilon_T = \frac{\text{Range}}{\text{Range} + \text{Approach}} \]  

Nellis and Klien [72] present a modified definition of effectiveness for a cooling coil (dehumidifier) based on humidity ratio and specific enthalpy of the moist air stream, and, they provide several examples of the use of these effectivenesses for cooling coil design in their textbook.

Jaber and Webb [73] proposed a modified definition of effectiveness based on an analogy between counterflow heat exchangers and counterflow cooling towers. They defined the maximum possible heat transfer rate as the product of the minimum modified mass flow rate \( \dot{m}_{\text{mod}} \) and the maximum air side enthalpy potential difference:

\[ \dot{m}_{\text{mod}} = \frac{\dot{m}_{w}c_{p,w}}{f'} \]  

where

\[ f' = \frac{dh_{a,sat}}{dT_w} \]  

\[ \dot{m}_{\text{mod}} = \min(\dot{m}_{w,\text{mod}}, \dot{m}_{da}) \]  

where \( f' \) is the average slope of the saturated moist air enthalpy with water temperature. It should be noted that they neglect the effect of evaporation on \( \dot{m}_w \) and assume that the saturation enthalpy curve for moist air is linear with temperature. This definition is examined in more detail in a succeeding section (Sec. 3.1.1).

The above literature review suggests a need to critically examine the existing effectiveness definitions and to identify one which is robustly applicable to a large class of HME devices. Thus, this paper seeks to answer the following questions: Can a single definition of effectiveness be given which will apply to all types of HME devices? For what situations can one apply the existing definitions of effectiveness? What is the significance of the heat capacity rate ratio to HME devices and how
can it be defined without any approximations about fluid properties? What is the analogy of a balanced heat exchanger in the case of a heat and mass exchanger? How does thermal balancing affect the effectiveness definitions? Which is the 'balanced' thermodynamic state in a HME devices?

3.1.1 New heat and mass exchanger (HME) terminology

In this section, new terminology for HME devices is defined. This includes an energy-based effectiveness and a modified heat capacity rate ratio.

Energy effectiveness

An energy based effectiveness, analogous to the effectiveness defined for heat exchangers, is given in Eq. (3.5):

$$
\varepsilon = \frac{\Delta \dot{H}}{\Delta \dot{H}_{\text{max}}} \tag{3.5}
$$

This definition is based on the maximum change in total enthalpy rate that can be achieved in an adiabatic heat and mass exchanger. It is defined as the ratio of change in total enthalpy rate to maximum possible change in total enthalpy rate. The maximum possible change in total enthalpy rate can be of either the cold or the hot stream, depending on the heat capacity rate of the two streams. The stream with the minimum heat capacity rate dictates the thermodynamic maximum that can be attained. To elucidate this concept, consider the example of a counterflow cooling tower.

Figure 3-1 illustrates the Second Law limitations imposed on such a device. Here, 'wb,1' is the wet-bulb point of the air at the inlet to the humidifier and 'a,2' is the exit air state. The air is assumed to be saturated at the inlet and hence, $T_{wb,1} = T_{a,1}$. The saturation line connecting the point 'wb,1' to 'a,2' represents one possible process path for the humidification process.

The maximum dry bulb temperature that can be achieved by the saturated air at the exit of the humidifier is the water inlet temperature (indicated by point 'a,3'). From Fig. 3-1, it is seen that the maximum enthalpy change possible ($\Delta \dot{H}_{\text{max}}$) for
Figure 3-1: Psychometric chart illustrating an example of the Second Law limits on a counterflow cooling tower operation.

saturated air entering the humidifier occurs if the air can be brought to saturation at the water inlet temperature. The required energy is drawn from the water stream, which may or may not have the capacity rate \( \dot{m}_w c_{p,w} \) necessary to supply that amount of energy within the limits imposed by the air and water inlet temperatures. If the water stream lacks sufficient capacity, the maximum change in total enthalpy rate \( \Delta \dot{H}_{\text{max}} \) will be that which cools the water to the air inlet temperature. In this case the outlet air will be cooler than the water inlet temperature, and it may or may not be saturated. That is, for any given case, a particular range of exit relative humidities is possible (corresponding to points from 'a,2' to 'a,2' shown in Fig. 3-1). Hence, another parameter is required to fix the exit state of the air apart from effectiveness. In this analysis, the exit relative humidity is treated as a free design variable which can be controlled by adjusting the dimensions of, say, the packing of the cooling tower.
It is important to note that the energy effectiveness concept applies to all types of HME devices, not simply counterflow cooling towers. This includes humidifiers and dehumidifier used in HDH systems.

Heat capacity rate ratio

In the limit of infinite heat transfer area for a pure heat exchanger, the entropy generation rate in the exchanger is entirely due to what is known as thermal imbalance or remanent irreversibility. This thermal imbalance is associated with conditions at which the heat capacity rate ratio is not equal to unity [74]. In other words, a heat exchanger is said to be thermally 'balanced' at a heat capacity rate ratio of one. This concept of thermodynamic balancing, even though very well known for heat exchangers, was only recently extended to HME devices (and is reported in the Sec. 3.2 of this chapter). It is important to establish a reliable definition for the heat capacity rate ratio for an HME in order to understand its influence on selecting the appropriate definition of effectiveness.

In some cases, the heat capacity rate of one of the streams of a simultaneous heat and mass exchangers cannot be defined readily. For example, in a counterflow cooling tower (where hot water is losing heat and mass to humid air), it is not possible to define the change in enthalpy of the moist air stream as the product of specific heat capacity at constant pressure and change in temperature because the humidity ratio also affects moist air enthalpy, \( h_a \):

\[
h_a = f\{T, p, \omega\}
\]  

(3.6)

If the effect of pressure variation on enthalpy is neglected,

\[
dh_a = c'_p, a \cdot dT + \left( \frac{\partial h_a}{\partial \omega} \right)_{p,T} \cdot d\omega
\]  

(3.7)

where

\[
c'_p, a \equiv \left( \frac{\partial h_a}{\partial T} \right)_{p,\omega}
\]  

(3.8)
Hence, the heat capacity rate ratio cannot be calculated as it is for heat exchangers.

Braun et al. [75] found a way around this problem by defining an effective heat capacity of the moist air stream as slope of the saturated specific enthalpy line evaluated at water temperatures:

\[ C_{pA} = \frac{\partial h_{a,\text{sat}}}{\partial T_w} \]  

(3.9)

which is identical to the enthalpy correction factor \( f' \), Eq. (3.3), defined by Jaber and Webb [73]. This approximation has reasonable accuracy (under certain operating conditions) for cooling tower cases. However, in certain cases where the humidity levels are relatively high (e.g., in a direct contact counterflow air humidifier or an indirect contact dehumidifier), a sizable error is induced.

Therefore, we introduce a modified heat capacity ratio based on the total enthalpy rate change which is accurate in all ranges of humidity and temperature levels. This is defined using an analogy to heat exchangers. For heat exchangers,

\[ HCR_{HE} = \frac{\dot{m}_c c_{p,c}}{\dot{m}_h c_{p,h}} \]  

(3.10)

This can be rewritten as

\[ HCR_{HE} = \left( \frac{\Delta \dot{H}_{\text{max},c}}{\Delta \dot{H}_{\text{max},h}} \right) \]  

(3.11)

since the maximum temperature difference in \( \Delta \dot{H}_{\text{max},c} \) and \( \Delta \dot{H}_{\text{max},h} \) is the same (i.e., \( \Delta \dot{H}_{\text{max},k} = \dot{m}_k c_{p,k} \cdot (T_{h,i} - T_{c,i}) \), where \( k = c \) or \( h \)). Similarly, for an HME device, we define

\[ HCR = \left( \frac{\Delta \dot{H}_{\text{max},c}}{\Delta \dot{H}_{\text{max},h}} \right) \]  

(3.12)

The validity of this definition is proven using several examples in Sec. 3.2.

**Non-dimensional entropy generation**

For heat exchangers, it is shown in Sec. 3.2.1 that a non-dimensional entropy generation term \( \frac{S_{gen}}{(m c_p)_{\text{min}}} \) appears in the derivation. Similarly, we define a non-dimensional term for heat and mass exchangers.
Case I, $\Delta \dot{H}_{\text{max},w} < \Delta \dot{H}_{\text{max},a}$;

$$\frac{\dot{S}_{\text{gen}}}{(\bar{m} \cdot c_p)_{\text{min}}} = \frac{\dot{S}_{\text{gen}}}{(\bar{m}_w \cdot c_{p,w})}$$ (3.13)

Case II, $\Delta \dot{H}_{\text{max},w} > \Delta \dot{H}_{\text{max},a}$;

$$\frac{\dot{S}_{\text{gen}}}{(\bar{m} \cdot c_p)_{\text{min}}} = \frac{\dot{S}_{\text{gen}}}{(\bar{m}_w \cdot c_{p,w} \cdot \text{HCR})}$$ (3.14)

where, $\bar{m}_w$ is the average mass flow rate of water through the cooling tower and the term $(\bar{m}_w \cdot c_{p,w}) \cdot \text{HCR}$ can be thought of as an equivalent moist air heat capacity (since HCR is the heat capacity rate ratio as shown in later section). In this paper, $\bar{m}_w$ has been calculated as the average of the mass flow rates of water at the inlet and the outlet.

$$\bar{m}_w \approx \frac{\dot{m}_{w,i} + \dot{m}_{w,o}}{2}$$ (3.15)

The typical maximum difference in water mass flow rate from inlet to outlet in a humidifier or a cooling tower is 5% to 10%. Hence, the inlet water mass flow can also be used as a good approximation to the average water mass flow for the cooling tower cases presented in this paper.

**Limiting value of energy effectiveness**

The effectiveness of an HME defined by Eq. (3-1) varies from zero to a maximum value that might be less than one. The maximum value depends significantly on the heat capacity ratio (HCR) defined above and is constrained by the Second Law and by transport processes.

An example of the variation of the maximum effectiveness is shown in Fig. 3-2. The curves are plotted at various values of HCR versus exit relative humidity. Each curve consists of two linear segments: one, the horizontal segment at high relative humidities ($\geq 0.9$) and; two, the sloped straight line at lower relative humidities. The first (horizontal) segment consists of data points that are constrained by the
Second Law ($\dot{S}_{gen} = 0$); in this segment the outlet air temperature does not reach the water inlet temperature ($T_{a,o} < T_{w,i}$). The second segment consists of points that are constrained by the temperature cross, that is the outlet air temperature reaches the inlet water temperature ($T_{a,o} = T_{w,i}$) but entropy generation is non-zero and positive ($\dot{S}_{gen} > 0$). This second segment (the line with a positive slope) is linear because enthalpy varies almost linearly with relative humidity in the range of temperature changes considered here.

To incorporate the two constraints in the effectiveness definition, the maximum value of enthalpy change can be redefined as follows:

$$\Delta H_{max}^{mod} = \varepsilon_{max} \cdot \min \left( \Delta H_{max,w}, \Delta H_{max,a} \right)$$  \hspace{1cm} (3.16)
and Eq. (3.5) would be modified to:

\[ \varepsilon_{\text{mod}} = \frac{\Delta \dot{H}}{\Delta H_{\text{mod}}^{\text{max}}} \]  \hspace{1cm} (3.17)

Equation (3.17) ensures that the effectiveness varies from 0 to 1 (as for heat exchangers). However, this modification to the definition of effectiveness will make it very cumbersome to evaluate effectiveness a priori. Hence, the definition given in Eq. (3.5) is recommended and will be used in this thesis.

### 3.1.2 Equations and modeling details

This section discusses the conservation equations for indirect and direct contact HME devices. The following reasonable approximations are made in the conservation equations.

- The processes involved operate at steady-state conditions.
- There is no heat loss from the components to the surroundings.
- Kinetic and potential energy terms are neglected in the energy balance.

#### Direct contact heat and mass exchangers

Consider a counterflow cooling tower (Fig. 3-3) in which one fluid stream is pure water and the other stream is a mixture of air and water vapor. Since the dry air that enters the device in the humid air stream all leaves in the humid air stream, the mass flow rate of dry air is constant:

\[ m_{da} = m_{da,i} = m_{da,o} \]  \hspace{1cm} (3.18)

A mass balance on the water in the cooling tower gives the mass flow rate of the water leaving the humidifier in the water stream:

\[ m_{w,o} = m_{w,i} - m_{da} (\omega_{a,o} - \omega_{a,i}) \]  \hspace{1cm} (3.19)
Based on Eq. (3.5), the energy effectiveness, $\varepsilon$, may be written in terms of mass flow rates, temperatures, and humidity ratios. However, in order to determine the maximum possible change in enthalpy rate, it must be known whether the air stream or the water stream is the hot stream.

When the water enters hotter than the air, the ideal condition that the water stream can attain is that the temperature at the exit equals the inlet air wet-bulb temperature. This corresponds to the enthalpy driving force (which is nothing but the enthalpy potential difference between the two streams driving the heat and mass transfer) becoming zero at the water exit [76]. The ideal condition that the moist air stream can reach is saturation at the inlet water temperature. As explained earlier this is a limit imposed by the rate processes ($T_{a,o} \leq T_{w,i}$). When the air enters hotter than the water stream, the ideal conditions that can be attained by the air and the water is different to the case with hot water entering the HME. These again correspond to the driving enthalpy difference becoming zero for the respective streams.

Based on the above discussion, the effectiveness definition of a counterflow direct contact HME device with hot water entering is written as follows:
Case I, $\Delta \dot{H}_{\text{max},w} < \Delta \dot{H}_{\text{max},a}$:

$$\varepsilon = \frac{\dot{m}_{w,i}h_{w,i} - \dot{m}_{w,o}h_{w,o}}{\dot{m}_{w,i}h_{w,i} - \dot{m}_{w,o}h_{w,o}^{\text{ideal}}}$$  \hspace{1cm} (3.20)

Case II, $\Delta \dot{H}_{\text{max},w} > \Delta \dot{H}_{\text{max},a}$:

$$\varepsilon = \frac{\dot{m}_{da}(h_{a,o} - h_{a,i})}{\dot{m}_{da}(h_{a,o}^{\text{ideal}} - h_{a,i})}$$  \hspace{1cm} (3.21)

Note that the First Law for the cooling tower gives,

$$0 = \frac{\dot{m}_{da}(h_{a,o} - h_{a,i})}{\Delta \dot{H}_{a}} + \frac{\dot{m}_{w,i}h_{w,i} - \dot{m}_{w,o}h_{w,o}}{\Delta \dot{H}_{w}}$$  \hspace{1cm} (3.22)

where $\Delta \dot{H}_{w}$ is the change in total enthalpy rate for the feed water stream and $\Delta \dot{H}_{a}$ is the change in total enthalpy rate of the moist air stream. One can similarly write down the effectiveness definition when hot air enters the tower.

**Indirect contact heat and mass exchangers**

Now consider a counterflow dehumidifier (Fig. 3-4) in which one fluid stream is pure water and the other stream is a mixture of air and water vapor. The air-vapor mixture is transferring heat to the water stream. In this process, some of the water vapor in the mixture condenses out and forms a separate condensate stream. Since all the dry air in the air stream and the water in the other fluid stream that enters the dehumidifier also leaves the device, the mass flow rate of dry air and mass flow rate of the water is constant.

$$\dot{m}_{da} = \dot{m}_{da,i} = \dot{m}_{da,o}$$  \hspace{1cm} (3.23)

$$\dot{m}_{w,o} = \dot{m}_{w,i}$$  \hspace{1cm} (3.24)

The mass flow rate of the condensed water can be calculated using a simple mass
balance:
\[ \dot{m}_{pw} = \dot{m}_{da} (\omega_{a,o} - \omega_{a,i}) \]  
(3.25)

To calculate the maximum total enthalpy rate change possible, the inlet temperatures and mass flow rates must be known. As explained before, the ideal condition corresponds to the enthalpy driving force becoming zero at the water exit or the air exit. The ideal condition that the air can reach at the exit is saturation at the inlet temperature of water. The water can at best reach the dry bulb temperature of the air at inlet. Again, this corresponds to the enthalpy driving force reaching zero at the air inlet end.

Based on the above discussion, the effectiveness definition of a counterflow indirect contact HME device is as follows:

Case I, \( \Delta \dot{H}_{\text{max, w}} < \Delta \dot{H}_{\text{max, a}} \):

\[ \varepsilon = \frac{h_{w,i} - h_{w,o}}{h_{w,i} - h_{w,0}^{\text{ideal}}} \]  
(3.26)
Case II, $\Delta \dot{H}_{\text{max},w} > \Delta \dot{H}_{\text{max},a}$:

$$
\Delta = \frac{\dot{m}_{\text{da}} (h_{a,o} - h_{a,i}) + \dot{m}_{\text{pw}} h_{\text{pw}}}{\dot{m}_{\text{da}} (h_{a,o} - h_{a,i}^{\text{ideal}}) + \dot{m}_{\text{pw}} h_{\text{pw}}} 
$$

(3.27)

Note that the First Law for the dehumidifier can be expressed as,

$$
0 = \dot{m}_{\text{da}} (h_{a,i} - h_{a,o}) - \dot{m}_{\text{pw}} h_{\text{pw}} + \dot{m}_{\text{w}} (h_{w,i} - h_{w,o})
$$

(3.28)

where $\Delta \dot{H}_w$ is the change in total enthalpy rate for the feed water stream and $\Delta \dot{H}_a$ is the change in total enthalpy rate of the moist air stream.

### 3.1.3 Solution technique

The solution of the governing equations was carried out using Engineering Equation Solver (EES) [77] which uses accurate equations of state to model the properties of moist air and water. EES evaluates water properties using the IAPWS (International Association for Properties of Water and Steam) 1995 Formulation [78]. Dry air properties are evaluated using the ideal gas formulations presented by Lemmon et al. [79]. Moist air properties are evaluated assuming an ideal mixture of air and steam using the formulations presented by Hyland and Wexler [80]. Moist air properties from EES are in close agreement with the data presented in ASHRAE Fundamentals [81] and pure water properties are equivalent to those found in NIST's property package, REFPROP [82].

EES is a numerical solver, and it uses an iterative procedure to solve the equations. The convergence of the numerical solution is checked by using the following two variables: (1) ‘Relative equation residual’ — the difference between left-hand and right-hand sides of an equation divided by the magnitude of the left-hand side of the equation; and (2) ‘Change in variables’ — the change in the value of the variables within an iteration. The calculations converge if the relative equation residuals is lesser than $10^{-6}$ or if change in variables is less than $10^{-9}$. These are standard values used to check convergence in EES. There are several publications which have
previously used them for thermodynamic analysis [83, 84].

3.2 Control volume based second law based design of HME devices

Before we study the applicability of the energy effectiveness definition (section 3.3), it is important to understand the concept of entropy generation and thermodynamic balancing of HME devices.

A few researchers [85–87] have previously attempted to optimize the second law design of heat and mass exchange devices by studying the sources of irreversibilities at the local level. Carrington and Sun [86] presented the following expression for the volumetric entropy generation rate as a sum of the three components that arise in combined heat and mass transfer processes. These include a mass transfer component, a heat transfer component and a coupled heat and mass transfer component, neglecting the irreversibilities due to fluid flow.

\[
\dot{S}_{gen}' = \frac{k}{T^2} \cdot (\nabla T)^2 + \frac{2\rho^2 \bar{D}'}{M_A M_B C} (\nabla T) \cdot (\nabla x_A) + \frac{\rho^2 \bar{D}_{AB}}{M_A M_B x_A x_B} \cdot (\nabla x_A)^2 \tag{3.29}
\]

The first term on the right hand side of Eqn. 3.29 is the heat transfer component of the total irreversibility. In heat exchangers (without phase change or mass transfer, for example) this is the only term that causes the irreversibilities, other than the fluid flow irreversibilities. Hence, by balancing the stream-to-stream temperature difference in the heat exchangers we can minimize entropy production for a fixed effectiveness. In the following section it is shown how the temperature balance minimizes entropy production in a heat exchanger. However, the same is not true for a combined heat and mass exchange device, since in some situations the second and third term (related to mass diffusion irreversibilities) in Eqn. 3.29 can play a

\footnote{Because the energy effectiveness definition is to be tested at various degrees of thermodynamically balancing of HME devices.}
major role. Which component dominates the irreversibility depends on how steep the
temperature and mass concentration gradients \((x)\) are [88].

To exactly evaluate the contribution of each of the terms in Eqn. 3.29 on total irreversibility one has to perform a volume integral that requires knowledge of the local temperature and mass concentration profiles throughout the device. Hence, it is useful to apply a simple control volume approach to identify the key variables governing the process. This paper develops a control volume procedure that facilitates minimization of the entropy generation in simultaneous heat and mass exchangers. The method uses an enthalpy-based effectiveness and modified heat capacity rate ratio, which is suitable for on-design analysis of adiabatic two-stream components within a cycle.

### 3.2.1 Expressions for entropy generation

#### Entropy generation in heat exchangers

Several researchers have previously studied entropy production in heat exchangers [89–93]. In this section, we draw upon this literature to develop an understanding of how entropy production can be minimized for heat exchangers. Conservation equations for a counterflow heat exchanger shown in Fig. 3-5 are as follows. It is assumed that there is no phase change in either the hot or the cold fluid streams and that the heat exchanger has no heat loss to the environment.

![Figure 3-5: Control volume for a heat exchanger.](image)

Energy conservation is expressed by
\[ \dot{m}_c \cdot (h_{2,c} - h_{1,c}) = \dot{m}_h \cdot (h_{2,h} - h_{1,h}) \]  
(3.30)

and if the specific heat capacities are constant

\[ \dot{m}_c c_{p,c} \cdot (T_{2,c} - T_{1,c}) = \dot{m}_h c_{p,h} \cdot (T_{2,h} - T_{1,h}) \]  
(3.31)

Here, we have assumed that both the streams are incompressible and that the pressure change is negligible between the inlet and outlet.

The Second Law gives,

\[ \dot{S}_{gen} = \dot{m}_c \cdot (s_{2,c} - s_{1,c}) + \dot{m}_h \cdot (s_{1,h} - s_{2,h}) \]
\[ = \dot{m}_c c_{p,c} \cdot \ln \left( \frac{T_{2,c}}{T_{1,c}} \right) + \dot{m}_h c_{p,h} \cdot \ln \left( \frac{T_{1,h}}{T_{2,h}} \right) \geq 0 \]  
(3.33)

The heat exchanger effectiveness is defined in the usual fashion as actual to maximum heat transfer:

\[ \varepsilon \equiv \frac{\dot{Q}}{\dot{Q}_{\text{max}}} \]  
(3.34)

\[ \dot{Q}_{\text{max}} = (\dot{m} \cdot c_p)_{\text{min}} \cdot (T_{2,h} - T_{1,c}) \]  
(3.35)

We define the heat capacity rate ratio (HCR) as

\[ \text{HCR} = \frac{\dot{m}_c \cdot c_{p,c}}{\dot{m}_h \cdot c_{p,h}} = \frac{\dot{C}_c}{\dot{C}_h} \]  
(3.36)

Using these equations we can write the entropy generation in two possible ways,

Case I, \( \dot{C}_c < \dot{C}_h \):

\[ \frac{\dot{S}_{gen}}{\dot{C}_c} = \frac{1}{\text{HCR}} \cdot \ln \left\{ 1 - \varepsilon \cdot \text{HCR} \left( 1 - \frac{T_{1,c}}{T_{2,h}} \right) \right\} + \ln \left\{ 1 + \varepsilon \left( \frac{T_{2,h}}{T_{1,c}} - 1 \right) \right\} \]  
(3.37)

Case II, \( \dot{C}_h < \dot{C}_c \):

\[ \frac{\dot{S}_{gen}}{\dot{C}_h} = \text{HCR} \cdot \ln \left\{ 1 + \frac{1}{\text{HCR}} \varepsilon \left( \frac{T_{2,h}}{T_{1,c}} - 1 \right) \right\} + \ln \left\{ 1 - \varepsilon \left( 1 - \frac{T_{1,c}}{T_{2,h}} \right) \right\} \]  
(3.38)
Hence, it can be seen that entropy production depends on these parameters:

\[
\frac{\dot{S}_{\text{gen}}}{C_{\text{min}}} = f\left\{ \left( \frac{T_{2,h}}{T_{1,c}} - 1 \right), \text{HCR}, \varepsilon \right\}
\]  

(3.39)

In Fig. 3-6, we have shown that for various values of effectiveness and a fixed temperature ratio, the non-dimensional entropy generation is minimized at HCR=1. A simple calculation shows that this is true for all values of temperature ratio and effectiveness. At this condition, the heat exchanger is said to be ‘balanced’. It is important to understand that the stream-to-stream temperature difference is constant throughout the length of the heat exchanger in this condition (see Eqn. 3.31 and 3.36). In other words, the driving force (temperature difference) is balanced. The driving force for energy transfer in a combined heat and mass exchanger is a combination of both the temperature difference and the concentration difference. In the following sections, we try to understand how to balance these driving forces for HME devices.

Figure 3-6: Non-dimensional entropy generation versus heat capacity rate ratio for counterflow heat exchangers; \( \frac{T_{2,h}}{T_{1,c}} = 0.5 \).
Entropy generation in heat and mass exchangers

Here we develop an expression for entropy generation in a heat and mass exchanger to identify the variables that affect $\dot{S}_{\text{gen}}$ in these devices. The example of a counterflow cooling tower (Fig. 3-3) is taken. An energy balance gives

$$\dot{m}_{w,i} \cdot h_{w,i} - \dot{m}_{w,o} \cdot h_{w,o} = \dot{m}_{da} \cdot (h_{a,o} - h_{a,i})$$

(3.40)

The enthalpy of moist air (air-water vapor mixture), can be alternatively defined as,

$$h_a - h_{\text{ref}} = c_{p,da} \cdot (T - T_{\text{ref}}) + \omega \cdot [c_{p,v} \cdot (T - T_{\text{ref}}) + h_{fg}]$$

(3.41)

In the above equation, it is assumed for purposes of discussion that the water vapor and dry air are ideal gases and also that their specific heat capacities are constant (in the calculations of Section 3.2.2, exact properties are used). Taking these ideas into consideration, we define the component effectiveness carefully before we look at examples of balancing the heat and mass exchanger. Using Eqns. 3.41, we write the following expression for $T_{w,o}$:

$$T_{w,o} = \left\{ \frac{\dot{m}_{w,i} \cdot c_{p,w}}{\dot{m}_{w,i} \cdot c_{p,w} - \dot{m}_{da} \cdot (\omega_o - \omega_i) \cdot c_{p,w}} \right\} T_{w,i}
+ \left\{ \frac{\dot{m}_{da} \cdot (c_{p,da} + c_{p,v} \cdot \omega_i)}{\dot{m}_{w,i} \cdot c_{p,w} - \dot{m}_{da} \cdot (\omega_o - \omega_i) \cdot c_{p,w}} \right\} T_{a,i}
- \left\{ \frac{\dot{m}_{da} \cdot (c_{p,da} + c_{p,v} \cdot \omega_o)}{\dot{m}_{w,i} \cdot c_{p,w} - \dot{m}_{da} \cdot (\omega_o - \omega_i) \cdot c_{p,w}} \right\} T_{a,o}
+ \left\{ \frac{\dot{m}_{da} \cdot h_{fg,o}}{\dot{m}_{w,i} \cdot c_{p,w} - \dot{m}_{da} \cdot (\omega_o - \omega_i) \cdot c_{p,w}} \right\} \omega_o
+ \left\{ \frac{\dot{m}_{da} \cdot h_{fg,i}}{\dot{m}_{w,i} \cdot c_{p,w} - \dot{m}_{da} \cdot (\omega_o - \omega_i) \cdot c_{p,w}} \right\} \omega_i$$

(3.42)

where, $h_{fg,o}$ and $h_{fg,i}$ are evaluated at the inlet and outlet air temperatures. From the definition of effectiveness, we have two cases. Case I, $\Delta \dot{H}_{\text{max,w}} < \Delta \dot{H}_{\text{max,a}}$:

$$T_{w,o} = \left\{ \frac{\dot{m}_{w,i}}{\dot{m}_{w,i} - (\omega_o - \omega_i) \cdot \dot{m}_{da}} \right\} T_{w,i} \cdot (1 - \varepsilon) + \varepsilon \cdot T_{a,i}$$

(3.43)
Case II, $\Delta H_{\text{max},w} > \Delta H_{\text{max},a}$:

$$T_{a,o} = \begin{cases} 
    c_{p,da} + c_{p,v} \cdot \omega_i & (1 - \varepsilon) T_{a,i} + \varepsilon \cdot T_{w,i} \\
    c_{p,da} + c_{p,v} \cdot \omega_o & -(1 - \varepsilon) \cdot \left\{ \frac{(\omega_o \cdot h_{fg,o} - \omega_i \cdot h_{fg,i})}{c_{p,da} + c_{p,v} \cdot \omega_o} \right\} 
\end{cases}$$

(3.44)

Applying the Second Law to the control volume, we can derive the following expression for entropy production in a cooling tower (see Appendix B):

$$\frac{\dot{S}_{\text{gen}}}{m_{w,i} \cdot c_{p,w}} = \ln \left( \frac{T_{w,o}}{T_{w,i}} \right) + \frac{\dot{m}_{da} \cdot c_{p,v}}{m_{w,i} \cdot c_{p,v}} \cdot \ln \left( \frac{T_{a,o}}{T_{a,i}} \right)$$

$$+ \frac{\dot{m}_{da} \cdot c_{p,v}}{m_{w,i} \cdot c_{p,v}} \cdot \frac{T_{w,i} \cdot c_{p,v}}{T_{w,o}} \cdot \ln \left( \frac{P_{\text{sat},w,o}}{P_{\text{total}}} \right)$$

$$+ \frac{\dot{m}_{da} \cdot R_w}{m_{w,i} \cdot c_{p,w}} \cdot \ln \left( \frac{1 + 1.608 \cdot \omega_o}{1 + 1.608 \cdot \omega_i} \right)$$

$$- \frac{\dot{m}_{da} \cdot R_{w,i}}{m_{w,i} \cdot c_{p,w}} \cdot \omega_o \cdot \ln \left( 1 + \frac{1}{1.608 \cdot \omega_o} \right)$$

$$+ \frac{\dot{m}_{da} \cdot R_w}{m_{w,i} \cdot c_{p,w}} \cdot \omega_i \cdot \ln \left( 1 + \frac{1}{1.608 \cdot \omega_i} \right)$$

(3.45)

where $P_{\text{sat}}$ is the saturation vapor pressure of water. Using Eqns. (3.42 - 3.44) we can see that entropy has the following functional form;

$$\frac{\dot{S}_{\text{gen}}}{m_{w,i} \cdot c_{p,w}} = f_1 \left\{ \frac{T_{w,i}}{T_{a,i}}, \phi_o, \phi_i, \varepsilon, R_1, R_2, R_3, R_4, \frac{\dot{m}_{da}}{m_{w,i}} \right\}$$

(3.46)

where: $R_1 = \frac{\dot{m}_{da} \cdot c_{p,da}}{m_{w,i} \cdot c_{p,w}}$; $R_2 = \frac{\dot{m}_{da} \cdot R_{a}}{m_{w,i} \cdot c_{p,w}}$; $R_3 = \frac{\dot{m}_{da} \cdot R_{w,i}}{m_{w,i} \cdot c_{p,w}}$; $R_4 = \frac{\dot{m}_{da} \cdot c_{p,v}}{m_{w,i} \cdot c_{p,w}}$; $P^* = \frac{P_{\text{total}}}{P_{\text{sat},w,o}}$.

Also, we can see that the parameters in Eqn. 25 have the following functional form:

$$R_1, R_2, R_3, R_4 = f_2 \left\{ \frac{\dot{m}_{da}}{m_{w,i}}, \frac{c_{p,da}}{c_{p,w}}, \frac{R_{a}}{c_{p,w}}, \frac{R_{w,i}}{c_{p,w}}, \frac{R_{w}}{c_{p,w}} \right\}$$

(3.47)
and

$$P^* = J_3\left\{ P_{\text{total}}, \frac{\dot{m}_{\text{da}}}{\dot{m}_{\text{w},i}}, T_{a,i}, T_{w,i}, \varepsilon, \phi_o, \phi_i \right\}$$  \hspace{1cm} (3.48)$$

Therefore, we can write Eqn. (25) as,

$$\frac{\dot{S}_{\text{gen}}}{\dot{m}_{\text{w},i} \cdot c_{p,w}} = f_4\left\{ T_{w,i}, T_{a,i}, \phi_o, \phi_i, \varepsilon, \frac{\dot{m}_{\text{da}}}{\dot{m}_{\text{w},i}}, \frac{c_{p,da}}{c_{p,w}}, \frac{R_{da}}{R_w}, \frac{R_w}{c_{p,w}}, \frac{c_{p,w}}{c_{p,w}}, P_{\text{total}} \right\}$$  \hspace{1cm} (3.49)$$

In Section 3.3, we show how the non-dimensional entropy production can be minimized using this understanding.

### 3.2.2 Condition for minimum entropy generation

#### Effect of mass flow rate ratio

In previous section, we showed that the non-dimensional entropy generation of a cooling tower is a function of the inlet temperatures, the component effectiveness, the exit and inlet relative humidities, the ratio of certain specific heat capacities ($\left(\frac{c_{p,da}}{c_{p,w}}, \frac{R_{da}}{R_w}, \frac{R_w}{c_{p,w}}, \frac{c_{p,w}}{c_{p,w}}\right)$, the absolute pressure and the mass flow rate ratio. Figure 3-7 illustrates the importance of this understanding.

The mass flow rate ratio is plotted against the non-dimensional entropy generation term ($\frac{\dot{S}_{\text{gen}}}{(\dot{m}_{\text{w},i} \cdot c_{p,w})_{\text{min}}}$) for various values of inlet water temperature ($T_{w,i}$) of 55°C to 70°C, an inlet air temperature ($T_{a,i}$) of 34°C, an exit relative humidity ($\phi_o$) of 70%, an inlet relative humidity ($\phi_i$) of 90% and an effectiveness ($\varepsilon_{da}$) of 80%. It can be observed that the non-dimensional entropy generation is minimized at a particular value of mass flow rate ratio. This is found to be true for all values of input variables (inlet temperatures, inlet and exit relative humidities and component effectiveness). Hence, we can conclude that at fixed inlet conditions the optimum point for non-dimensional entropy generation is dependent only on the mass flow rate ratio.

In certain cases (e.g., in Fig. 3-7 for $T_{w,i} = 70^\circ C$) the minimum entropy generation point cannot be attained within the constraints ($\dot{S}_{\text{gen}} \geq 0; T_{w,i} \geq T_{a,o}$). Also, there are points of zero entropy generation at effectiveness less than 1. These correspond to points of maximum effectiveness.
Optimization using modified heat capacity ratio

The optimum point for non-dimensional entropy generation occurs at a particular value of mass flow rate ratio, but this value varies with change in input variables (inlet temperatures, inlet and exit relative humidities and component effectiveness). In this section, we normalize the optimum point by using the modified heat capacity rate ratio defined in Section 3.1.3. We also look at the effect that the different input variables have on the non-dimensional entropy generation.

Effect of water inlet temperature ($T_{w,i}$) Figure 3-8 shows the non-dimensional entropy production plotted against modified heat capacity rate ratio for various values of water inlet temperature at fixed values of $T_{a,i}, \phi_o, \phi_i$ and $\varepsilon$. For this figure, the mass flow rate ratio is varied from 1.2 to 29 to generate cases with HCR from 0.2 to 2.5. Two important observations can be made from this graph: (1) Irrespective of the value of $T_{w,i}$, non-dimensional entropy generation is minimized at HCR=1; (2) There
is not a monotonic correlation between the non-dimensional entropy generation and the water inlet temperature. This trend is consistent for various (fixed) values of $T_{a,i}, \phi_o, \phi_i$ and $\varepsilon$.

**Effect of air inlet temperature ($T_{a,i}$)** Figure 3-9 illustrates the effect that the air inlet temperature has on non-dimensional entropy production. As was the case with the previous example, the entropy production is minimized at HCR=1 at all values of $T_{a,i}$. This trend is for fixed values of $T_{w,i}, \phi_o, \phi_i$ and $\varepsilon$.

**Effect of exit relative humidity of the air ($\phi_o$)** We had noted in Section 3.1 that to completely balance a heat and mass exchange device we have to balance the driving force of energy transfer, which is a combination of the temperature and concentration difference. In cooling towers, this concentration difference is represented by the difference in relative humidity of the air stream (between inlet and outlet, for example). Figure 3-10 shows that cooling towers are balanced at HCR=1 for different values of exit relative humidity. It can also be observed that non-dimensional entropy production varies only slightly with $\phi_o$ (the curves are almost on top of each other). This is because the range of exit relative humidities possible for the presented case is small.

**Effect of inlet relative humidity of the air ($\phi_i$)** Figure 3-11 illustrates the effect of inlet air relative humidity on the non-dimensional entropy production. As we had observed previously, the entropy production minimizes at HCR=1 irrespective of the value of the independent variables. Also the entropy production is inversely correlated with $\phi_i$.

**Effect of pressure drop** Figure 3-12 illustrates the effect of pressure drop on the non-dimensional entropy production. For this example, the air side pressure drops are taken to be varying from 0 to 10 kPa. The entropy production minimizes at HCR=1 irrespective of how large or small the pressure drop is. Also entropy production increases with increase in pressure drop.
Figure 3-8: Effect of water inlet temperature on entropy generation; \( T_{w,i} = 34^\circ C \); \( \varepsilon = 0.8; \phi_o = 0.9; \phi_t = 0.6; \dot{m}_{w,in}/\dot{m}_d = 1.2 - 29; P = 100 \text{ kPa} \).

Figure 3-9: Effect of air inlet temperature on entropy generation; \( T_{a,i} = 55^\circ C \); \( \varepsilon = 0.8; \phi_o = 0.6; \phi_t = 0.8; \dot{m}_{w,in}/\dot{m}_d = 1.2 - 29; P = 100 \text{ kPa} \).
Figure 3-10: Effect of outlet air relative humidity on entropy generation in a cooling tower; $T_{w,i} = 50^\circ C; T_{a,i} = 34^\circ C; \varepsilon = 0.7; \phi_i = 0.5; \dot{m}_{w,in}/\dot{m}_{da} = 1.2 - 29; P = 100 kPa.$

Figure 3-11: Effect of inlet air relative humidity on entropy generation in a cooling tower; $T_{w,i} = 50^\circ C; T_{a,i} = 34^\circ C; \varepsilon = 0.7; \phi_o = 0.9; \dot{m}_{w,in}/\dot{m}_{da} = 1.2 - 29; P = 100 kPa.$
Figure 3-12: Effect of air side pressure drop on entropy generation in a cooling tower; $T_{w,i} = 50^\circ C; T_{a,i} = 34^\circ C; \varepsilon = 0.7; \phi_o = 0.9; \phi_i = 0.6; \dot{m}_{w,in}/\dot{m}_{da} = 1.2 - 29; P = 100$ kPa.

From the various examples in this section, it is clear that a combined heat and mass exchanger is ‘balanced’ and the entropy production is minimum when the modified heat capacity rate ratio is 1. Also, the functional dependence of the non-dimensional entropy production is

$$\frac{\dot{S}_{gen}}{(\dot{m} \cdot c_p)_{\min}} = f_5 \left\{ T_{w,i}, T_{a,i}, \phi_o, \phi_i, \varepsilon, HCR, \frac{c_{p,da}}{c_{p,w}}, \frac{R_{da}}{R_w}, \frac{R_w}{c_{p,w}}, \frac{c_{p,v}}{c_{p,w}}, P_{total} \right\}$$  \hspace{1cm} (3.50)

where

$$HCR = \left( \frac{\Delta H_{max,c}}{\Delta H_{max,h}} \right)$$  \hspace{1cm} (3.51)

3.2.3 Section summary

In this section, the following significant conclusions are arrived at from this study:

1. The entropy production can be usefully non-dimensionalized by dividing it by the heat capacity rate of the minimum heat capacity stream in the heat and
2. A modified heat capacity rate ratio (HCR) for simultaneous heat and mass exchangers was defined as the ratio of the maximum possible change in total enthalpy rate of the cold stream to the maximum possible change in total enthalpy rate of the hot stream \( \text{HCR} = \frac{\Delta H_{\text{max,c}}}{\Delta H_{\text{max,h}}} \).

3. The non-dimensional entropy production for a cooling tower is dependent on the following variables:

\[
\frac{S_{\text{gen}}}{(m \cdot c_p)_{\text{min}}} = f \{ T_{w,i}, T_{a,i}, \phi_o, \phi_l, \varepsilon, \text{HCR}, c_{p,\text{da}}, R_{\text{da}}, R_{\text{w}}, c_{p,\text{w}}, c_{p,v}, P_{\text{total}} \}
\]

It is noted that this includes performance parameters \((\varepsilon, \phi_o)\), inlet variables \((T_{w,i}, T_{a,i}, \frac{m_{\text{w,i}}}{m_{\text{da}}})\), pressure \((P_{\text{total}})\) and fluid properties \((c_{p,\text{da}}, c_{p,\text{w}}, c_{p,v}, R_{\text{da}}, R_{\text{w}})\).

Operationally, HCR can be varied by only changing the mass flow rate ratio \((\frac{m_{\text{w,i}}}{m_{\text{da}}})\). The component effectiveness contains the effects of the exchanger dimensions and material properties and is varied in an on-design sense.

4. The non-dimensional entropy production is minimized and the heat and mass exchanger is 'balanced' at HCR=1. The experimental proof of the same is described in Chapter 6.

### 3.3 Applicability of energy effectiveness

In this section, the differences and similarities between the various definitions of effectiveness and their relationship to the energy effectiveness are examined. How the heat capacity rate ratio affects the aforementioned relationships is also discussed. All graphs in this section plot the various effectivenesses versus the energy based effectiveness. Additionally, the non-dimensional entropy generation rate is also plotted versus the energy based effectiveness.

#### 3.3.1 Heat exchangers

In this section, it is shown that the energy effectiveness of an HME in the limiting case of no mass transfer in a counterflow humidifier is equivalent to that of a counterflow
heat exchanger. When mass is not transferred, the humidity ratio of the moist air stream remains unchanged from inlet to outlet. Therefore,

$$\omega_i = \omega_o$$  \hspace{1cm} (3.52)

Since the humidity ratio of the air stream remains constant, the effectivenesses reduce to the following Case I, water is the minimum heat capacity stream (HCR > 1)

$$\varepsilon = \frac{h_{w,i} - h_{w,o}}{h_{w,i} - h_{w,\text{ideal}}}$$

$$\varepsilon = \frac{c_{p,w} (T_{w,i} - T_{w,o})}{c_{p,w} (T_{w,i} - T_{w,\text{ideal}})} = \frac{c_{p,w} (T_{w,i} - T_{w,o})}{c_{p,w} (T_{w,i} - T_{w,i})}$$  \hspace{1cm} (3.53)

Case II, moist air is the minimum heat capacity stream (HCR < 1)

$$\varepsilon = \frac{h_{a,i} - h_{a,o}}{h_{a,i} - h_{a,\text{ideal}}} = \frac{c_{p,a} (T_{a,i} - T_{a,o})}{c_{p,a} (T_{a,i} - T_{a,i})}$$  \hspace{1cm} (3.54)

Finally, the maximum value of the above two equations gives the energy effectiveness expression

$$\varepsilon = \frac{\dot{m}_w c_{p,w} (T_{w,i} - T_{w,o})}{C_{\text{min}} (T_{w,i} - T_{a,i})}$$

$$\varepsilon = \frac{\dot{m}_d a c_{p,a} (T_{a,i} - T_{a,o})}{C_{\text{min}} (T_{a,i} - T_{w,i})}$$  \hspace{1cm} (3.55)

which is the usual definition for the effectiveness of a two stream heat exchanger [94].

### 3.3.2 Direct contact heat and mass exchangers

**Counterflow humidifiers with saturated air at the inlet**

The comparison of various definition of effectiveness for a counterflow humidifier is shown in Fig. 3-13. In this example, the device operates at atmospheric pressure with the inlet water temperature at 70°C, and the moist air entering saturated at 30°C and exiting saturated. In this case, the device is operating at a heat capacity rate...
ratio of less than one which means that moist air is the minimum heat capacity rate stream. For the given case the maximum values of $\varepsilon$, $\varepsilon_h$ and $\varepsilon_w$ are about 0.9 and that of $\varepsilon_T$ is about 0.7. The entropy generation rate becomes negative when these values are exceeded. Moreover, the values of $\varepsilon$, $\varepsilon_h$ and $\varepsilon_w$ correlate very well with each other, but $\varepsilon_T$ differs. This is because $\varepsilon_T$ is a water-temperature-based definition, and, in the current case the minimum heat capacity stream is moist air.

![Image of graphs showing effectiveness and normalized entropy generation rate]

Figure 3-13: Comparison of different effectiveness definitions in a counterflow humidifier with saturated air at the inlet and when moist air is the minimum heat capacity stream (HCR < 1).

Figure 3-14 shows the comparison for a situation in which HCR > 1, which means that the minimum heat capacity stream is the water. In this case, the values of $\varepsilon$ and $\varepsilon_T$ correlate very well with each other but $\varepsilon_h$ and $\varepsilon_w$ differ significantly from $\varepsilon$. This indicates that in situations where the HCR > 1, the temperature based effectiveness is a good approximation for the energy effectiveness, provided the inlet air is saturated. We will discuss the correlation for unsaturated air entering the inlet in Sec. 3.3.2.

Figure 3-15 illustrates the comparison for a situation in which HCR = 1. This condition is called the thermally balanced condition [21], and remanent irreversibility
Figure 3-14: Comparison of different effectiveness definitions in a counterflow humidifier when water is the minimum heat capacity stream (HCR > 1).

is a minimum in this condition. Since the temperature profiles and the humidity levels are balanced, all the various definitions of effectiveness are very similar in this case. This further demonstrates the concept of thermal balancing. In heat exchangers, this concept is well known and corresponds to the balancing of the temperature profiles [21, 74].

Another interesting observation that can be made from the figures in this section is that the maximum value of effectiveness is lower at the balanced condition and increases as HCR moves further away from a balanced condition (see Table 3.2). This is because irreversibility is lower at HCR = 1 compared to unland the entropy production becomes negative (See Figs. 3-13, 3-14 and 3-15) at a lower value of effectiveness. Hence, a lower value of maximum effectiveness exists for the balanced condition. At values of HCR sufficiently away from one (like the case were HCR = 3.75 in the table) the energy effectiveness has a maximum value of one.
Figure 3-15: Comparison of different effectiveness definitions in a counterflow humidifier at thermally balanced condition (HCR = 1).

Counterflow humidifier with unsaturated air at inlet

The case where the inlet air in unsaturated is discussed below. Table 3.3 illustrates the effect of this condition on the maximum values of effectivenesses. The comparisons are very similar to the saturated air condition discussed above.

Table 3.2: Examples of maximum effectiveness for a counterflow cooling tower with following boundary condition: $T_{w,i} = 70\,^\circ\text{C}$; $p_{w,i} = 1\,\text{atm}$, $T_{a,i} = 30\,^\circ\text{C}$; $p_{a,i} = 1\,\text{atm}$, $\phi_i = 1.0$.

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<tr>
<th>Thermal balance</th>
<th>Maximum value of effectiveness</th>
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</thead>
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<tr>
<td></td>
<td>$\varepsilon$</td>
</tr>
<tr>
<td>unbalanced, HCR = 0.85</td>
<td>0.8753</td>
</tr>
<tr>
<td>unbalanced, HCR = 3.75</td>
<td>1</td>
</tr>
<tr>
<td>balanced, HCR = 1</td>
<td>0.7937</td>
</tr>
</tbody>
</table>

97
Table 3.3: Examples of maximum effectiveness for a counterflow cooling tower with following boundary condition: $T_{w,i} = 70^\circ C$, $p_{w,i} = 1$ atm, $T_{a,i} = 30^\circ C$, $p_{a,i} = 1$ atm, $\phi_i = 0.5$.

<table>
<thead>
<tr>
<th>Thermal balance</th>
<th>Maximum value of effectiveness</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>$\varepsilon$</td>
</tr>
<tr>
<td>unbalanced, HCR = 0.85</td>
<td>0.7803</td>
</tr>
<tr>
<td>unbalanced, HCR = 4</td>
<td>1</td>
</tr>
<tr>
<td>balanced, HCR = 1</td>
<td>0.776</td>
</tr>
</tbody>
</table>

Counterflow humidifier with air entering hotter than water

The implications of having hotter air entering a direct contact HME on the ideal conditions that be achieved by the fluid streams were discussed in the previous section. Figure 3-16 illustrates the comparison of the effectivenesses for this boundary condition in a thermally balanced situation. It is observed that the values of $\varepsilon$, $\varepsilon_h$ and $\varepsilon_T$ correlate reasonably well with each other but $\varepsilon_w$ differs. This is because $\varepsilon_w$ is a humidity-based definition and the humidity change alone does not capture the change in total energy very well in hot air cases.

![Figure 3-16: Comparison of different effectiveness definitions at thermally balanced condition (HCR = 1) for counterflow humidifier with air entering hotter than water.](image-url)
3.3.3 Indirect contact heat and mass exchangers

Indirect contact or surface type HME are different from direct contact type HME in that the fluid streams are not mixed. Hence, it is important to investigate the comparison between the different effectiveness definitions. Also, it is interesting to investigate the influence of thermal balancing in these devices.

Figure 3-17 illustrates that at the thermally balanced condition, the dehumidifier can be defined by any of the four values of effectivenesses (which is similar to the direct contact counterpart). Also, when HCR $< 1$ and HCR $> 1$, all trends are similar to the direct contact type.

![Figure 3-17](image)

Figure 3-17: Comparison of different effectiveness definitions at thermally balanced condition (HCR = 1) for counterflow dehumidifier.

It was previously observed that for the direct contact HME the maximum value of effectiveness is lower at the balanced condition and increases as HCR moves further away from a balanced condition (Table 3.2). Table 3.4 shows that for counterflow indirect contact HME the maximum value of effectiveness is close to one for cases when HCR = 1 and HCR > 1. This is a significant observation and shows that the
Table 3.4: Examples of maximum effectiveness for a counterflow air dehumidifier with following boundary condition: $T_{m,i} = 30\, ^{\circ}\text{C}$; $p_{m,i} = 1\, \text{atm}$, $T_{a,i} = 70\, ^{\circ}\text{C}$; $p_{a,i} = 1\, \text{atm}$, $\phi_i = 1.0$.

<table>
<thead>
<tr>
<th>Thermal balance</th>
<th>Maximum effectiveness</th>
</tr>
</thead>
<tbody>
<tr>
<td>balanced, HCR = 1</td>
<td>$\varepsilon$</td>
</tr>
<tr>
<td>unbalanced, HCR = 0.25</td>
<td>1</td>
</tr>
<tr>
<td>unbalanced, HCR = 2</td>
<td>1</td>
</tr>
</tbody>
</table>

The performance of these exchangers reaches a true optimum at HCR = 1 and can be effectively used in engineering applications to optimize the design as is explained in Chapter 5.

### 3.4 Chapter conclusions

In this paper, the following significant conclusions have been reached:

1. A simple definition for energy effectiveness, which can be applied to all types of HMEs, has been developed. It is based on the total energy change of each fluid stream participating in the transfer processes.

2. A reliable definition for the modified heat capacity rate ratio, without any simplifying assumptions on the fluid properties, has been developed. This definition is based on the enthalpy flowrates for either stream, to account for the air stream humidity and interstream mass transfer.

3. Temperature, humidity and enthalpy based effectivenesses are each applicable in some cases, depending on the value of HCR and $\phi_i$. The range of applicability is noted in Table 3.5.

4. Using the comparison of the different effectivenesses at various values of HCR and considering the non-dimensional entropy generation, the concept of thermal balancing of heat and mass exchange devices was demonstrated.

5. There is a maximum value of effectiveness for certain configurations ($0 < \varepsilon < \varepsilon_{\max}$) which is lesser than 1.
6. The entropy production can be usefully non-dimensionalized by dividing it by the heat capacity rate of the minimum heat capacity stream in the heat and mass exchanger \( \frac{\dot{S}_{\text{gen}}}{(m\cdot c_p)_{\text{min}}} \).

7. A modified heat capacity rate ratio (HCR) for simultaneous heat and mass exchangers was defined as the ratio of the maximum possible change in total enthalpy rate of the cold stream to the maximum possible change in total enthalpy rate of the hot stream \( \text{HCR} = \frac{\Delta H_{\text{max,c}}}{\Delta H_{\text{max,h}}} \).

8. The non-dimensional entropy production for a cooling tower (or humidifier) is dependent on the following variables:

\[
\frac{\dot{S}_{\text{gen}}}{(m\cdot c_p)_{\text{min}}} = f \{ T_{w,i}, T_{a,i}, \phi_o, \phi_i, e, \text{HCR}, \frac{c_{p,a}}{c_{p,w}}, \frac{R_{da}}{R_{uw}}, \frac{R_{uw}}{c_{p,w}}, P_{\text{total}} \}
\]

It is noted that this includes performance parameters \((e, \phi_o)\), inlet variables \((T_{w,i}, T_{a,i}, \frac{m_{w,i}}{m_{da}})\), pressure \((P_{\text{total}})\) and fluid properties \((c_{p,a}, c_{p,w}, c_{p,v}, R_{da}, R_{uw})\). Operationally, HCR can be varied by only changing the mass flow rate ratio \((\frac{m_{w,i}}{m_{da}})\). The component effectiveness contains the effects of the exchanger dimensions and material properties and is varied in an on-design sense.

9. The non-dimensional entropy production is minimized and a control volume of the heat and mass exchanger is 'balanced' at HCR=1.
Table 3.5: Applicable range of various definitions of effectiveness of simultaneous heat and mass exchange devices.

<table>
<thead>
<tr>
<th>Reference</th>
<th>Effectiveness defined</th>
<th>Range of applicability</th>
</tr>
</thead>
<tbody>
<tr>
<td>Mandi et al. [68]; Cheremisinoff and Cheremisinoff [71]</td>
<td>$\varepsilon_T = \frac{\Delta T}{\Delta T_{ideal}}$</td>
<td>Cooling towers and humidifiers when HCR $\geq 1$</td>
</tr>
<tr>
<td>Nellis and Klien [72]</td>
<td>$\varepsilon_w = \frac{\Delta \omega}{\Delta \omega_{ideal}}$</td>
<td>Humidifier, dehumidifiers and cooling coils when HCR $\leq 1$; but not applicable for parallel flow when $\phi_i &lt; 1$</td>
</tr>
<tr>
<td>Nellis and Klien [72]</td>
<td>$\varepsilon_h = \frac{\Delta h_a}{\Delta h_{a,ideal}}$</td>
<td>Humidifier, dehumidifiers and cooling coils when HCR $\leq 1$</td>
</tr>
<tr>
<td>Jaber and Webb [73]</td>
<td>$\varepsilon_J = \frac{\dot{Q}<em>{act}}{m</em>{\text{min}} \Delta h_{a,ideal}}$</td>
<td>All heat and mass exchangers (but needs evaluation of slope of moist air enthalpy line)</td>
</tr>
<tr>
<td>Present work</td>
<td>$\varepsilon = \frac{\dot{Q}<em>{act}}{Q</em>{max}}$</td>
<td>All heat and mass exchangers; all situations</td>
</tr>
</tbody>
</table>
Chapter 4

Theoretical control-volume based analysis of existing embodiments of the HDH system

From the literature review in Chapter 2, it has been found that no study has carried out a detailed thermodynamic analysis in order to optimize the system performance of existing HDH cycles for either the water and air heated designs. In this chapter, the thermodynamic performance of these HDH cycles is analyzed by way of a theoretical cycle analysis. The control-volume based models for the humidifier and the dehumidifier, proposed in the previous chapter, are used to perform this analysis. The concept of minimum entropy generation in these basic HDH systems is developed using a new non-dimensional parameter defined in the previous chapter, HCR. The experimental validation of the theoretical results developed herein is given in Chapter 6.

In performing the analysis described in this chapter, the following reasonable approximations have been made:

- The processes operate at steady-state conditions.

- There is no heat loss from the humidifier, the dehumidifier, or the heater to the ambient.

- Pumping and blower power is negligible compared to the energy input in the
heater.

- Kinetic and potential energy terms are neglected in the energy balance.

- The water condensed in the dehumidifier is assumed to leave at a temperature which is the average of the humid air temperatures at inlet and outlet of the dehumidifier.

- It was previously shown that the use of pure water properties instead of seawater properties does not significantly affect the performance of the HDH cycle at optimized mass flow rate ratios \(^{[70]}\). Hence, only pure water properties are used in this chapter.

Engineering Equation Solver \(^{[77]}\) was used to perform the analysis. The fluid property packages and the solution technique used were already described in Section 3.1.3. The governing equations and equations for the energy effectiveness of the humidifier and the dehumidifier are the same as those described in Section 3.1.2. The energy input in the heater is given by the change in enthalpy rate of the fluid being heated.

**Terminology used**

1. *Top temperature*: In HDH systems, either water or air is heated (for example, in a solar collector). The top temperature of the cycle is the temperature of the fluid being heated at the exit of the heater.

2. *Bottom temperature*: The feedwater to the dehumidifier enters the cycle at the bottom temperature of the cycle.

3. *Terminal temperature difference* (TTD): is the stream-to-stream temperature difference at either end of the heat and mass exchanger (humidifier or dehumidifier) \(^{[95]}\).

4. *Pinch point temperature difference*: is the minimum local stream-to-stream temperature difference at any point within the heat and mass exchanger and may
be lower than both the terminal temperature differences \([95]\) in the humidifier and the dehumidifier depending on the inlet conditions being considered. In some cases, however, the pinch may be equal to one of the terminal temperature differences. In most HDH systems, the pinch point is equal to TTD in the dehumidifier and is lower than TTD in the humidifier.

4.1 Water heated HDH cycle

One of the most commonly studied HDH cycles is the closed-air open-water water-heated (CAOW) cycle (see Figure 4-1). A comprehensive study of parameters which affect the performance of this cycle has not been reported in literature. Such a study will help to understand the ways by which the performance of this basic cycle can be improved and hence, is reported below. The parameters studied include top and bottom temperatures of the cycle, mass flow rate of the air and water streams, the humidifier and dehumidifier effectivenesses and the operating pressure. The performance of the cycles depend on the mass flow rate ratio (ratio of mass flow rate of seawater at the inlet of the humidifier to the mass flow rate of dry air through the humidifier), rather than on individual mass flow rates. Hence, in this and all the preceding sections the mass flow rate ratio is treated as a variable. This variation with mass flow rate ratio was also noted by other investigators \([35, 37, 96]\).

4.1.1 Effect of relative humidity of the air entering and exiting the humidifier \((\phi_{a,1}, \phi_{a,2})\).

The humidifier and dehumidifier can readily be designed such that the relative humidity of air at their exit is one. Hence, in this paper the exit air from these components is considered to be saturated. However, the exit relative humidity is indicative of the performance of the humidifier and the dehumidifier; and hence, understanding how the variation of these parameters changes the performance of the system is important.

Figure 4-2 illustrates the effect that relative humidity of air at the humidifier inlet
Figure 4-1: Schematic diagram of water heated closed air open water HDH cycle.

and exit can has on the performance of the cycle (GOR). For this particular case, the top \((T_{w,2})\) and bottom temperatures \((T_{w,0})\) were fixed at 80°C and 35°C respectively. Humidifier and dehumidifier effectivenesses \((\epsilon_h, \epsilon_d)\) were fixed at 90%. Mass flow rate ratio was fixed at 5. It can be observed that for a variation of \(\phi_{a,2}\) from 100 to 70% the performance of the system (GOR) decreases by roughly 3%, and for the same change in \(\phi_{a,2}\) the effect is roughly 34%.

This difference suggests that the relative humidity of the air at the inlet of the humidifier has a much larger effect on performance. These trends were found to be consistent for all values of mass flow rate ratios, temperatures and component effectivenesses. This, in turn, suggests that the dehumidifier performance will have a larger impact on the cycle performance. This issue is further investigated in the following paragraphs.
Figure 4-2: Effect of relative humidity on performance of the WH-CAOW HDH cycle.

Figure 4-3: Effect of component effectiveness of humidifier on performance of the WH-CAOW HDH cycle.
4.1.2 Effect of component effectiveness ($\varepsilon_h, \varepsilon_d$).

Figure 4-3 and 4-4 illustrate the variation of performance of the cycle at various values of component effectivenesses. In Fig. 4-3, the top temperature is fixed at 80°C, the bottom temperature is fixed at 30°C and the dehumidifier effectiveness is fixed at 80%. The mass flow rate ratio was varied from 1 to 6. It is important to observe that there exists an optimal value of mass flow rate ratio at which the GOR peaks. It can also be observed that the increase in performance is fairly linear with increasing humidifier effectiveness, $\varepsilon_h$. In Fig. 4-4, the top temperature is fixed at 80°C, the bottom temperature is fixed at 30°C and the humidifier effectiveness is fixed at 80%. The cycle performance changes more dramatically for higher values of dehumidifier effectiveness. These trends are consistent for various values of top and bottom temperatures. Hence, a higher dehumidifier effectiveness is more valuable than a higher humidifier effectiveness for the performance (GOR) of the cycle.

In the previous discussion, we have observed that the dehumidifier exit air relative humidity ($\phi_{a,1}$) is more important than the humidifier exit air relative humidity ($\phi_{a,2}$). Hence, based on these results, we can say that for a water heated cycle the performance of the dehumidifier is more important than the performance of the humidifier.

4.1.3 Effect of top temperature ($T_{w,2}$)

Figure 4-5 illustrates the effect of the top temperature on the cycle performance (GOR). For this particular case, the bottom temperature ($T_{w,0}$) was fixed at 35°C and humidifier and dehumidifier effectivenesses were fixed at 92%. Top temperature ($T_{w,2}$) was varied from 50°C to 90°C. The optimal value of mass flow rate ratio increases with an increase in top temperature. Depending on the humidifier and dehumidifier effectiveness itself this trend changes. At lower component effectivenesses, the top temperature has no or little effect on the cycle performance. This result is counter-intuitive. However, it can be explained using the modified heat capacity rate ratio.

The modified heat capacity rate ratio (HCR), was defined in Section 3.2, as the
ratio of maximum possible enthalpy change in the cold stream to the maximum possible enthalpy change in the hot stream. It was found that the entropy generation in a heat and mass exchange device is minimized (for a given effectiveness and inlet conditions) when $HCR = 1$ ('balanced' condition). We are going to use this understanding here to explain the trends obtained at various top temperatures.

Figure 4-6 shows the variation of GOR with the heat capacity rate ratio of the dehumidifier ($HCR_d$). It can be seen that GOR maximizes at $HCR_d = 1$. The maximum occurs at a balanced condition for the dehumidifier which, as we have shown in the preceding paragraphs is the more important component. Further, it can be noticed from Fig. 4-7 that the degree of balancing of the humidifier at the optimum GOR condition reduces ($HCR_h$ moves farther away from 1) as the top temperature increases. Hence, the irreversibility of the humidifier (and the total irreversibility of the system) increases with increase in top temperature. A system with higher total irreversibility has a lower GOR [70]. This explains the decrease in GOR with the increase in top temperature.
Figure 4-5: Effect of top brine temperature on performance of the WH-CAOW HDH cycle.

Figure 4-6: HCR of dehumidifier versus GOR at various top brine temperatures.
Figure 4-7: HCR of humidifier versus GOR at various top brine temperatures.

Also, as the top temperature increases the dehumidifier is balanced at higher mass flow ratio and hence the optimum value of GOR occurs at higher mass flow ratios.

4.1.4 Effect of bottom temperature ($T_{w,0}$).

The bottom temperature of the cycle ($T_{w,0}$) is fixed by the seawater temperature at the location where the water is drawn. Figure 4-8 illustrates a case with top temperature of 80°C and component effectivenesses of 92%. A higher bottom temperature of the cycle results in a higher value of GOR as illustrated in the figure. This result can again be understood by plotting HCR of humidifier and dehumidifier versus the GOR of the system (Figs. 4-9 and 4-10). The degree of balancing of the humidifier at the optimum condition for GOR decreases with decrease in bottom temperature. Hence, the irreversibilities in the humidifier (and the total irreversibility of the system) increases with decreasing bottom temperature and GOR declines.
Figure 4-8: Effect of feedwater temperature on performance of the WH-CAOW HDH cycle.

Figure 4-9: HCR of dehumidifier versus GOR at various feedwater temperatures.
From the discussions in this subsection we have observed that the performance of the cycle (GOR) is a function of the following.

\[
\text{GOR} = f(\text{HCR}_h, \text{HCR}_d, e_h, e_d, T_{w,2}, T_{w,0}, \phi_{a,2}, \phi_{a,1})
\]

(4.1)

The values of GOR reported in this section for the CAOW water-heated cycle are within 20% of the experimental value obtained by Nawayseh [40] for the same boundary conditions. Further experimental validation is presented in Chapter 6.

4.2 Single and multi-stage air heated cycle

A simple [44, 45, 47, 48] air-heated cycle is one in which air is heated, humidified, and dehumidified. Current simulations have found that the GOR for this cycle is very low (GOR<1; only slightly better than a solar still). It is important to understand the reasons for this poor performance. The air in this cycle is heated and immediately sent to a humidifier where it is saturated. The air also gets cooled during the humidification
process since it is at a higher temperature than the water stream. Thus, heat is lost to the water stream in the humidifier. In the water-heated cycle, the air stream is heated in the humidifier. This further facilitates heat recovery in the dehumidifier, which is absent in an air heated system. Hence, the performance is much lower in an air-heated system.

To improve the performance of air-heated systems, Chafik [44, 97] proposed a multi-stage cycle. A three stage cycle was illustrated using a psychrometric chart in Chapter 2 (Fig. 2-5). The air in this cycle is heated and sent to a humidifier where it is saturated. It is then further heated and humidified again. The idea behind this scheme was to increase the exit humidity of the air so that water production can be increased. Chafik was able to increase the exit humidity from 4.5% (by weight) for a single stage system to 9.3% for a 4 stage system. We reproduce this result for the same cycle under similar operating conditions. However, we also observed that the GOR of the cycle rises by only 9% (Fig. 4-11). This is because the increased water production comes at the cost of increased energy input. This, in turn, is because the multi-staging does not improve the heat recovery in the humidification process. Chafik reported very high cost of water production of 28.65 Euro/m³ due in part to low energy efficiency of the system.

4.3 Chapter conclusions

A comprehensive study to understand and optimize the performance of previously studied HDH cycle configurations has been carried out. The following significant conclusions are reached:

1. The performance of a basic water-heated cycle depends on: (a) the water-to-air mass flow rate ratio; (b) the humidifier and dehumidifier effectivenesses; (c) top and bottom temperatures; and (d) relative humidity of air at the exit of the humidifier and the dehumidifier.

2. There is a specific value of the water-to-air mass flow rate ratio at which the
performance of the system is optimal. This optimal point is characterized by a thermodynamically balanced condition in the dehumidifier. The balanced condition as explained in the previous chapter is given by a modified heat capacity rate ratio of 1. This finding is important, as it is fundamental to design algorithms for HDH systems with mass extraction and injections that are developed in Chapter 5.

3. As shown in the table below, previously studied multi-stage and single-stage air heated cycles have low energy efficiency compared to the water heated HDH cycle.

4. The performance of existing HDH systems is only $1/60^{th}$ of the reversible GOR. This shows the extent of the thermodynamic losses (irreversibility) in these systems. Much of the remainder of this thesis (Chapters 5, 7, and 8) is dedicated to improving the thermal design of the HDH cycle so as to reduce the thermodynamic losses.
Table 4.1: Comparison of GOR of HDH cycles under representative boundary conditions

<table>
<thead>
<tr>
<th>CYCLE</th>
<th>GOR</th>
</tr>
</thead>
<tbody>
<tr>
<td>Single stage Air heated cycle</td>
<td>0.78</td>
</tr>
<tr>
<td>Four-stage Air heated cycle</td>
<td>0.85</td>
</tr>
<tr>
<td>Water heated cycle</td>
<td>2.5</td>
</tr>
<tr>
<td>Reversible GOR (see Appendix A)</td>
<td>122.5</td>
</tr>
</tbody>
</table>
Chapter 5

Thermodynamic balancing of HME devices and the HDH system by mass extraction and injection

When finite time thermodynamics is used to optimize the energy efficiency of thermal systems, the optimal design is one which produces the minimum entropy within the constraints of the problem (such as fixed size or cost). In this chapter, this well-established principle is applied to the thermal design of combined heat and mass exchange devices (dehumidifiers, and humidifiers) for improving the energy efficiency of HDH systems.

Mass extractions and injections which vary the water-to-air mass flow rate ratio along the fluid flow path in the humidifier and the dehumidifier is known to improve the energy efficiency of HDH systems. In the present chapter, we report a comprehensive thermodynamic analysis to understand how to design for the aforementioned mass extractions and injections in the HDH system (Fig. 2-6). This design (discussed in the succeeding sections) draws upon the fundamental observation that there is a single value of water-to-air mass flow rate ratio (for any given boundary conditions and component effectivenesses) at which the system performs optimally (see Chapter 4).

Despite all of the publications on the subject (see Chapter 2.1.4), several questions
remain unanswered. These include: (1) evaluating the upper limit on performance of a HDH system with mass extractions and injections; (2) developing design algorithms to completely balance HME devices and HDH systems; (3) understanding the effect of balancing the humidifier as opposed to balancing the dehumidifier on the performance of the HDH system; and (4) examining the effect of number of extractions on the HDH system.

5.1 Thermal balancing in simultaneous heat and mass transfer devices

A major portion of the entropy produced in the HDH system is due to the heat and mass transfer mechanisms occurring in the humidifier and the dehumidifier. Mistry et al. [98] demonstrated that at an optimal water-to-air mass flow rate ratio, 70% or more of all the entropy produced in the water-heated HDH system was produced in the humidifier and the dehumidifier. In order to reduce the entropy production of the system we have to address the entropy produced in the humidifier and dehumidifier. In this section, we revisit the algorithm (previously developed Chapter 3) for control volume balancing of HME devices and extend it to continuous and discrete balancing of these devices using mass extractions and injections. We also propose an appropriate alternative to the 'component effectiveness' (Chapter 3) and the 'temperature pinch' [51, 52, 54, 58] techniques of modeling HME devices.

5.1.1 ‘Control volume’ balancing

To understand thermodynamic balancing in HME devices let us consider the simpler case of a heat exchanger first. In the limit of infinite heat transfer area, the entropy generation rate in this device will be entirely due to what is known as thermal imbalance or remanant irreversibility. This is associated with conditions at which the heat capacity rate of the streams exchanging heat are not equal [74]. In other words, a heat exchanger (with constant heat capacity for the fluid streams) is said to be thermally
‘balanced’ (with zero remanent irreversibility) at a heat capacity rate ratio of one. This concept of thermodynamic balancing, very well known for heat exchangers, was extended to HME devices in Chapter 3.

In order to define a thermally ‘balanced’ state in HME devices, a modified heat capacity rate ratio for combined heat and mass exchange was defined by analogy to heat exchangers as the ratio of the maximum change in total enthalpy rate of the cold stream to that of the hot stream. The maximum changes are defined by defining the ideal states that either stream can reach at the outlet of the device. For example, the ideal state that a cold stream can reach at the outlet will be at the inlet temperature of the hot stream and that a hot stream can reach at the outlet will be at the inlet temperature of the cold stream. The physics behind this definition are explained in chapter 3.

\[
HCR = \left( \frac{\Delta H_{\text{max,c}}}{\Delta H_{\text{max,h}}} \right)
\]  

(5.1)

It was shown previously that at fixed inlet conditions and effectiveness, the entropy generation of a combined heat and mass exchange device is minimized when the modified heat capacity rate ratio (HCR) is equal to unity (Chapter 3.2). Further, a recent study [88] has shown that for a fixed heat transfer rate, condensation rate, and HME size, the entropy generation in a dehumidifier approaches a minimum when HCR approaches unity. Thus, we could say that HCR being unity defines the balanced state for HME devices irrespective of whether it is a fixed effectiveness or a fixed hardware condition. However, this is a ‘control volume’ balanced state wherein the design does not include mass extractions and injections. We will now try to extend the control volume concept to that of complete thermodynamic balancing in HME devices by variation of water-to-air mass flow rate ratio along the process path.
5.1.2 Enthalpy pinch: novel parameter to define performance of HME device

To clearly visualize the simultaneous heat and mass transfer process, we consider the example of a dehumidifier and plot a temperature versus enthalpy diagram (Fig. 5-1). In section 4 of a recent publication [58], we explained in detail the various approximations involved in such graphical representations. The approximations involved in Fig. 5-1 are also summarized in appendix A of the present paper.

![Figure 5-1: Temperature versus enthalpy diagram representing the dehumidification process highlighting the maximum change in enthalpy rates (per kg of dry air) that can be achieved by each of the fluid streams ($\Delta h_{\text{max},c}$ and $\Delta h_{\text{max},h}$) and the terminal enthalpy pinches ($\Psi_c$ and $\Psi_h$).](image)

In Fig. 5-1, e to f represents the process path for dehumidification of the moist air and a to b represents the process path for energy capture by the seawater stream. $f'$ and $b'$ represent the hypothetical ideal states the moist air and water stream would have, respectively, reached if the dehumidifier had been of infinite size. Hence, $h^*|_f - h^*|_{f'}$ (represented as $\Psi_h$) and $h^*|_b - h^*|_b$ (represented as $\Psi_c$) is the loss in enthalpy
rates (per unit amount of dry air circulated in the system) because of having a “finite-sized” HME device. This is the loss that we cannot reduce by thermal balancing of the device at a control volume balanced condition (without increasing the area associated with the heat and mass transfer in the device). For a given device, this is the loss that represents the energy effectiveness of the device (ε) and is directly related to the conventional definition of an exchanger effectiveness definition. This definition of effectiveness [20, 95] for a heat and mass exchanger is given as:

$$\varepsilon = \frac{\Delta \dot{H}}{\Delta \dot{H}_{\text{max}}} \tag{5.2}$$

The maximum change in total enthalpy rate is the minimum of that for the cold and the hot stream.

$$\Delta \dot{H}_{\text{max}} = \min(\Delta \dot{H}_{\text{max},c}, \Delta \dot{H}_{\text{max},h}) \tag{5.3}$$

McGovern et al. [58] proposed that it is advantageous to normalize enthalpy rates by the amount of dry air flowing through the system for easy representation of the thermodynamic processes in enthalpy versus temperature diagrams. We use this concept throughout this publication and derive the following equation from Eq. (5.2) by dividing the numerator and the denominator by the mass flow rate of dry air ($\dot{m}_{da}$).

$$\varepsilon = \frac{\Delta h^*}{\Delta h^*_{\text{max}}} = \frac{\Delta h^*}{\Delta h^* + \Psi_{TD}} \tag{5.4}$$

$$\Psi_{TD}$$ is the loss in enthalpy rates at terminal locations because of having a “finite-sized” HME device and is defined as follows:

$$\Psi_{TD} = \min \left( \frac{\Delta \dot{H}_{\text{max},c}}{\dot{m}_{da}} - \Delta h^*, \frac{\Delta \dot{H}_{\text{max},h}}{\dot{m}_{da}} - \Delta h^* \right) = \min(\Psi_c, \Psi_h) \tag{5.6}$$

In the case of a heat exchanger, $$\Psi_{TD}$$ will be analogous to the minimum termi-
nal stream-to-stream temperature difference (TTD). Assuming the hot stream is the minimum heat capacity stream, we may derive the equations for the effectiveness of a heat exchanger as given below (Eqs. 5.8 & 5.9).

\[
\varepsilon_{HE} = \frac{(m_{c_p})_h \Delta T_h}{(m_{c_p})_h (T_{h,in} - T_{c,in})} = \frac{\Delta T_h}{\Delta T_h + (T_{h,out} - T_{c,in})} \quad (5.9)
\]

The extension to the case where the cold stream is the minimum heat capacity stream is similar. By comparing Eqs. 5.5 & 5.9, the analogy is clear.

TTD is seldom used to define performance of a heat exchanger in thermodynamic analyses; the temperature pinch is the commonly used parameter. The difference is that the pinch is the minimum stream-to-stream temperature difference at any point in the heat exchanger and not just at the terminal locations. Like temperature pinch, \( \Psi \) can be defined as the minimum loss in enthalpy rate due to a finite device size at any point in the exchanger and not just at the terminal locations. This is accomplished, as shown in Fig. 5-2, by considering infinitely small control volumes represented by just two states (g for air and i for water). We can define the ideal states for each of these real states as \( g' \) and \( i' \). The local \( \Psi \) at this location can be defined as the minimum of \( h_{i'g} - h_{igi} \) (represented as \( \Psi_2 \)) and \( h_{ig} - h_{igi} \) (represented as \( \Psi_1 \)). Thus, the general definition of \( \Psi \) will be as follows:

\[
\Psi = \min_{local} (\Delta h^*_{\max} - \Delta h^*) \quad (5.10)
\]

Hence, based on the arguments presented in this section, we can say that \( \Psi \) for a HME device is analogous to temperature pinch for an HE, and it can be called the 'enthalpy pinch'. We recommend that, because of the presence of the concentration difference as the driving force for mass transfer in HME devices, a temperature pinch or a terminal temperature difference should not be used when defining the performance of the device.
Figure 5-2: Temperature versus enthalpy diagram for the dehumidification process highlighting 'loss in ideal enthalpy' or enthalpy pinch at any given location ($\Psi_{local}$) as a measure of local effectiveness in HME devices.

The energy effectiveness is another commonly used performance metric for HEs [95] and HMEs [20]. But, this is a control volume parameter and accounts for only terminal differences. In order to design for balancing, we need to consider local differences. Consider the temperature profile of a humidification process as shown in Fig. 5-3: the 'pinch' point does not occur at the terminal locations but rather at an intermediate point. This behaviour is not captured if we define the performance of the device by an energy effectiveness. In the extreme case, as demonstrated in Fig. 22 of Miller et al. [99], high values of effectiveness for the humidifier could lead to an internal temperature and concentration cross. $\Psi$ does not have this problem since it is a local parameter and is, hence, used to define the performance of HME devices (humidifiers and dehumidifiers) in this study.
Figure 5-3: Temperature versus enthalpy diagram representing the humidification process highlighting the ‘pinch point’ occurring at an intermediate location rather than at a terminal one.

5.1.3 Mass extractions or injections based balancing

As described in Sec. 5.1.1, a value of unity for the modified heat capacity rate ratio defines a thermally balanced state for a control volume without extractions. For such a case HCR is not equal to unity at all locations in the device. With mass extractions or injections we can vary the slope of the water line such that HCR is one throughout the device. This is the operating condition at which the HME device is completely balanced. We rewrite the expression for HCR in terms of $\Psi_c$ and $\Psi_h$ to understand this concept.

\[
\text{HCR} = \frac{\Delta h_{\text{max},c}}{\Delta h_{\text{max},h}} = \frac{\Delta h^* + \Psi_c}{\Delta h^* + \Psi_h} \tag{5.11}
\]
To vary the water-to-air mass flow rate ratio such that \( \text{HCR}=1 \) at every location in the device (or conversely \( \Psi = \) constant at every point) we need extractions or injections at every point (the number of extractions of injections are infinitely many). We call this “continuous thermodynamic balancing”. Even though this has theoretical significance in understanding systems with mass extraction and injection, in practice it will be difficult to achieve. Hence, we also evaluate balancing a HME device with a finite number of injections. In the cases reported in this paper, we investigate a single extraction or injection alone.

As can be understood by looking at Figs. 5-1 and 5-2, in a ‘control volume’ balanced dehumidifier without extractions or injections, the local \( \Psi \) is minimum at the two terminal locations (also see Eq. 5.13), and at all intermediate points it is higher. This results from the nature of the temperature-enthalpy diagram as discussed in more detail in Sec. 5.2. The local variation of \( \Psi \) in the control volume balanced case is illustrated in Fig. 5-41. As may be observed from the figure, a single injection brings \( \Psi \) to a minimum value at one intermediate location (or conversely brings \( \text{HCR}=1 \) at that location and the two terminal ones). In the case of the number of injections approaching infinity, local value of \( \Psi \) can be minimum and constant throughout the length of the device (Eq. 5.14). The direction of the injection of air is to the dehumidifier. Since, we need to vary the water-to-air mass rate ratio to balance the device (and not individual mass flow rates) we can equivalently inject water into the (counterflow) dehumidifier.

Figure 5-5 illustrates the effect of continuous and single injections on the total irreversibility in the dehumidifier. The entropy produced per unit amount of water condensed is reduced to a quarter with continuous injection and to 3/5th with a single injection. This is representative of an optimal case. Such a large reduction demonstrates the importance of thermodynamic balancing for heat and mass exchangers.

---

\[ \Psi_{TD,c} = \Psi_{TD,h} \] (5.13)

\[ \text{when HCR}=1 \text{ for the CV, } \]

\[ \Psi = \text{constant} \] (5.14)

---

\(^1\)For the \( z \)-axis in Fig. 5-4, the specific enthalpy per kg of dry air (used to describe the control volume location in Figs.(5-1-5-2)) is normalised by the total heat duty (\( \Delta h^* \)). This convention is used in the rest of the thesis.
Figure 5-4: A plot of local enthalpy pinch values ($\Psi_{local}$) relative to the overall enthalpy pinch ($\Psi$) to illustrate the effect of extractions in a dehumidifier with the control volume balanced case.

Figure 5-5: Effect of extraction on the irreversibility in the dehumidifier evaluated at $T_a = 20^\circ C$; $T_e = 70^\circ C$; $\Psi_{deh} = 20$ kJ/kg dry air; HCR =1.
5.1.4 Functional form for continuous thermodynamic balancing

Considering Eq. 5.14, we can write down the closed form expressions [Eqs. (5.15-5.20)] for the temperature and humidity ratio profiles for the fluid streams in a completely balanced dehumidifier and humidifier. If the process path for air (represented in an enthalpy-temperature diagram) follows a function $\xi$ (Eq. 5.15) then the mass flow rate ratio is varied in the dehumidifier such that the seawater process path is the same function of enthalpy, but shifted by $\Psi$ (Eq. 5.17). A similar shift in the enthalpy is also followed in the humidity profile (Eqns. 5.16 & 5.18).

\[
\begin{align*}
T_a &= \xi(h^*) \\
\omega &= \eta(h^*) \\
T_w &= \xi(h^* - \Psi) \\
\omega_{\text{int}} &= \eta(h^* - \Psi)
\end{align*}
\]

An example of a temperature and humidity profile in a dehumidifier with continuous injection is illustrated in Fig. 5-6. It can be seen from Fig. 5-6 that a dehumidifier with continuous mass injections (such that HCR = 1 throughout the device) has a profile close to a constant driving humidity difference\(^2\) rather than a constant temperature difference. This is a very significant conclusion and is further corroborated by results obtained from a transport process analysis by Thiel & Lienhard [88]. It also leads us to conclude that balancing for temperature differences alone (as carried out by all previous studies reviewed in Chapter 2) will not lead to a thermodynamic optimum.

For a completely balanced humidification device, the concept is similar. For a moist air line represented by Eq. 5.15 & 5.16, the humidifier water lines will be given

\(^2\)Driving humidity difference is calculated as the difference in the local humidity ratio of the bulk air stream (avluate at a bulk temperature) and the humidity ratio of the interface (evaluated as saturated at the interface temperature).
Figure 5-6: An illustration of (a) temperature and (b) humidity ratio profiles in an dehumidifier with complete thermodynamic balancing by continuous injection.
The complete injection profiles can be obtained by only varying the water-to-air mass flow rate ratio. This can be done by continuous extraction or injection of either the air or the water (or both) from or into the HME device.

5.2 Modeling of HDH systems

In the current section, we use the concepts of thermodynamic balancing developed for HME devices and apply them to the HDH system design. An embodiment of the system under study is illustrated in Fig. 2-6.

5.2.1 System without extractions

A temperature-enthalpy diagram for the HDH system without extractions (illustrated earlier in Fig. 2-6) is shown in Fig. 5-7. The process line for the air is represented by the saturation line ‘ef’ in the humidifier and the dehumidifier. The uncertainty in the calculated performance of the HDH system as a result of the approximation that air is saturated all along its process path is small and is discussed in detail in Sec. 5.3.3. The seawater process line is represented by ‘ab’ in the humidifier, by ‘bc’ in the heater and by ‘cd’ in the dehumidifier.

A detailed algorithm to design this system using the top brine temperature, the feed water temperature and the component enthalpy pinches as input variables is elucidated in Fig. C-1 of Appendix C. The design of the HDH system using temperature-enthalpy diagrams was also previously discussed by other researchers [49, 51, 55, 58]. A temperature pinch was used in that study instead of an enthalpy pinch used in the current publication. As illustrated in Fig. C-1, the solution is iterative and the thermophysical properties are evaluated as described in Sec. E.1.
Figure 5-7: Temperature profile representing the HDH system without extractions or injections. Boundary conditions: \( T_a = 20^\circ C; T_c = 80^\circ C; \Psi_{deh} = \Psi_{hum} = 20 \text{ kJ/kg dry air.} \)

Other than the energy and mass conservation equations described in Chapter 3.1.2, the understanding that the slope of the water line in the temperature versus enthalpy diagram can be used to evaluate the mass flow rate ratio at any given point in the HME devices is important to the analysis:

\[
slope = \frac{dT_w}{dh^*} = \frac{1}{m_r c_p, w} \tag{5.21}
\]

Further, the entropy of the various states evaluated using the temperature-enthalpy diagram may be used to evaluate the mass flow rate in the humidification and the dehumidification devices.
5.2.2 System with infinite extractions and injections

Equations (5.15-5.20) are fundamental to designing systems with infinite extraction such that the remanent irreversibility in one of the humidifier or the dehumidifier is zero. Fig. 5-8 illustrates the application of the aforementioned equations in system design via temperature versus location diagrams. From a pinch point perspective, the temperature pinch in the humidifier and the dehumidifier are at different terminal ends in the 'dehumidifier balanced' and 'humidifier balanced cases'. The enthalpy pinch, however, is minimum and constant at all points in the dehumidifier and humidifier in the two respective cases.

The detailed procedure to model the system with infinite extractions illustrated in Fig. C-2 of Appendix C. In developing this procedure we have put in a place a constraint that the state (temperature and humidity) of the injected stream is the same as the stream it is injected into. This is done to avoid generating entropy due to mixing streams which are at dissimilar states. Further, it is important to note that air in the dehumidifier has the same inlet and outlet temperature and humidity unlike water which has a different streamwise temperature in the humidifier and the dehumidifier (due to the presence of the heater). Thus for the HDH system under study in this chapter, it is not possible to perform water extractions and injections without either generating entropy due to mixing or without limiting the number of extractions. Hence, air extraction is studied in this publication.

5.2.3 System with a single extraction and injection

It is, perhaps, more practical to apply a finite number of extractions and injections in the HDH system. Hence, we study the effect of a single extraction in this publication along with that of infinite extractions. Fig. 5-9 illustrates a temperature profile of a system with a single extraction and injection. In the illustrated case, the air was extracted from the dehumidifier at the state 'ex' and injected in a corresponding location in the humidifier with the same state 'ex' to avoid generating entropy during the process of injection. This criteria for extraction is applied for all the cases
Figure 5-8: Temperature profiles representing the HDH system with continuous extractions to completely balance (a) dehumidifier and (b) humidifier. Boundary conditions: $T_a = 20^\circ$C; $T_c = 80^\circ$C; $\Psi_{deh} = \Psi_{hum} = 20 \text{ kJ/kg dry air}$.
reported in this paper since it helps us study the effect of thermodynamic balancing, independently, by separating out the effects of a temperature and/or a concentration mismatch between the injected stream and the fluid stream passing through the HME device (which when present can make it hard to quantify the reduction in entropy generated due to balancing alone).

The detailed procedure to model the system with a single air extraction and injection is illustrated in Fig. C-3 of Appendix C.

5.2.4 Property packages

- The thermophysical properties of seawater were evaluated using the correlations presented by Sharqawy et al. [100].

- Thermophysical properties of pure water are evaluated using the IAPWS (International Association for Properties of Water and Steam) 1995 Formulation [78].

- Moist air properties are evaluated assuming an ideal mixture of air and steam using the formulations presented by Hyland and Wexler [80].

- Moist air properties thus calculated are in close agreement with the data presented in ASHRAE Fundamentals [81] and pure water properties are equivalent to those found in NIST's property package, REFPROP [82].

5.3 Results and discussion

In this section, we investigate the effect thermodynamic balancing can have on the energy performance of the HDH system. First, we attempt to design a completely reversible HDH system. Then, we use this as a basis to investigate the effect of having finite-sized systems, of balancing the humidifier versus the dehumidifier, and of the number of extractions.
Figure 5-9: Temperature profile representing the HDH system with a single extraction. Boundary conditions: $T_a = 20^\circ$C; $T_c = 80^\circ$C; $\Psi_{deh} = \Psi_{hum} = 20$ kJ/kg dry air.
5.3.1 Continuous extractions with “infinitely large” HME devices: the upper limit on HDH performance

In section 5.1, we explained in detail how the ‘remanent’ irreversibility (defined by Bejan [74]) is brought down to zero and complete thermodynamic balancing is achieved in a HME device. We use the closed form expressions [Eqs. (5.15–5.20)] presented in Sec. 5.1.4 to design a completely reversible HDH system. To achieve this we need to consider an infinitely large dehumidifier and humidifier (with enthalpy pinch, \( \Psi_{deh} = \Psi_{hum} = 0 \) kJ/kg dry air).

Figure 5-10 illustrates the mass flow rate ratio and HCR profiles for a HDH system with 100% effective humidifier and dehumidifier and complete thermodynamic balancing in the dehumidifier. It may be observed that the water-to-air mass flow rate ratio has to be varied from 1 to 31 in a continuous manner to achieve a spatially constant HCR of unity in the dehumidifier. For the system with the extraction profile as shown in Fig. 5-10, and at a feed temperature \((T_a)\) of 20°C, salinity of 35 g/kg and a top brine temperature \((T_c)\) of 80°C, the GOR was found to be 109.7 and the RR was 7.6%. The total entropy produced per unit amount of water distilled in the system was minimized to \(10^{-3}\) kJ/kg-K.

The GOR achievable in a completely reversible HDH cycle may be evaluated using Eq. (A.7) derived in Appendix A. For a feed temperature \((T_a)\) of 20°C, a top brine temperature \((T_c)\) of 80°C and a recovery ratio of 7.6%, the reversible GOR that can be achieved is 123.3.

Thus, with complete thermodynamic balancing (infinite extractions and injections) and infinite system size, the performance of the HDH system is about 88% of the reversible limit. Complete reversibility cannot be achieved because it is only possible to fully balance either the dehumidifier or the humidifier in a given system and not both in the same design\(^3\) (in Sec. 5.3.4, we show that balancing either the

---

\(^3\)Both the humidifier and dehumidifier can be balanced in the same design only if a way to modify the process path for the air is possible. For example, if we were able to tailor different enthalpy-temperature functions for the moist air line in the humidifier and dehumidifier by modifying the physics of these processes. However, this is very hard to realise in a real design and hence, in the current publication we make the reasonable assumption that both the process paths are along the
Figure 5-10: Mass flow rate ratio and HCR profile for complete thermodynamic balancing in a HDH system with 100% effective humidifier and dehumidifier. Boundary conditions: $T_a = 20^\circ C$; $S = 35$ g/kg; $T_c = 80^\circ C$; $\Psi_{deh} = \Psi_{hum} = 0$ kJ/kg dry air; $N = \infty$; System performance: $\text{GOR} = 109.7$; $\text{RR}=7.6\%$. 
dehumidifier or the humidifier yields similar results). Thus, we conclude that the upper limit for HDH performance is below the reversible limit for thermal desalination systems.

5.3.2 Effect of finite system size

In the preceding section, we investigated 'infinitely' large HDH systems. This, of course, is a theoretical exercise to understand the performance limit of the system. In a real system, the humidifier and the dehumidifier will have an enthalpy pinch greater than zero. For example, a five-stage bubble column dehumidifier described in Chapter 9 has a $\Psi$ of 15 kJ/kg dry air.

Figure 5-11 illustrates the effect of having finite size humidifier and dehumidifier on the system performance. As may be observed, the GOR values drop off rapidly as the enthalpy pinch increases. For example, when the enthalpy pinch is around 15 kJ/kg in the dehumidifier and the humidifier, the GOR is about 5 with infinite extractions. This is a large reduction from the GOR of 109.7 for the $\Psi = 0$ case. It leads us to conclude that thermodynamic balancing works best in systems with low enthalpy pinches in the dehumidifier and the humidifier. Further evidence corroborating this conclusion is described in Sec. 5.3.5.

5.3.3 Uncertainty associated with 'saturated air' approximation

Figure 5-11 also helps us quantify the uncertainty associated with assuming the process path for air to be along the saturation line. Thiel & Lienhard [88] performed boundary layer analysis on a dehumidifier and found that (based on the mass-averaged and energy-averaged definition of the 'bulk' state) the air follows a path different from the saturation curve (with a maximum deviation of about 10% in terms of the humidity ratio and the enthalpy associated with the terminal and the intermediate states in the process path). From Fig. 5-11, it may be observed that the propagated saturation line (see sec. 5.3.3).
Figure 5-11: Effect of having finite-size HME devices on the performance of the HDH system with infinite extractions highlighting the maximum possible uncertainty associated with using the saturation line as the air process path. Boundary conditions: $T_a = 20^\circ$C; $S = 35$ g/kg; $T_c = 80^\circ$C; $N = \infty$; $HCR_{deh} = 1$. 

* calculated using saturated air approximation

× calculated using mass averaged boundary layer data [15]
uncertainty in the GOR value due to the aforementioned deviation from the saturation line approximation is small. The uncertainty is less than 1% at $\Psi$ values close to zero and reaches a maximum uncertainty of 11% at a $\Psi$ of 27 kJ/kg of dry air ($\Psi$ values greater than 27 kJ/kg are not of interest in this publication because of reasons stated later in this section). It is important to note that at $\Psi = 27$ kJ/kg of dry air, a 11% variation corresponds to an uncertainty of only 0.3 in terms of the GOR value.

5.3.4 Comparison of dehumidifier balanced and humidifier balanced systems

In Fig. 5-8, we illustrated the temperature profiles for two HDH systems: one with a balanced dehumidifier and the other with a balanced humidifier. In this section, we compare the performance of these two systems at various values of enthalpy pinch. As may be observed from Fig. 5-12, the performance is fairly similar. At lower values of enthalpy pinch ($\Psi < 7$ kJ/kg dry air) the dehumidifier balanced system has a slightly higher performance and at higher values of enthalpy pinch the humidifier balanced system is marginally better.

To understand the similar GOR values for the two systems studied in this section, let us consider Fig. 5-13. The entropy generated in the humidifier and the dehumidifier per kilogram of water desalinated in the system is illustrated for a fixed top brine temperature, feed water temperature and enthalpy pinches in the humidifier and the dehumidifier. When we completely balance the dehumidifier for this system, we reduce the entropy generated in the dehumidifier to a quarter of that in a system without mass extractions and injections. However, the entropy generated in the humidifier is increased by 65%. While we are balancing the dehumidifier, the humidifier is moving away from the balanced state. In the system with a completely balanced humidifier, the entropy generation in the humidifier is reduced to less than a third of that in a system without mass extractions or injections. The entropy generated in the dehumidifier changes little. The total entropy generated in the system per kg of water desalinated is about the same for both the system discussed here and hence
Figure 5-12: Comparison of performance of the HDH system with infinite extractions for complete thermodynamic balancing of humidifier with that for complete thermodynamic balancing of the dehumidifier. Boundary conditions: $T_a = 20^\circ\text{C}; S = 35\text{ g/kg}; T_c = 80^\circ\text{C}; N = \infty; \text{HCR}_{deh} = 1.$
these systems have a similar GOR value. We have observed a similar trend for other boundary conditions too.

In conclusion, based on studying the changes in entropy generated due to balancing in the various cases reported in this section, it was found that the reduction in total system entropy generation due to continuous balancing is very similar at the same enthalpy pinches for the 'dehumidifier balanced' and the 'humidifier balanced' systems. Hence, the GOR was also found to be similar.

5.3.5 Effect of number of extractions

The effect of the number of extractions (at various enthalpy pinches) on the performance of the HDH system is shown in Fig. 5-14. Several important observations can be made from this chart.

First, it may be observed that thermodynamic balancing is effective in HDH cycles only when the humidifier and the dehumidifier have an enthalpy pinch less than about 27 kJ/kg dry air. For various boundary conditions it has been found that beyond the aforementioned value of enthalpy pinch the difference in performance (GOR) with that of a system without any extractions or injections is small (less than 20%). Further, at very low values of the enthalpy pinch ($\Psi \leq 7$ kJ/kg dry air) in the humidifier and the dehumidifier, continuous balancing with infinite number of extractions and injections was found to give much better results than that with a single extraction and injection. For the top brine temperature of 80°C, a feed water temperature of 20°C and 'infinitely' large humidifier and dehumidifier ($\Psi_{hum} = \Psi_{deh} = 0$ kJ/kg dry air), the GOR was found to be 8.2 for a single extraction (compared to a GOR of 109.7 for a similar system with infinite extractions). At higher values of enthalpy pinch ($7 < \Psi \leq 15$), a single extraction reduced the entropy generation of the total system roughly by a similar amount as an infinite number of extractions. At even higher values of enthalpy pinch ($15 < \Psi \leq 27$), a single extraction outperforms infinite extractions. This is a very surprising result. We try to understand this by looking at how the infinite and single extraction balancing affect the entropy generation in the humidifier and dehumidifier (see Fig. 5-15).
Figure 5-13: Reduction in total system irreversibility with complete thermodynamic balancing of either the humidifier or the dehumidifier in HDH. Boundary conditions: $T_a = 20^\circ$C; $S = 35$ g/kg; $T_c = 80^\circ$C; $\Psi_{deh} = \Psi_{hum} = 20$ kJ/kg dry air; $HCR_{deh}=1$ or $HCR_{hum}=1$. 
Figure 5-14: Effect of number of extractions (for thermodynamic balancing) on the performance of the HDH system with finite and infinite size HME devices. Boundary conditions: $T_a = 20^\circ C; S = 35 \text{ g/kg}; T_c = 80^\circ C; \text{HCR}_{deh}=1$. 

- $N\rightarrow\infty$ 
- $N=1$ 
- $N=0$
Figure 5-15 illustrates the entropy generated in the humidifier and the dehumidifier in systems with zero, one and infinite extractions/injections at component enthalpy pinches of 20 kJ/kg dry air. It may be observed that when continuous extractions are applied, the entropy generated in the balanced component (the dehumidifier) is reduced but the entropy generated in the humidifier is increased. In other words, the humidifier is ‘de-balanced’ as the dehumidifier is balanced. For the single extractions case, even though the entropy generated in dehumidifier is reduced by a smaller amount than that in the infinite extractions case, the humidifier is not de-balanced. Thus, the total entropy generated is lower in the single extraction case and the GOR is higher.

5.4 Chapter conclusions

In the first half of this chapter, a detailed study of thermodynamic balancing in HME devices is carried out. The following is a summary of the main conclusions of that study.

1. A novel “enthalpy pinch” has been defined for combined heat and mass exchange devices. This definition is analogous to the temperature pinch traditionally defined for heat exchangers. The enthalpy pinch ($\Psi$) combines stream-to-stream temperature and humidity ratio differences, and is directly related to the effectiveness of the device. We recommend it for use in thermodynamic analysis of systems containing HME devices.

2. Closed form equations for the temperature and humidity ratio profiles of a completely and continuously balanced HME device with zero ‘remanent’ irreversibility is presented in this paper for the first time in literature.

3. It is observed that this state of complete thermodynamic balancing (in humidifiers and dehumidifiers) is closer to a state of constant local humidity ratio difference than to that of a constant stream-to-stream temperature difference.
Figure 5-15: Effect of extraction on total system irreversibilities. Boundary conditions: $T_a = 20^\circ C; S = 35 \text{ g/kg}; T_c = 80^\circ C; \Psi_{deh} = \Psi_{hum} = 20 \text{ kJ/kg dry air}; \text{HCR}_{deh}=1.$
4. By continuous injection of mass in a dehumidifier, the entropy generation in the device can be brought down to $\frac{1}{4}$th of that in a device without injections. By a single extraction it can be brought down to $\frac{3}{5}$th. Either water or air may be injected from the dehumidifier in these cases.

Further, these observations were used in the second part of the chapter for the design of thermodynamically balanced HDH systems, and the following are the salient features of that part of the study.

1. Detailed algorithms for design of HDH systems with mass extractions and injections using the temperature versus enthalpy diagram have been developed in this chapter. These were developed for both continuous and discrete extractions and injections.

2. An *almost* completely reversible HDH system was designed using an "infinitely large" humidifier and dehumidifier with continuous mass extraction and injection. A theoretical gained-output-ratio of 109.7 approaching the reversible limit of 123.3 was evaluated for this ideal system with the total entropy generation approaching zero ($\frac{\dot{S}_{g}}{\dot{m}_{w}} \approx 10^{-3}$ kJ/K·kg water produced).

3. The uncertainty of the final results reported in the paper associated with the approximation of the air being saturated at all points in the humidification and dehumidification processes was evaluated to be reasonably small based on the boundary layer data from Thiel and Lienhard [88].

4. It is found that the performance of an HDH system with a completely balanced humidifier and that with a completely balanced dehumidifier are similar. This is explained by examining the entropy generated in each component in the system in each case.

5. It is found that thermodynamic balancing is effective in HDH only when the HME devices have an appropriately low enthalpy pinch ($\Psi \lesssim 27$ kJ/kg dry air).
6. At very low values of the enthalpy pinch ($\Psi \leq 7$ kJ/kg dry air) in the humidifier and the dehumidifier, continuous balancing with an infinite number of extractions and injections was found to give much better results than that with a single extraction and injection. At higher values of enthalpy pinch ($7 < \Psi \leq 15$), a single extraction reduced the entropy generation of the total system by a similar amount as infinite extractions. At even higher values of enthalpy pinch ($15 < \Psi \leq 27$), single extraction outperformed infinite extractions and at $\Psi > 27$, thermodynamic balancing has no significant effect on the performance of the HDH system.

These results are experimentally validated in a pilot scale HDH unit. The details of the same are described in the following chapter.
Chapter 6

Experimental investigation of thermal design of HME devices and HDH systems

Maximization of energy efficiency of a thermal system is at its essence the minimization of the total entropy generation in the system (and in the environment) within the constraint of a fixed system size or cost. In this study, we perform such an optimization for simultaneous heat and mass exchange devices with a fixed size constraint, specifically for the thermal design of humidification dehumidification (HDH) desalination systems.

A pilot scale experimental unit that incorporates the thermodynamic cycle shown in Fig. 6-1 (with a daily water production of 700 liters) has been constructed and instrumented, detailed experiments conducted on the system are reported in this publication. This unit also facilitates detailed experimentation on the humidification device and the water heated CAOW HDH system without mass extraction. These measurements substantiate the previously developed theory for such extractions (see Chapter 5), and they also provide significant new insights.
Figure 6-1: Schematic diagram of a water-heated, closed-air, open-water humidification-dehumidification desalination system with mass extraction and injection of the moist air stream.

Summary of design theory

The theoretical framework for design of HME devices for implementation in the HDH system has been developed over a few preceding chapters of the current thesis. The linchpin in this theoretical work is the definition of a novel parameter known as the ‘modified heat capacity rate ratio’ (HCR). A brief summary of the definition of this parameter and its significance to thermal design of HME devices and the HDH system is given below.

It was shown previously that at fixed inlet conditions and effectiveness, the entropy generation of a combined heat and mass exchange device is minimized when
the modified heat capacity rate ratio (HCR) is equal to unity [21]. Figure 3-9 illustrated an example of this result for a humidifier. This figure elucidated that the non-dimensional entropy production (as defined in chapter 3) plotted against modified heat capacity rate ratio for various values of air inlet temperature at fixed values of energy effectiveness and inlet conditions of air and water streams. Irrespective of the value of air inlet temperature, non-dimensional entropy generation is minimized at HCR=1. It was also found that this result is true irrespective of the values of the other fixed conditions.

Further, a recent study [88] has shown that for a fixed heat transfer rate, condensation rate, and HME size, the entropy generation in a dehumidifier approaches a minimum when HCR approaches unity. Miller [101] developed numerical models to simulate heat and mass transfer in a fixed area HME device (humidifier and dehumidifier) and reported that at HCR=1 the non-dimensional entropy generation is minimized for these devices. Thus, we could say that HCR being unity defines the balanced state for HME devices regardless of whether it is a fixed effectiveness or a fixed hardware condition.

Past work also showed the importance of HCR to the performance of the water heated HDH system shown in Fig. 6-1 without mass extractions or injections [19]. Figure 4-6 shows the variation of performance of the system (gained output ratio or GOR) with the heat capacity ratio of the dehumidifier (HCRd). It can be seen that GOR is maximized at HCRd = 1. The maximum occurs at a balanced condition for the dehumidifier, which is the more irreversible component in this particular cycle.

Thus, we had theoretically shown that HCR=1 is the balanced state for HME devices and is the criteria for optimal performance of HDH systems. This chapter experimentally validates those conclusions. Moreover, the fixed hardware analysis of the HDH system and the importance of HCR to that analysis is reported in this chapter.
6.1 Control volume balancing of HME devices

As described in Sec. 6, theoretical considerations show that a modified heat capacity rate ratio (HCR) of 1 will lead to minimum entropy generation in a fixed effectiveness or fixed hardware device and that the condition should be established to optimize the thermal performance of the HDH cycle [19]. In this section, this important conclusion is investigated experimentally.

6.1.1 Experimental details

A 3 meter tall packed bed humidifier with a 0.278 m² cross section packed with heat transfer fills was built to transfer heat and mass simultaneously between an air stream and a heated water stream (a 3D rendering of the unit is shown in Fig. 6-2). The column was built out of transparent acrylic and was split into three sections: (1) the air chamber or the bottom section - where the air enters the system and the liquid is discharged from the system; (2) the middle section with the fill material in it where direct contact heat and mass exchange between the water droplets and the air stream is affected; and (3) the top section where the hot water from a heater is sprayed on the packing. The three sections were held together by flanges and were made easy to disassemble. The air is circulated via forced draft using an air blower and the water is circulated via a pump. The flow rate of the water is adjusted by an inline valve attached to the pump. Inside the humidifier are blocks of corrugated plastic sheets. In the current setup, polypropylene packing material from Brentwood Industries (model number: CF1200 MA) with a specific area of 226 m²/m³ was used. Further details about construction of the humidifier is explained by Maximus St. John [102].

The flow rate of water is measured by a rotameter and the inlet temperature of the water can be adjusted by the inline electric water heater. The flowrate of air is measured via a separate rotameter. The inlet and outlet water and air temperatures are measured by standard K-type thermocouples. The thermocouples and the data logging system have an uncertainty of ± 0.1°C. The rotameters used for air flow measurements have a range of 5-50 ft³/min (2360 - 23600 cm³/s) with a least count
Figure 6-2: Three dimensional rendering of the experimental humidifier unit.
of ± 0.2 ft³/min (± 94.4 cm³/s). The rotameter used for water flow measurement has a range of 0.4 - 2 US gallon/min (25.2 - 126.2 cm³/s) with a least count of 0.02 gpm (1.26 cm³/s). Visual observation showed the appearance of supersaturation of air with water vapor at the exit. This is also a commonly known fact among researchers in the field. Exit humidity measurements were, hence, not required for these sets of experiments.

6.1.2 Results

Minimum effectiveness at the balanced condition Figure 6-3 illustrates the energy effectiveness and the heat capacity rate ratio of the humidifier at various hot water inlet temperatures to the device. This was measured at a fixed value of air inlet dry and wet bulb temperatures, a fixed value of water-to-air mass flow rate ratio and at atmospheric pressure. The maximum uncertainty on the calculated effectiveness value is ±1% and that on the heat capacity rate ratio is ±1.5%.

It may be immediately observed that there exists a water inlet temperature at which the energy effectiveness is minimum. This corresponds to the case closest to a HCR of 1. This result is consistent with that demonstrated theoretically [101]. It is also consistent with the well-established theory for heat exchangers which have a minimum value of effectiveness at the 'balanced' state.

Condition for minimum entropy generation Figure 6-4 illustrates that there exists a particular mass flow rate at which non-dimensional entropy generated in the device is minimized. This is at fixed inlet air condition and fixed inlet water temperature. At different values of these boundary conditions, the same result was found to be true. The minimum that is observed also corresponds to the case closest to an HCR of 1. This is consistent with the theoretical observation that irreversibility is minimized at HCR of unity [21, 88, 101].

Effect of HME size We have previously observed that thermodynamic balancing is most effective when the size of the HME device is large and the driving forces are
Figure 6-3: Effectiveness and heat capacity rate ratio versus water inlet temperature in the humidifier. Boundary conditions: $T_e = 32^\circ C; m_e = 2.85; T_{wb,e} = 20^\circ C; P = 101.3$ kPa; $V_h = 0.27$ m$^3$

Figure 6-4: Effect of mass flow rate ratio on non-dimensional entropy generation in the humidifier. Boundary conditions: $T_e = 32^\circ C; T_c = 60^\circ C; T_{wb,e} = 20^\circ C; P = 101.3$ kPa; $V_h = 0.27$ m$^3$
relatively small [23]. Here, we investigate the effect of HME size on the effectiveness and enthalpy pinch of the device. To calculate the enthalpy pinch, a temperature enthalpy diagram was simulated using the measured exit states of air and water. The air was approximated to follow the saturated air process path. This approximation was found to be reasonable in Chapter 5. Figure 6-5 illustrates that as we increase the packing volume (each block of packing adds 0.07 m³ of volume), the effectiveness increases rapidly till it flattens out at 80% and the enthalpy pinch drops rapidly from 80 to 14.8 kJ/kg of dry air.

6.1.3 Section conclusions

From the data presented in this section, we can draw the following important conclusions about performance of HME devices:
1. A modified heat capacity rate ratio of 1, under the fixed hardware condition and at fixed inlet states for the two streams exchanging heat and mass, sets the optimum mass flow rate ratio at which the entropy generation in the device is minimized. This is consistent with the on-design (fixed effectiveness) theory developed earlier.

2. As is well-established in regular heat exchanger theory, for an HME device as well the energy effectiveness is minimized at a HCR of 1.

3. A small enthalpy pinch of 14.8 kJ/kg dry air is obtained in the pilot scale humidification device built. This corresponds to a temperature pinch of 2.8°C.

### 6.2 HDH system experiments

Figure 6-6 is a photograph of the pilot scale CAOW water heated HDH unit built. The humidifier used in the unit was already described in detail in Sec. 6.1.1. Hooked up to the humidifier were four polypropylene plate-and-tube dehumidifiers (setup in series) procured from George Fischer LLC. A photograph of the same is shown in Fig. 6-7.

In this dehumidification device, air flows through the center and exchanges energy with a water stream flowing through a series of plates in plane with the cross section. The air stream is dehumidified in the process producing a condensate stream and the water stream is preheated before it is sent to the electric heater where the energy is input for the desalination process. The plates each consist of 35 parallel 0.189” (0.48 cm) ID tubes fed and drained by two section headers. Barriers at opposite ends of the headers divide it such that water can only flow from one side of the header to the other via the parallel tubes connecting them. As shown in Figure 6-7, water enters the header through an inlet in one corner and flows through the parallel tubes to the opposite header section where it drains into the header below it. Since the headers separate the hot air flow from the external environment, heat loss in these devices is reduced. Each of these dehumidifiers had a heat transfer area of 2 m² (for a total of 8
Figure 6-6: Photograph of the 700 liter/day water-heated, closed-air, open-water humidification-dehumidification desalination system with mass extraction and injection of the moist air stream described in this chapter.
Water condensed in the dehumidifiers from the moist air stream is collected in a condensate reservoir (not shown in the figures above). The air in the system is circulated using a GAST R2303A blower. The blower is driven using a 1/3 HP AC motor powered by 3 phase 240 V current. The speed of the AC motor is controlled using a frequency modulator in order to vary the mass flow rate of air at will. A TACO 0011 centrifugal pump is used to move the water around in the system. An inline globe valve was used to regulate the flowrate of water through the system. A 12.7 kW Rheem EcoSense tankless electric water heater is used for heating the water stream before sending it to the humidification device.

The flow rate of water and air are measured by rotameters. All inlet and outlet water and air temperatures were measured by standard K-type thermocouples. The uncertainties associated with these sensor measurements were describe earlier in Sec. 6.1.1. It has been calculated that the maximum total uncertainty on the performance parameter of interest, GOR, is ± 5%.

In the following section, we look at the effect of three important parameters on the performance (GOR) of HDH systems without mass extraction: (1) the mass flow rate ratio; (2) the top brine temperature (the temperature of the brine at the exit of the
heater); and (3) the feed water temperature. Further, the effect of mass extractions and injections is investigated.

6.2.1 Effect of mass flow rate ratio

Figure 6-9 illustrates the effect of the water-to-air mass flow rate ratio on the performance of the water heated CAOW HDH system (illustrated in Fig. 6-1) without mass extraction or injection. In this plot, GOR is normalized by the optimal GOR in all of the cases considered in the figure. This normalization is performed for all the data in the chapter. Absolute values of GOR are tabulated in Sec. 6.2.5. As was observed by several other researchers [35, 37, 46] and in our previous HDH modeling efforts [19, 22, 26], there is a particular value of mass flow rate ratio at which the HDH system performance is optimal.

The optimum mass flow rate ratio corresponds to a modified heat capacity ratio of unity for the dehumidifier (HCRd=1). This again is consistent with on-design (fixed component effectiveness) data formerly presented by us [19]. The maximum GOR thus occurs at a balanced condition for the dehumidifier which is the more irreversible component in the system for the given feed water temperature, top brine temperature, and cycle configuration.

6.2.2 Effect of top brine temperature

The top brine temperature (TBT) is an important parameter in the design of thermal desalination systems. One would expect that for a fixed size of the humidifier and the dehumidifier, a higher TBT will lead to a higher system efficiency. Figure 6-9 illustrates the same for HDH systems. This figure is plotted at a feed temperature of 25°C and at HCRd = 1.

It may be observed that the performance at a TBT of 60°C is only 55% of that at a TBT of 90°C. This is an important conclusion which can be only obtained from fixed hardware (off-design) data. When the components in HDH are modelled as fixed effectiveness (on-design), the size of the components increase/decrease with change in
Figure 6-8: Effect of mass flow rate ratio on the performance of the HDH system without mass extractions. Boundary conditions: $T_a = 20^\circ$C; $T_c = 80^\circ$C; $N = 0$; $V_h = 0.27 \text{ m}^3$; $A_d = 8 \text{ m}^2$.

Figure 6-9: Effect of top brine temperature on the performance of the HDH system without mass extractions. Boundary conditions: $T_a = 25^\circ$C; $HCR_d = 1$; $N = 0$; $V_h = 0.27 \text{ m}^3$; $A_d = 8 \text{ m}^2$. 
boundary conditions. For example, at a lower TBT a component effectiveness of 80% will need a much larger component than at a higher TBT. For this reason, on-design data at certain boundary conditions would suggest that at a lower TBT, we obtain a higher GOR [19].

In commercial systems, the TBT is limited depending on the constituents of the water being treated. The formation of hard scales on heat transfer equipment because of precipitation of scaling components limits the TBT in seawater systems to between 65-90°C.

6.2.3 Effect of feed water temperature

Figure 6-10 further illustrates the usefulness of the HCR concept. Here, we plot the GOR versus feed water temperature at fixed TBT and water-to-air mass flow rate ratio.

It may be observed that there is an optimum value for feed water temperature at which the performance is maximized. This corresponds to a heat capacity rate ratio of unity in the dehumidifier (HCRd=1). This further reinforces the idea that HCR=1 is the balanced condition for HME devices.

6.2.4 Effect of mass extraction and injection

The enthalpy pinch model for analysis of HME devices in HDH system was discussed in Chapter 5.1.2. In that work, algorithms for continuous and discreet extractions/injections in the HDH system with finite and infinite HME size were formulated. Several important conclusions based on that on-design study were made. It was found that a single mass extraction and injection of moist air was as efficient or better than infinite extractions above component enthalpy pinches of 7 kJ/kg_{dw}. In Sec. 6.1.2, we showed that the enthalpy pinch in our pilot unit is greater than the aforementioned value. Hence, a single extraction of air was made in the current pilot unit from the humidifier at a height in the humidifier which split the packing volume into 20% after extraction and 80% before extraction (the extraction was made closer to the...
Figure 6-10: Effect of feed water temperature on the performance of the HDH system without mass extractions. Boundary conditions: $T_c = 90^\circ C$; $m_r = 2.4$; $N = 0$; $V_h = 0.27 \text{ m}^3$; $A_d = 8 \text{ m}^2$. 
top of the humidifier dividing the packing volume in the top to bottom in the ratio 20:80). This split was chosen based on on-design modeling which suggested the optimal point of extraction in terms of CV volume location (using calculations presented in Chapter 5). The air was injected back in to the dehumidifier between the first and second heat exchanger (out of the 4 devices in total). This was designed to minimize temperature and concentration mismatch between the two air streams at the point of injection, in order to limit entropy produced due to mixing.

**Optimal amount of air extraction**

Figure 6-11 illustrates the effect of mass flow rate extracted on the increase in performance of the HDH system. The increase in performance of the HDH system is calculated as the ratio of the GOR with extraction to that without extraction. In the with and without extractions cases the top brine temperature, the feedwater temperature, the water flow rate and total air flow entering the humidifier from the dehumidifier (measured at state ‘f’ shown in Fig. 6-1) are held fixed. In the zero extraction case, this aforementioned ratio is 1 and increases with better balancing. The amount of air extracted is also normalized against total air flow.

It may be observed that the performance is optimal at a particular amount of extraction. In this particular case, where the top temperature is 90°C and the feed temperature is 25°C, the optimum amount of extraction is around 33%. The GOR is enhanced by up to 40%. The trends are similar at different boundary conditions and the maximum enhancement in GOR with a single extraction of air was found to be about 55%.

As would be expected, the maximum performance corresponded to the minimum average of local enthalpy pinches in the dehumidifier ($\overline{\psi}_{local,d}$). This is consistent with the principal purpose of thermodynamic balancing: to drive the process with a minimum driving force and at a correspondingly smaller entropy generation for a fixed system size.
Figure 6-11: Effect of mass flow rate of air extracted on the performance of the HDH system. Boundary conditions: $T_a = 25^\circ C; T_c = 90^\circ C; N = 1; V_h = 0.27 \text{ m}^3; A_d = 8 \text{ m}^2$. 

![Graph showing the effect of air extraction on performance](image-url)
Figure 6-12: Effect of top brine temperature on the performance of the HDH system with a single air extraction from the humidifier to the dehumidifier. Boundary conditions: $T_a = 25^\circ$C; HCR=1; $N = 1$; $V_h = 0.27 \text{ m}^3$; $A_d = 8 \text{ m}^2$.

**Effect of top brine temperature**

Figure 6-12 illustrates the effect of TBT on the enhancement of GOR due to a single extraction and injection in the HDH system.

It may be immediately observed that there exists an optimum mass of extraction in each case (corresponding to the case in which the average local enthalpy pinch is minimum). It may also be observed that the effect of the extraction is larger at a higher TBT. This was found to be consistently true at other values of feed temperature and packing volume. The results also make thermodynamic sense because at higher TBT the reversible efficiency of the cycle is higher. One would expect that the effect of balancing, which is to bring the system closer to reversibility, is higher when a higher reversible GOR is possible. This is an important design conclusion which means that it is desirable to design the extractions with the maximum TBT that is possible.
Table 6.1: Peak Performance of the pilot HDH system

<table>
<thead>
<tr>
<th>System/features</th>
<th>without extraction</th>
<th>with extraction</th>
<th>Units</th>
</tr>
</thead>
<tbody>
<tr>
<td>Humidifier packing volume</td>
<td>0.3</td>
<td></td>
<td>m³</td>
</tr>
<tr>
<td>Dehumidifier area</td>
<td>8</td>
<td></td>
<td>m²</td>
</tr>
<tr>
<td>TBT</td>
<td>90</td>
<td></td>
<td>°C</td>
</tr>
<tr>
<td>Feed water temperature</td>
<td>25</td>
<td></td>
<td>°C</td>
</tr>
<tr>
<td>GOR</td>
<td>2.6</td>
<td>4.0</td>
<td></td>
</tr>
<tr>
<td>Specific electricity consumption</td>
<td>0.4</td>
<td>0.45</td>
<td>kWhₑ/m³</td>
</tr>
<tr>
<td>Recovery ratio</td>
<td>8</td>
<td>10</td>
<td>%</td>
</tr>
<tr>
<td>Peak water production</td>
<td>0.7</td>
<td></td>
<td>m³/day</td>
</tr>
</tbody>
</table>

6.2.5 Peak Performance of the pilot HDH system

The peak performance of the HDH system described in this publication is shown in Table 6.1. It can be seen that the HDH system with a single extraction has a GOR of 4.0 and low electrical energy consumption of 0.4 kWhₑ/m³. The system also has a peak water production of 700 liters/day and a recovery ratio of 10%. Brine recirculation can be incorporated to increase the recovery to very high values (if desired).

It is also noted that for the enthalpy pinches achieved at peak performance (Ψₗ ≈ Ψᵅ ≈ 19 kJ/kgₐₗ), the GOR of the system calculated previously based on detailed numerical simulations was 2.75 without extraction and 4.5 with a single extraction an injection [23]. This is within about 10% of the values we obtain here experimentally. Given the fact that the experiment unit had some heat losses to ambient (about 5-8%) and that the entropy production associated with mixing at the point of injection of air in the dehumidifier was non-zero, the aforementioned small deviation in GOR from the numerical prediction is expected.

6.3 Chapter conclusions

New experimental data from a pilot HDH unit is presented in this chapter. This data is reconciled with the on-design data presented by us previously. The following conclusions are drawn based on this study:
1. HCR=1 (i.e. the point at which the maximum change in enthalpy rates of either stream exchanging energy is equal) represents a thermally balanced state for a simultaneous heat and mass exchange device.

2. For a water heated CAOW HDH system without mass extractions, the $HCR_d=1$ represents the state at which the GOR is maximized.

3. HDH systems without mass extractions need to be operated at as high a top brine temperature as is possible in order to ensure a high GOR.

4. Mass extractions from the humidifier to the dehumidifier increase the GOR of the water heated CAOW HDH system by up to 55%.

5. The optimum extraction mass flow rate corresponds to the case in which a minimum average local enthalpy pinch is achieved in the device.

6. At a higher top brine temperature, the enhancement in GOR due to mass extractions is higher.

Finally, the pilot system built had a maximum GOR of 4.0±0.2 and is currently the state of the art in HDH systems in terms of energy efficiency when compared to systems reported in literature (see Chapter 2).
Chapter 7

Mechanical compression driven varied pressure HDH system

All existing HDH systems operate at a single pressure (normally at atmospheric pressure) and consist of three subsystems: (a) an air and/or water heater; (b) a humidifier or an evaporator; and (c) a dehumidifier or a condenser. These are simple systems and are relatively easy to design and fabricate. However, the thermal performance of these single pressure systems is very limited (a maximum Gained Output Ratio or GOR of about 4). This is because the single pressure HDH system has three intrinsic disadvantages from a thermal performance perspective: (1) low water vapor content in air (low humidity ratio) at atmospheric pressure; (2) extra thermal resistance to heat transfer because of the presence of the carrier gas (air) in the condenser; and (3) lower energy recovery compared to MSF and MED systems. The third point is especially important because, unlike MSF and MED systems, multi-staging the HDH system does not yield any increase in performance (Chapter 4). In this chapter, simple means to address the aforementioned demerits of the HDH system using the tools of classical thermodynamics are described.

Effect of operating pressure on the humidity ratio of moist air

All previous HDH systems in literature have been designed to operate at atmospheric pressure. However, to increase the vapor content of moist air the systems need to
be operated at sub-atmospheric pressures. Figure 7-1 illustrates this concept in a psychrometric chart. For example, at a dry bulb temperature of 65 °C the humidity ratio of moist air is increased two fold when the operating pressure is reduced from 100 kPa to 50 kPa.

![Psychrometric Chart](image)

**Figure 7-1:** Effect of pressure on humidity ratio of saturated moist air.

However, if the entire HDH system is operated under this reduced pressure, the increase in thermal performance is relatively low. This is because: (1) the energy recovery is limited (for the same reasons as for the atmospheric pressure systems); and (2) the humidity ratio at the dehumidifier exit is also increased, limiting the water productivity [19].

### 7.1 Variable pressure HDH cycle

In this chapter, a new HDH cycle to improve the energy efficiency of HDH is described. The proposed cycle operates the humidifier and dehumidifier at different pressures.
As shown in Fig. 7-2, the pressure differential is maintained using a compressor and an expander. The humidified carrier gas leaving the humidification chamber is compressed in a mechanical compressor and then dehumidified in the condenser or the dehumidifier. The dehumidified carrier gas is then expanded to recover energy in form of a work transfer. The expanded carrier gas is then send to the humidification chamber. The carrier gas is thus operated in a closed loop. The feed seawater is preheated in the dehumidifier before it is sent to the humidification chamber thus recovering some of the work input to the compressor in form of thermal energy which is given back to the carrier gas stream during the humidification process. The brine from the humidification chamber is then disposed.

Figure 7-2: Schematic diagram of mechanical compression driven HDH system

Figure E.2 illustrates an example of the cycle on a psychrometric chart. 1-2 is the air humidification process that is approximated to following the saturation line. 2-3 is the compression process in which the humidified air is compressed to a higher pressure and temperature. 3-4 is the dehumidification process. The state 4 is assumed to be saturated in this example. 4-1 is the air expansion process where some of the energy input in the compressor is recovered.
Figure 7-3: Mechanical compression driven HDH cycle represented in psychrometric coordinates.

7.2 Terminology used

In this section, the terminology used in the analysis is defined. This includes an energy-based effectiveness, an isentropic efficiency for the compressor and expander, a modified heat capacity rate ratio for the heat and mass exchange devices, and the system performance parameters. Energy-effectiveness and HCR have been defined before in Chapter 3.

7.2.1 Isentropic efficiency

The performance of the compressor and expander are defined by an isentropic efficiency. For a mechanical compressor, the isentropic efficiency is defined as the ratio of the reversible to actual work input.

$$\eta_{com} = \frac{W_{rev}}{W} \quad (7.1)$$
For an expander, the isentropic efficiency is defined as the ratio of the actual to reversible work output.

\[ \eta_e = \frac{\dot{W}}{\dot{W}_{\text{rev}}} \]  

(7.2)

### 7.2.2 System and performance parameters

As a first step for understanding the improved performance of the new HDH cycles the following system and performance parameters are defined.

1. **Specific work consumption (SW):** is the amount of electrical energy (in kJₑ) consumed to produce one kg of fresh water. This parameter is used commonly for defining the performance of work driven desalination systems.

\[ \text{SW} = \frac{\dot{W}_{\text{in}} - \dot{W}_{\text{out}}}{\dot{m}_{\text{pw}}} \]  

(7.3)

The specific work consumption can be rewritten as follows

\[
\text{SW} = \frac{\dot{W}_{\text{in}} - \dot{W}_{\text{out}}}{\dot{m}_{\text{pw}}} = \left( \frac{\dot{m}_{\text{da}} \cdot \omega_{H,o}}{\dot{m}_{\text{pw}}} \right) \left( \frac{\dot{m}_{\text{pw}}}{1/\text{VPR}} \right)
\]  

(7.4)

Thus, SW is a function of two new system parameters - vapor productivity ratio (VPR) and specific net work (SNW).

2. **Vapour productivity ratio (VPR):** is defined as the ratio of the rate at which water is produced by the system to the rate at which water vapor is compressed in the system.

\[ \text{VPR} = \frac{\dot{m}_{\text{pw}}}{\dot{m}_{\text{da}} \cdot \omega_{H,o}} \]  

(7.5)

VPR is a measure of how effective the humidifier and dehumidifier are at producing water given a fixed compression ratio, and expander and compressor
efficiency. The value of VPR will always be less than 1, as water cannot be produced at a rate greater than that at which it flows into the dehumidifier. For example if the vapor productivity ratio is 0.25, this means for every four units of vapor that are compressed in the system, only one unit of water is produced. Evidently, VPR should be maximised to avoid water vapor from being needlessly compressed.

3. Specific net work (SNW) : is the net work input to the system per unit amount of vapor compressed.

\[
SNW = \frac{\dot{W}_{in} - \dot{W}_{out}}{\dot{m}_{da} \cdot \omega_{H,o}} \tag{7.6}
\]

In the mechanical compression driven HDH system, compression of the carrier gas is an energetic loss which is only partially recovered as work in the expander and as heat in the dehumidifier. SNW is indicative of the work imparted to the useful component of the fluid mixture circulating in the system.

## 7.3 Equations and modeling details

This section discusses the conservation equations for the expander and compressor are provided below.

### 7.3.1 Compressor

Consider a mechanical compressor which provides the driving pressure difference to the moist air stream by means of a work transfer \(\dot{W}_{in}\). The First Law for the compressor can be expressed as

\[
\dot{W}_{in} = \dot{m}_{da} (h_{a,o} - h_{a,i}) \tag{7.7}
\]
The isentropic efficiency for the compressor can be defined as:

\[ \eta_{com} = \frac{h_{a, o}^{rev} - h_{a, i}}{h_{a, o} - h_{a, i}} \]  

where the exit state from the compressor (which is at the dehumidifier inlet pressure) is calculated using the Second Law for the reversible case.

\[ s_{a, o}^{rev} = s_{a, i} \]  

7.3.2 Expander

The First Law for the expander can be expressed as

\[ \dot{W}_{out} = \dot{m}_{da} (h_{a, i} - h_{a, o}) - (\dot{m}_w \cdot h_w)_{\text{condensate}} \]  

The isentropic efficiency for the expander can be defined as:

\[ \eta_e = \frac{\dot{m}_{da} (h_{a, i} - h_{a, o}) - (\dot{m}_w \cdot h_w)_{\text{condensed}}}{\dot{m}_{da} (h_{a, i} - h_{a, o}^{rev}) - (\dot{m}_w \cdot h_w)^{rev}} \]  

where the exit state from the expander (which is at the humidifier inlet pressure) in the reversible case is calculated using the Second Law.

\[ \dot{m}_{da} (s_{a, i} - s_{a, o}^{rev}) - (\dot{m}_w \cdot s_w)^{rev} = 0 \]  

Solution technique is same as that presented in Chapter 3.1.3

7.4 Results and discussions

7.4.1 Parametric study

This section investigates the importance of various parameters on the overall performance of the variable pressure cycle driven by a mechanical compressor. Under-
standing the effect of these parameters is necessary to optimize the design of the cycle. The parameters studied include the mass flow rate of the air and water streams, the expander and compressor efficiencies, the humidifier and dehumidifier effectivenesses, the operating humidifier pressure, the air side pressure drops in the dehumidifier and humidifier, and the pressure ratio provided by the compressor.

**Optimum Second Law performance.** It was previously shown (Chapter 4) that the performance of the HDH cycle depends on the mass flow rate ratio (ratio of mass flow rate of seawater at the inlet of the humidifier to the mass flow rate of dry air through the humidifier), rather than on individual mass flow rates. Moreover, we have also shown that there is an optimum performance at fixed input conditions and this occurs at a modified heat capacity rate ratio of unity (HCR=1) for either the humidifier or the dehumidifier. For mechanical compression driven HDH, the Second Law optimum occurs at a balanced condition for the humidifier. An example of this result is shown in Fig. 7-4.

Hence, in this and all the subsequent sections only the optimum performance values are reported.

**Effect of component performance** ($\eta_{\text{com}}, \eta_e, \epsilon_H, \epsilon_D$). Figure 7-5 illustrates the variation in performance of the cycle at various values of isentropic efficiencies and HMX effectivenesses. In this figure, one of the effectivenesses or efficiencies is varied at a time while the others are fixed. The dehumidifier and humidifier effectiveness is fixed at 80% and the isentropic efficiencies are fixed at 100% except in the cases in which they are varied. The air side and water side pressure drop is assumed to be zero in both the humidifier and the dehumidifier, and seawater is assumed to enter the system at 30°C. The pressure ratio was fixed at 1.2.

It is observed that while a higher efficiency compressor and expander are vital for a low specific work consumption, the compressor efficiency is of greater relative importance. This general trend has also been observed for various other boundary conditions. It is important to note that, relatively, the performance of the cycle is
Figure 7-4: Effect of modified heat capacity ratio of humidifier on specific work and specific entropy generation. $T_{w, in} = 30^\circ$C; $\varepsilon_H = \varepsilon_D = 80\%$; $\eta_{com} = \eta_e = 100\%$; $P_H = 40$ kPa; $P_D = 48$ kPa.

Figure 7-5: Effect of component efficiency or effectiveness on cycle performance for $T_{w, in} = 30^\circ$C; $P_H = 33.33$ kPa; $P_D = 40$ kPa.
less sensitive to the humidifier and dehumidifier performance.

**Effect of pressure ratio** ($P_D/P_H$) **and dehumidifier pressure** ($P_d$). Figure 7-6 illustrates the effect of pressure ratio and humidifier pressure on cycle performance. Firstly, at a lower pressure ratio, the specific work is lower (indicating a higher system performance). The lower limit on pressure ratio required in the compressor is imposed by the dehumidifier minimum terminal temperature difference. For the present simulations the pressure ratio was varied from 1.2 to 2.4.

The reason for lower SW at lower pressure ratios can be explained using Fig. 7-7. At lower design pressure ratios, the vapor productivity ratio is lower. As already explained in Section 8.1.1, this is an expected trend. SNW increases with increasing pressure ratio and the slope with which the SNW increases is much greater than that for the increase in VPR. Specific work is the ratio of SNW to VPR (See Eqn. 7.4); and hence, at lower pressure ratios, we get a higher performance. In Fig. 7-6 it can also be observed that a lower dehumidifier pressure gives a lower specific work. This is explained using the variation of SNW and VPR with dehumidifier pressure as shown in Fig. 7-8. Both SNW and VPR increase with increase in design dehumidifier pressure. VPR increases slowly compared to SNW and hence the specific work consumption decreases with lower dehumidifier pressures.

**Effect of air side pressure drop** ($\Delta P_H, \Delta P_D$). The air side pressure drop can be substantial in heat and mass exchange (HME) devices if the design is not performed to optimize it. Figures 7-9 and 7-10 illustrate the effect of pressure drop of the air stream through the HME devices on the overall performance of the system. As the pressure drop increases, the specific work consumption increases rather drastically. Hence, it is vital to design the HME devices such that the pressure drop is minimal.

At higher values of pressure drop there is an optimum pressure ratio at which the specific work is minimum. The pressure drop in the dehumidifier and humidifier increase the specific work by a similar amount.
Figure 7-6: Effect of pressure ratio and dehumidifier pressure on cycle performance for $T_{sw,in} = 30^\circ$C; $\varepsilon_H = \varepsilon_D = 80\%$; $\eta_{com} = \eta_e = 100\%$.

Figure 7-7: The effect of pressure ratio on specific net work and vapor productivity ratio to explain the trend in Fig. 7-6.
Figure 7-8: The effect of dehumidifier pressure on specific net work and vapor productivity ratio to explain the trend in Fig. 7-6.

Figure 7-9: Effect of air-side pressure drop in the humidifier on cycle performance for $T_{\text{air, in}} = 30^\circ$C; $\varepsilon_H = \varepsilon_D = 80\%$; $\eta_{\text{com}} = \eta_e = 100\%$; $P_D = 40$ kPa.
7.4.2 Selection of expansion device.

This section investigates the use of a throttle in place of a mechanical expansion device in the variable pressure system, downstream of the dehumidifier. Here, the throttle is modeled as an isenthalpic device. Figure 7-11 illustrates the performance loss because of using a throttle. It is clearly observed that, when using a throttle, the cycle has very high specific work consumption.

The reason for the low performance is shown in Fig. 7-12. This figure illustrates the entropy generation in each of the devices for certain boundary conditions. It can be immediately observed that for the cycle with the throttle, the entropy generation is very high because the process in the throttle is highly irreversible. We have previously proved [70] that the performance is inversely proportional to total entropy generated in the system. Hence, the irreversibility in the throttling process causes the system performance to drop significantly.
Figure 7-11: Effect of using a throttle versus using an air expander in the two pressure cycle for $T_{sw,in} = 30^\circ$C; $\varepsilon_H = \varepsilon_D = 80\%$; $\eta_{com} = 100\%$; $\eta_e = 0$ or 100\%; $P_D = 40$ kPa.

Figure 7-12: Entropy generation in the throttle and the air expander cycles for $T_{sw,in} = 30^\circ$C; $\varepsilon_H = \varepsilon_D = \eta_{com} = \eta_e = 90\%$; $P_H = 40$ kPa; $P_D = 50$ kPa.
7.5 Comparison with other HDH cycles

In Table 7.1, the mechanical compression driven HDH systems are compared to existing designs, including air heated and water heated HDH systems. A power production efficiency ($\eta_{PP}$) of 40% is used to convert the work consumed to heat, and the comparison is done based on a modified GOR defined as follows:\textsuperscript{1}:

$$\text{GOR} = \frac{m_{pu} \cdot h_{fg}}{W_{in} - W_{out} + \dot{Q}_{in}}$$  \hspace{1cm} (7.13)

These values were calculated for a minimum terminal temperature difference of 5 K in dehumidifier and 3 K in humidifier. It is observed that the new cycle has a much higher energy efficiency than existing HDH systems. Collaborators in King Fahd University of Petroleum and Minerals (KFUPM) have been working on a thermal energy driven mechanical compression HDH cycle and it has been found that the GOR of that system is around 6 to 9 \textsuperscript{103}. An embodiment of that cycle is shown in Fig. 7-13. This shows the promise of the varied pressure design and validates the $\eta_{PP}$ based conversion to GOR in this section. It is also noted that thermodynamic balancing can be incorporated in this novel cycle by mass extraction and injection of water to increase the efficiency further.

Table 7.1: Comparison of mechanical compression HDH with other HDH desalination technologies

<table>
<thead>
<tr>
<th>Technologies</th>
<th>GOR</th>
</tr>
</thead>
<tbody>
<tr>
<td>Water heated HDH</td>
<td>2.5</td>
</tr>
<tr>
<td>Water heated HDH with thermodynamic balancing</td>
<td>4</td>
</tr>
<tr>
<td>Mechanical energy driven mechanical compression driven HDH</td>
<td>6 to 8</td>
</tr>
</tbody>
</table>

\textsuperscript{1}It should be noted that $\dot{Q}_{in}$ in Eqn. 7.13 is typically at a low temperature compared to the high temperature $\dot{Q}$ used in $W_{in}$ implied by $\eta_{PP}$.  

183
7.6 Chapter conclusions

1. A novel desalination cycle based on a variable pressure humidification dehumidification concept has been described in this manuscript. Various features of this cycle have been discussed in detail.

2. A parametric study explaining the influence of various system and component variables on system performance is described. It has been found that important design parameters include the expander and compressor efficiencies, air side pressure drops in the humidifier and the dehumidifier, and the pressure ratio provided by the compressor.

3. The possibility of using a throttle instead of a mechanical expander was examined and it was found that the cycle with the throttle has a much higher energy
requirement because of high irreversibility in the throttling process.

4. The thermal energy driven mechanical compression HDH cycle has much higher performance than existing HDH cycles.

Based on the details provided in this chapter, pilot scale experimental units are under construction at KFUPM in Saudi Arabia.
Chapter 8

Thermal compression driven hybrid HDH-RO system

Increasing the top brine temperature (and correspondingly the temperature of the steam used as heat source) in a thermal desalination system can decrease the amount of heat required to drive the process. In existing thermal desalination systems like MSF and MED, a practical limit on the top brine temperature is associated with the formation of calcium sulphate (hard) scales. For typical seawater concentrations and recovery ratios when brine temperatures exceed 90-110°C (the corresponding steam temperature being 100-120°C), the solubility of calcium is such that it precipitates as calcium sulphate. This forms hard scales on heat exchanger surfaces impeding heat transfer. Hence, in MED and MSF the temperature of the heating steam is limited to 100°C to 120°C. Carrier gas based thermal desalination systems can potentially operate well above this limit. This paper describes the features of such a novel system. The conceptual details of this system are discussed later in this section. MED-TVC also operates with higher temperature steam but for operational considerations (to reduce the specific volume of the steam drawn from the Rankine cycle) rather than to reduce specific energy consumption. The details of the effect of increasing the heating steam temperature on the energy consumption of MED-TVC systems have been documented by Kamali et al. [104].

First, let us quantify the effect of increasing the heating steam temperature by
considering a simple black box model (shown in Fig. 8-1) of a thermal desalination system. This system is steam driven (like MSF and MED). The brine and pure water are assumed for simplicity to leave the system at the same temperature as the inlet feed seawater. Heat is lost from the system at this same temperature. Also, the steam enters in a fully dry condition \((x = 1)\) and leaves as a liquid at the inlet feed seawater temperature. The expression for the thermal energy required to run the black box system is derived below using the First and Second Law of Thermodynamics.

First Law on the black box gives

\[
(mh)_{st,in} + m_1 h_1 = (mh)_{st, out} + m_2 h_2 + m_3 h_3 + Q_{out}
\]  
(8.1)

Second Law on the black box gives

\[
(ms)_{st,in} + m_1 s_1 + \dot{S}_{gen} = (ms)_{st, out} + m_2 s_2 + m_3 s_3 + \frac{Q_{out}}{T_0}
\]  
(8.2)

Further we combine Eqs. 8.1 and 8.2 to give

\[
(mh)_{st,in} - (mh)_{st, out} = (ms)_{st,in} \cdot T_0 - (ms)_{st, out} \cdot T_0 + m_2 g_2 + m_3 g_3 - m_1 g_1 + \dot{S}_{gen} \cdot T_0
\]  
(8.3)

Figure 8-1: Black box model for steam driven thermal desalination system.
Here, \( g (= h - T_s) \) is the specific gibbs energy. Now we can write the change in total enthalpy rate of the steam, which is the thermal energy consumed by the system, as follows

\[
\Delta \dot{H}_{in} = (\dot{m}h)_{st,in} - (\dot{m}h)_{st,\text{out}} \\
= (\dot{S}_{in} - \dot{S}_{out})_{st} \cdot T_0 + \dot{m}_2g_2 + \dot{m}_3g_3 - \dot{m}_1g_1 \\
+ \dot{S}_{gen} \cdot T_0 \\
\left\{ \frac{\Delta \dot{H}_{in}}{\dot{m}_3} \right\} = \left\{ \frac{\dot{S}_{in,\text{st}} - \dot{S}_{out,\text{st}}}{\dot{m}_3} \right\} \cdot T_0 + \frac{1 - RR}{RR}g_2 + g_3 - \frac{1}{RR}g_1 \\
+ \frac{\dot{S}_{gen}}{\dot{m}_3} \cdot T_0 \tag{8.4}
\]

\( RR = \frac{\dot{m}_3}{\dot{m}_1} \) is the recovery ratio of the system.

From Eq. 8.5, it can be seen that the specific thermal energy required to drive a steam driven desalination system is directly proportional to the total entropy rate of steam used per unit amount of water distilled in the system. Increasing the steam temperature to the system reduces the aforementioned entropy rate. This is because of the fact that increasing the steam temperature reduces the specific entropy and increases the specific enthalpy. The increase in specific enthalpy, in turn, reduces the mass flow rate of steam required. Hence, by increasing the temperature both the mass flow rate of steam required and the specific entropy are reduced, reducing the total entropy rate of steam entering the system. This is expected to reduce the thermal energy consumed by the system. Let us quantify this effect by evaluating the least amount of thermal energy required, which is calculated assuming there is no entropy generation in the system.

\[
\left\{ \frac{\Delta \dot{H}_{\text{in,least}}}{\dot{m}_3} \right\} = \left\{ \frac{\dot{S}_{in,\text{st}} - \dot{S}_{out,\text{st}}}{\dot{m}_3} \right\} \cdot T_0 + \frac{1 - RR}{RR}g_2 + g_3 - \frac{1}{RR}g_1 \tag{8.6}
\]

Fig. 8-2 shows the effect of increasing the heating steam temperature (and correspondingly reducing the total entropy rate of the steam entering the system) on the least thermal energy required to produce 1 kg/s of water in the system. It is
noted here that for the following calculation second and the third term in the right hand side of Eq. 8.6 is constant. For this illustration, seawater feed is at a salinity of 35,000 ppm and temperature of 30°C. Seawater properties are evaluated using the correlations presented by Sharqawy et al. [100]. The curves are plotted at a recovery ratio (RR) of 50%. It is observed that by increasing the steam temperature from 90°C to 120°C (and correspondingly reducing $\dot{S}_{in}$ by 31.6%) the least thermal energy required is reduced by 27%. If the steam temperature can be further increased to 200°C (correspondingly reducing $\dot{S}_{in}$ by 59%) the least thermal energy required is reduced by 50%. These trends are similar to those presented for steam driven MED-TVC systems by several researchers including Kamali et al. [104].

![Graph](image)

Figure 8-2: Effect of the heating steam temperature (and correspondingly the total entropy rate of steam) entering a thermal desalination system (shown in Fig. 8-1) on the least thermal energy required to drive the system. $T_0 = 30°C$; $x_{st,in} = 1$; $T_{st,out} = 30°C$; $\dot{m}_g = 1$ kg/s; $S_1 = 35,000$ ppm; RR = 50%.
Hence, we conclude that fundamental to decreasing the specific energy consumption of a steam driven desalination system is to increase the temperature of heating steam and to reduce the total entropy rate of steam used per unit water produced in the system. The focus of this manuscript is a new desalination system which can run using steam at a high temperature without the problems of scale formation.

In this proposed system we operate the humidifier and dehumidifier at different pressures. As shown in Fig. 8-3, the pressure differential is maintained using a thermal vapor compressor (TVC) and an expander. Conceptually, this is similar to the variable pressure HDH system using a mechanical compressor that we described in the previous chapter except for the use of a TVC instead of a mechanical compressor. The humidified carrier gas leaving the humidification chamber is compressed in a TVC using a steam supply and then dehumidified in the condenser (or the dehumidifier). The dehumidified carrier gas is then expanded to recover energy in form of a work. The recovered work is used in a reverse osmosis unit to desalinate the brine from the humidifier. The expanded carrier gas is send to the humidification chamber. The carrier gas is thus operated in a closed loop. The feed seawater is preheated in the dehumidifier before it is sent to the humidification chamber thus recovering some of the energy input to the compressor in form of thermal energy which is given back to the carrier gas stream during the humidification process. As shown in Fig. 8-3, both the thermo-compression and the expansion process can lead to condensation of a small amount of water out of the carrier gas.

Figure 8-4 illustrates an example of the new desalination cycle on a psychrometric chart. Path 1-2 is the air humidification process that is approximated to follow the saturation line. Path 2-3 is the thermo-compression process in which the humidified air is compressed to a higher pressure and temperature. Path 3-4 is the dehumidification process which is also approximated to follow the saturation line at a higher pressure, $P_D$. Path 4-1 is the air expansion process where some of the energy input in the compressor is recovered.

In this system, we can use higher pressure and temperature steam ($T_{sat} > 120^\circ C$) without attaining a high top brine temperature. This is possible because the heating
Figure 8-3: Schematic diagram of thermal vapor compression driven HDH-RO system.

Figure 8-4: Psychrometric representation of thermal vapor compression driven HDH system.

192
steam is not brought in direct contact with the seawater, it is instead brought in contact with the vapor laden carrier gas. It is also made sure that the pressure ratio attained in the TVC is such that the moist air which exchanges heat with the brine in the dehumidifier is at a relatively low temperature. Thus, this new system can be designed such that the brine temperature does not exceed 60°C. For standard seawater concentrations, this is sufficient to avoid scale formation.

8.1 Reversible entrainment efficiency of the TVC

To define the performance of the TVC we use the entrainment efficiency. This is the ratio of the entrainment ratio (which is the ratio of the mass of carrier gas compressed per unit mass of steam given to the TVC) in an actual TVC to that in a TVC with zero entropy generation.

\[
\eta_{tvc} = \frac{ER}{ER_{rev}}
\]

(8.7)

\[
ER = \frac{\dot{m}_{a,i}}{\dot{m}_{st}}
\]

(8.8)

The reversible terms in the above equations refer to values taken during an ideal process where motive and entrained fluids at known thermodynamic states are reversibly and adiabatically brought to equilibrium at a defined discharge pressure. The features of this efficiency have been dealt with in detail in a separate publication [58].

8.1.1 System and performance parameters

As a first step for understanding the improved performance of the new HDH cycle, the following system and performance parameters are defined.

1. Gained-Output-Ratio (GOR): has been defined in Chapter 2. We here expand the definition for the steam driven cycle discussed in this chapter.
The net heat rate input to the TVC driven HDH system is given as

\[ \dot{Q}_{in} = \dot{m}_{st} \cdot (h_{st,in} - h_{st,out}) \]  

(8.9)

For the HDH system, \(h_{st,out}\) is a fully liquid state evaluated at the exit of the dehumidifier. Hence, Eq. 2.2 can be rewritten as follows

\[
\text{GOR} = \frac{\dot{m}_{pw} \cdot h_{fg}}{\dot{m}_{st} \cdot (h_{st,in} - h_{st,out})} \\
= \left\{ \frac{\dot{m}_{pw}}{\dot{m}_{da} \cdot \omega_{H,o}} \right\} \cdot \left\{ \frac{\dot{m}_{da} \cdot \omega_{H,o}}{\dot{m}_{st}} \right\} \cdot \left\{ \frac{h_{fg}}{h_{st,in} - h_{st,out}} \right\} \approx 1 \\
= \text{VPR} \cdot \text{ER}_{vap} 
\]

(8.10)

Thus, GOR is approximately equal to the product of two new system parameters - vapor productivity ratio (VPR), and vapor entrainment ratio (ER\(_{vap}\)). It is noted that the ratio of the latent heat of vaporization to the difference in specific enthalpy of steam between the inlet and exit of the system varies between 0.92 to 0.98 for the conditions considered in this paper. Hence, approximating this term as 1 for the sake of explaining some of the trends in the paper is reasonable. It should be noted that this approximation is not used to calculate GOR in this paper.

2. Equivalent electricity consumption (\(\dot{E}_c\)): is defined in terms of the amount of electricity that could have been produced using the thermal energy which was given to a desalination system. The electrical work is calculated assuming that the steam used in the desalination plant was instead expanded in a steam turbine.

\[
\dot{W}_{el} = \dot{m}_{st} \cdot (h_{st,tur,in} - h_{st,tur,out}) \cdot \eta_{gen} \quad \text{[kW]} 
\]

(8.12)

Where \(\eta_{gen}\) is efficiency of the electrical generator and is assumed to be 95%. In order to calculate \(h_{st,tur,out}\), the outlet steam temperature is taken to be the steam turbine exit temperature which is commonly employed in Rankine cycle
power plants (35°C). Also, an isentropic efficiency of 85% is assumed for the steam turbine.

\[
\hat{E}_c = \frac{\dot{W}_{st}}{\dot{m}_{pw} \cdot 3.6} \quad \text{[kWh/m}^3]\tag{8.13}
\]

The equivalent electricity consumption parameter facilitates comparison between thermal desalination systems and work driven desalination systems. For the latter, the actual work input is used in Eq. 8.13 instead of \(\dot{W}_{st}\).

3. Vapor productivity ratio (VPR): is defined as the ratio of the rate at which water is produced by the system to the rate at which water vapor is compressed in the system. This parameter is indicative of the efficiency of all the components in the system excluding the TVC. For the HDH-TVC-RO system, the value of this parameter will be greater than 1. A VPR of 12 means that for every kg of water vapor compressed in the TVC, 12 kg of water is produced.

\[
\text{VPR} = \frac{\dot{m}_{pw,D} + \dot{m}_{pw,RO}}{\dot{m}_{da} \cdot \omega_{H,o}} \tag{8.14}
\]

4. Vapor Entrainment Ratio (ER\text{vap}): is the amount of water vapor entrained by the TVC per unit amount of motive steam supplied.

\[
\text{ER}_{vap} = \frac{\dot{m}_{da} \cdot \omega_{H,o}}{\dot{m}_{st}} \tag{8.15}
\]

The energy input in compressing vapor in the carrier gas mixture is the useful part of the total energy input to the compressor. We are interested in the quantity of vapor entrained per unit steam supplied rather than the quantity of mixture entrained per unit steam supplied.

8.2 Equations and modeling details

This section discusses the conservation equations for the TVC and the RO unit. The conservation equations for the other devices in the system and the fluid property
packages and models used to solve the defined equations are already dealt with in earlier chapters.

Consider a TVC in which a mixture of air and water vapor is compressed from a low pressure $P_H$ to a higher pressure $P_D$, using a steam supply. Since all the dry air entering the TVC also leaves the device, the mass flow rate of dry air is constant. During the compression process, some of the water vapor in the mixture might condense out. The mass flow rate of the condensed water may be calculated by using a mass balance:

$$\dot{m}_{pw,tvc} = \dot{m}_{da}(\omega_{tvc,i} - \omega_{tvc,o}) + \dot{m}_{st}$$  (8.16)

The efficiency of the TVC defined in Eq. 8.7 may be written as

$$\eta_{tvc} = \frac{\dot{m}^{rev}_{st}}{\dot{m}_{st}}$$  (8.17)

Here $\dot{m}^{rev}_{st}$ is calculated from the following conservation equations for the reversible case:

$$\dot{m}^{rev}_{st} \cdot h_{st,tvc,in} = \dot{m}_{da}(h^{rev}_{a,tvc,o} - h_{a,tvc,i}) + \dot{m}_{puw,tvc} \cdot h_{puw,tvc}$$  (8.18)

$$\dot{m}^{rev}_{st} \cdot s_{st,tvc,in} = \dot{m}_{da}(s^{rev}_{a,tvc,o} - s_{a,tvc,i}) + \dot{m}_{puw,tvc} \cdot s_{puw,tvc}$$  (8.19)

For the reversible case, the discharged (dehumidifier inlet) pressure ($P_D$) is the same as the real case. The First and Second Law for the TVC can be expressed as

$$\dot{m}_{st} \cdot h_{st,tvc,in} = \dot{m}_{da}(h_{a,tvc,o} - h_{a,tvc,i}) + \dot{m}_{puw,tvc} \cdot h_{puw,tvc}$$  (8.20)

$$\dot{S}_{gen,tvc} = \dot{m}_{da}(s_{a,tvc,o} - s_{a,tvc,i}) - \dot{m}_{st} \cdot s_{st,tvc,in} + \dot{m}_{puw,tvc} \cdot s_{puw,tvc}$$  (8.21)

The work recovered in the expander $W_{out}$ produces further fresh water from the brine exiting the humidifier by operating a reverse osmosis (RO) unit for desalination of the brine. RO is taken to have an energy consumption of 3.5 kWh/m$^3$ at a recovery of 50%. These performance value are representative of RO systems desalinating...
seawater from a medium to large scale [105].

\[ \dot{n}_{pw,RO} = \frac{\dot{W}_{out}}{\dot{E}_{c,RO} \cdot 3.6} \]  

(8.22)

### 8.3 A typical embodiment of the novel cycle

Figure 8-5 shows a typical embodiment of the TVC driven HDH-RO cycle. The component performances are selected such that they are easily available off-the-shelf. An RO system performance of 3.5 kWh/m³ was used for the simulation. A feed seawater temperature of 30°C and a pressure ratio of 1.15 were used. Saturated steam at 50 bar was used for compressing moist air in the TVC.

Figure 8-5: A typical embodiment of the TVC driven HDH-RO cycle.

The HDH section of the system has been designed for a 1 kg/s water production.
The air and water mass flow rates were calculated for a modified heat capacity rate ratio (HCR) of 1 for the humidifier (details about the relevance of HCR of heat and mass exchangers to the performance of HDH cycles are described in previous chapters (Chapter 7 and 3). An overall system GOR of 9.46 and equivalent total electricity consumption of 22 kWh/m³ (from Eq. 8.12 & 8.13) are achieved in this embodiment.

8.4 Results and discussions

The various features of the novel TVC driven HDH cycle with RO are discussed in this section. The importance of various parameters on the overall performance of the cycle is investigated. Understanding the effect of these parameters is necessary to optimize the design of the cycle. The parameters studied include the motive steam pressure, the pressure ratio provided by the TVC, the expander efficiency, the operating humidifier pressure, and the air side pressure drops in the dehumidifier and humidifier. All calculations presented in this section are performed for a modified heat capacity ratio for the humidifier of unity. The typical embodiment of the cycle described in the previous section was used as a base case from which parameters were varied.

8.4.1 Effect of using an expander and a RO unit for additional water production

In a separate publication [106], we described the characteristics of an HDH system run using a TVC and a separate throttle valve (instead of an expander, as described in this paper). It had been found that the entropy generated in the throttle valve negates the advantage of using higher pressure steam. In other words, the \( \frac{\dot{S}_{\text{gen}}}{\dot{m}_{\text{pw}}} \) term in Eq. 8.5 is so high that the reduction in \( \frac{\dot{S}_{\text{in}}}{\dot{m}_{\text{pw}}} \) does not reduce specific energy consumption. Here, we improve on that cycle by replacing the isenthalpic throttling process by an expansion process which is close to an isentropic process. We use the work recovered by the expansion process in a reverse osmosis unit to produce
additional pure water from the brine exiting the humidifier. To quantify the effect of this change let us define a specific entropy generation in the expander \( I \), which characterises the specific irreversibility in the expansion process, as the ratio of the entropy generated in the expander to the total water produced in the system.

\[
I = \frac{\dot{S}_{\text{gen},e}}{\dot{m}_{\text{pw},D} + \dot{m}_{\text{pw},RO} + \dot{m}_{\text{pw},\text{HC}} + \dot{m}_{\text{pw},e} \to 0} \tag{8.23}
\]

Figure 8-6 illustrates the effect of having a less irreversible expansion process on the GOR of the thermal vapor compression driven system. It can be seen that as we go to an expansion process that is closer to being isentropic, the relative amount of irreversibility in the expansion process is greatly reduced and GOR is increased. As we increase the expander efficiency, we are producing more work for the RO unit. Hence, we expect that at high expander efficiencies the majority of the water will be produced in the RO section of the system.

Figure 8-7 represents the variation in the amount of water produced in the dehumidifier relative to the total water produced in the system with changes in expander efficiency. It can be seen that at lower expander efficiencies most of the water is produced in the dehumidifier and at higher expander efficiencies most of the water is produced in the RO unit. The high performance realizations of the system corresponds to the later case. This leads us to conclude that because of the irreversibilities in the overall system, for attaining high performance, it is better to recover energy as a work transfer in the expander rather than as a heat transfer in the dehumidifier. This translates into operating the HDH system as a combined desalination and power production system with the RO driven by the power produced in the expander.

### 8.4.2 Effect of heating steam conditions

Earlier in this chapter, we had shown that in order to reduce the heat required to run a thermal desalination system the total entropy flow entering the system needs to be reduced. In other words, the heating steam used needs to be at a lower entropy state or of a lower mass flow rate (higher enthalpy state). We will first investigate the
Figure 8-6: The effect of expander efficiency on performance of thermal vapor compression driven HDH-RO system and the specific entropy generated in the expander. $T_{\text{sat, in}} = 30^\circ\text{C}; \varepsilon_H = 60\%; \varepsilon_D = 70\%; \eta_{\text{vvc}} = 30\%; P_{st} = 5\text{ MPa}; P_H = 86.96\text{ kPa}; P_D = 100\text{ kPa}; x_{st, in} = 1; \text{HCR}_H = 1$.

Figure 8-7: The effect of expander efficiency on relative importance of the HDH process to the thermal vapor compression driven HDH-RO system. $T_{\text{sat, in}} = 30^\circ\text{C}; \varepsilon_H = 60\%; \varepsilon_D = 70\%; \eta_{\text{vvc}} = 30\%; P_{st} = 5\text{ MPa}; P_H = 86.96\text{ kPa}; P_D = 100\text{ kPa}; x_{st, in} = 1; \text{HCR}_H = 1$. 

200
effect of using higher pressure (saturated) steam. Figure 8-8 illustrates the increase in GOR and the decrease in equivalent electricity consumption when higher pressure steam is used. For this example, the component effectivenesses and efficiencies are fixed along with the operating pressures and feed seawater conditions. The air and water side pressure drops are assumed to be zero.

![Graph illustrating the effect of steam pressure on performance of thermal vapor compression driven HDH-RO system.](image)

Figure 8-8: The effect of steam pressure on performance of thermal vapor compression driven HDH-RO system. $T_{sw,in} = 30^\circ C$; $\varepsilon_H = 60\%$; $\varepsilon_D = 70\%$; $\eta_{vc} = 30\%$; $\eta_a = 50\%$; $P_H = 86.96$ kPa; $P_D = 100$ kPa; $x_{st,in} = 1$; $HCR_H = 1$.

We can clearly observe the strong impact that an increase in steam pressure can have on the performance of the HDH-TVC system. In this example, when the steam pressure is increased from 250 kPa to 1000 kPa (i.e. from a saturated steam temperature of 127.4°C to 179.9°C) the GOR is increased by 55%. It is also important to note that even though we use a higher pressure steam, the equivalent electricity consumption is still reduced by 14% for the aforementioned increase in steam pressure. This is because the mass flow rate of high pressure steam extracted from the (fictious) steam turbine is small and the corresponding work lost in the steam turbine
is reduced. Figure 8-9 illustrates the corresponding decrease in total entropy rate entering and leaving the system. The difference $\dot{S}_{in} - \dot{S}_{out}$ also decreases as we go to higher motive steam pressures. This is consistent with Eq. 8.5 and explains the increase in GOR from a thermodynamic perspective.

![Figure 8-9: The effect of steam pressure on total entropy rate of steam entering and leaving the system, to explain the trends in Fig. 8-8.](image)

In Fig. 8-10, we explain the increase in GOR by looking at the variations in two parameters- vapor entrainment ratio and vapor productivity ratio. In Sec. 8.1.1, we had shown that GOR is approximately equal to the product of these two parameters. When we increase the motive steam pressure, as is expected, the vapor entrainment ratio in the TVC increases. The physics behind this has been discussed in detail in a previous publication [58]. As we increase the steam pressure, the vapor productivity ratio is also increased slightly. For example, when the steam pressure is increased from 250 kPa to 1000 kPa, ER$_{vap}$ is increased by 30.8% and VPR is increased by 22.3%. This explains the increase of GOR by 55%.

Another way to reduce the entropy of the steam entering the system is to reduce
its mass flow. To reduce the mass flow of the steam entering the system we can increase the steam superheat. This increases the specific entropy of the steam entering. Thus the change in total entropy rate when we increase the steam temperature at a given pressure is a trade-off between the decrease in mass flow and increase in specific entropy. From several calculations it has been found that the decrease in mass flow always dominates and total entropy rate is reduced. However, this decrease is much smaller than that associated with the increase in steam pressure at saturation conditions. Considering this and the increase in material costs associated with producing superheated steam, it is better to operate our system using high pressure and saturated steam.

Figure 8-10: The effect of steam pressure on vapor entrainment ratio and vapor productivity ratio, to explain the trends in Fig. 8-8.
8.4.3 Effect of pressure ratio

Pressure ratio is defined as the pressure of moist air leaving the humidifier to the pressure of moist air entering the dehumidifier. It is essentially the pressure ratio across the TVC created by the steam supply. As we had already seen in the previous chapter, this can be the most important design parameter for varied pressure HDH systems. Figure 8-11 illustrates the change in GOR and equivalent electricity consumption when the pressure ratio is varied in an on-design sense. For this example, the component effectivenees and efficiencies are fixed along with the dehumidifier and the heating steam pressures, and the feed seawater conditions. The air and water side pressure drops are assumed to be zero. For the present calculations the pressure ratio was varied from 1.2 to 1.8. The lower limit on pressure ratio is imposed by the dehumidifier minimum terminal temperature difference. It is observed that as we go to a smaller pressure ratio the GOR increases and the equivalent electricity consumption decreases. For example, GOR is 13.6% higher and $E_c$ is 19.2% lower when the system is designed at a pressure ratio of 1.2 instead of 1.8.

This variation in the performance can be explained using Fig. 8-12. From this figure, it is observed that as we increase the pressure ratio, the vapor entrainment ratio is decreased. This is only to be expected since a larger pressure rise in the TVC will require a larger motive steam flow. Also, as we increase the pressure ratio the VPR is increased. This is because the dehumidification and humidification is carried out more effectively to produce water at larger pressure ratios. The increase in VPR is less than the decrease in vapor entrainment ratio. VPR is 43% higher and $ER_{\text{vap}}$ is 41% lower when the system is designed at a pressure ratio of 1.8 instead of 1.2. This explains the lower GOR at higher pressure ratios.

8.4.4 Effect of operating pressures

All previous HDH systems in literature have been designed to operate at atmospheric pressure. However, to increase the vapor content of moist air, the system needs to be operated at sub-atmospheric pressures. For example, at a dry bulb temperature of 65
Figure 8-11: Effect of pressure ratio on performance of thermal vapor compression driven HDH-RO system. $T_{sw,\text{in}} = 30^\circ\text{C}$; $\varepsilon_H = 60\%$; $\varepsilon_D = 70\%$; $\eta_{lvc} = 30\%$; $\eta_e = 50\%$; $P_D = 100\ \text{kPa}$; $P_{st} = 5\ \text{MPa}$; $x_{st,\text{in}} = 1$; $\text{HCR}_H = 1$.

Figure 8-12: The effect of pressure ratio on vapor entrainment ratio and vapor productivity ratio, to explain the trends in Fig. 8-11.
°C the humidity ratio of moist air is increased two-fold when the operating pressure is reduced from 100 kPa to 50 kPa. We now investigate the effect that this change in pressure can have on the system performance. Figure 8-13 illustrates the change in GOR and equivalent electricity consumption when the humidifier pressure is varied at a fixed pressure ratio. For this example, the component effectivenesses and efficiencies are fixed along with the heating steam pressure, and the feed seawater conditions. The air and water side pressure drops are assumed to be zero. It is observed that as we change the humidifier pressure there is very little change in GOR and $\dot{E}_c$.

![Graph showing GOR and equivalent electricity consumption](image)

Figure 8-13: Effect of humidifier pressure on performance of thermal vapor compression driven HDH-RO system. $T_{sw,in} = 30^\circ$C; $\varepsilon_H = 60\%$; $\varepsilon_D = 70\%$; $\eta_{vce} = 30\%$; $\eta_a = 50\%$; $P_D/P_H = 1.15$; $P_{st} = 5$ MPa; $x_{st,in} = 1$; HCR$_H = 1$.

VPR increases and ER$_{vap}$ decreases with decrease in humidifier pressure. The increase in VPR is almost at the same rate as the decrease in ER$_{vap}$. Hence, GOR is almost constant with variation in humidifier pressure. As we had already explained, when the humidifier operating pressure is reduced the amount of vapor content in the air-vapor mixture will increase greatly. Hence, as we go to a lower $P_H$ the amount of vapor leaving the humidifier ($\dot{m}_{da} \cdot \omega_{H,a}$) will increase. We recall that this term is the
numerator in VPR (see Eq. 8.14) and the denominator in ER\textsubscript{vap} (see Eq. 8.15).

Figure 8-14: The effect of humidifier pressure on vapor entrainment ratio and vapor productivity ratio, to explain the trends in Fig. 8-13.

8.4.5 Effect of air side pressure drop in heat and mass exchangers

Figures 8-15 & 8-16 illustrate the effect of air side pressure drop in the humidifier and the dehumidifier on the performance of TVC driven HDH systems. For this example, the component effectivenesses and efficiencies are fixed, along with the heating steam pressure and the feed seawater conditions.

It is observed that as the pressure drop increases, the GOR decreases rather drastically. This is because pressure drops result in lost compression work which cannot be recovered in the expander. Hence, it is vital to design the HME devices such that the pressure drop is minimal. The pressure drop in the dehumidifier and humidifier decreases GOR by a similar amount. At higher values of pressure drop there is an optimum pressure ratio at which the GOR is maximum.
Figure 8-15: Effect of air-side pressure drop in dehumidifier on performance of thermal vapor compression driven HDH-RO system. $T_{\text{ev,in}} = 30^\circ\text{C}; \varepsilon_H = 60\%; \varepsilon_D = 70\%; \eta_{\text{kvc}} = 30\%; \eta_e = 50\%; P_D = 100\,\text{kPa}; P_{st} = 5\,\text{MPa}; x_{\text{st,in}} = 1; \text{HCR}_H = 1$.

Figure 8-16: Effect of air-side pressure drop in humidifier on performance of thermal vapor compression driven HDH-RO system. $T_{\text{ev,in}} = 30^\circ\text{C}; \varepsilon_H = 60\%; \varepsilon_D = 70\%; \eta_{\text{kvc}} = 30\%; \eta_e = 50\%; P_D = 100\,\text{kPa}; P_{st} = 5\,\text{MPa}; x_{\text{st,in}} = 1; \text{HCR}_H = 1$.

208
8.5 Comparison with existing desalination techniques.

In this section, a niche is identified for the new technology proposed in this paper. In the absence of reliable cost data, we will do this by considering the scale of application (in m³ of water produced per day), the energy performance of various desalination systems and the quality of the energy source used. To evaluate and compare the performance of various thermal and electricity driven desalination technologies with that of the new system - gained output ratio (GOR) and equivalent electricity consumption $\dot{E}_c$ are used. GOR is a thermal energy based performance parameter (see Eq. 2.2). In order to calculate it for electricity driven systems, we use a power production efficiency ($\eta_{PP} = 40\%$) to convert electricity consumed to thermal energy.

Hence, for electrical energy driven systems,

$$\text{GOR} = \frac{h_{fg} \cdot \eta_{PP}}{\dot{E}_c \cdot 3.6} \quad (8.24)$$

Figure 8-17 plots both the equivalent electrical energy consumption and the GOR against the size of the system in m³/day. Performance of the various desalination processes is obtained from the widely cited review paper by Miller [107].

The performance values shown for the HDH-TVC-RO system is in two bands. The lower band of performance values correspond to the system with off-the-shelf components ($\eta_{tvc} = 30\%, \eta_e = 50\%, \varepsilon_H = 60\%$ and $\varepsilon_D = 70\%$) and the higher one corresponds to the system with custom built (and more efficient components) ($\eta_{tvc} = 40\%$ to 50%, $\eta_e = 80\%, \varepsilon_H = 70\%$ and $\varepsilon_D = 70\%$). Both bands of performance values are for systems using steam at 10 to 50 bar.

The application of this system is likely to be limited to a medium or large scale because medium pressure steam at 10-50 bar is expected to be uneconomical to produce on a small scale. In the medium to large scale market, the competing steam driven desalination technologies are MED, MSF and Rankine cycle driven RO. From Fig. 8-17 it is observed that HDH-TVC-RO system can potentially outperform MSF
Figure 8-17: Benchmarking of new HDH techniques against existing desalination systems.
and perform as well as MED in terms of GOR and equivalent electricity consumption. This performance is at a higher heating steam pressure (and temperature) than MSF and MED. MSF and MED can run using 80-120°C saturated steam. Also, it is common knowledge that Rankine cycle power plants require anywhere between 400-600°C steam temperature to run economically and efficiently. Such a steam source cannot be produced economically when RO is a stand alone desalination unit, it needs to be in a co-production environment coupled to a large scale Rankine cycle power plant for such a energy source to be available.

Hence, the niche for the HDH-TVC-RO technology is medium scale, stand alone, decentralized seawater desalination using medium pressure steam. It would be desirable to produce the steam using solar energy. Coastal communities which have high solar insolation, high water scarcity and inavailability of fossil fuels are a target for this technology. This is unlike other HDH technology described in this thesis which are applicable for small-scale, community-level water treatment.

8.6 Chapter conclusions

1. A novel carrier gas based thermal desalination system run using a thermal vapor compressor is described in this Chapter. This system can use steam at a higher temperature (and lower total entropy rate) than existing thermal desalination systems.

2. This system has been analyzed in detail in an on-design sense by assigning an energy effectiveness for the humidifier and dehumidifier, and an isentropic efficiency for the expander and a reversible entrainment efficiency for the TVC.

3. It has been found that recovering energy given by steam in the TVC as work in an expansion process leads to a more efficient system than using a isenthalpic throttling process. However, the use of an efficient expander leads to the majority of the water being produced in the RO unit coupled to it. In these scenarios, the HDH system itself operates as a power and water coproduction unit where
all of the power produced is used for further desalination in a RO unit.

4. For minimum specific energy consumption of the HDH-TVC-RO system, use of an efficient TVC and high pressure (\(P_{st} = 10\) to 30 bar) steam are most important.

5. It is also crucial to design the system for as low a pressure ratio as is possible. The air side pressure drops in the HME devices also play a significant role.

6. The absolute value of pressure in the humidifier and the dehumidifier have been found to have little effect on the system performance.

7. The HDH-TVC-RO system is appropriate for medium scale, stand alone, decentralized seawater desalination using medium pressure steam. In these situations a GOR of 20 and an equivalent electricity consumption of 9.5 kWh/m\(^3\) can be attained using components with high efficiencies.
Chapter 9

Multi-stage bubble columns for high heat rate dehumidification

When a non-condensable gas is present, the thermal resistance to condensation of vapor on a cold surface is much higher than in a pure vapor environment. This is, primarily, because of the diffusion resistance to transport of vapor through the mixture of non-condensable gas and vapor. Several researchers have previously studied and reported this effect [108-116]. There is a general consensus that, when even a few mole percent of non-condensable gas is present in the condensing fluid, the deterioration in the heat transfer rates could be up to an order of magnitude [117-122]. From experimental reports in literature it can be observed that the amount of deterioration in heat transfer is a very strong (almost quadratic) function of the mole fraction of non-condensable gas present in the condensing vapor. For this reason, a deaerator is usually used in power plants to prevent the accumulation of non-condensable gas in the steam condenser.

In HDH systems, a large percentage of air (60-90% by mass) is present by default in the condensing stream. As a consequence it has been found that, in these systems, the heat exchanger used for condensation of water out of an air-vapor mixture (otherwise known as dehumidifier) has very low heat and mass transfer rates (an ‘equivalent’ heat transfer coefficients as low as 1 W/m²·K in some cases [62, 63]). In this chapter, we propose to improve the heat transfer rate by condensing the vapor-gas mixture in
a column of cold liquid rather than on a cold surface by using a bubble column heat (and mass) exchanger. Bubble columns are extensively used as multiphase reactors in process, biochemical and metallurgical applications [123]. They are used especially in chemical processes involving reactions which have a very high heat release rate associated with them (such as the Fischer-Tropsch process used in the manufacture of synthetic fuels) [124, 125]. We propose to apply this device for condensation of the air-vapor mixture with a large percentage of air present in it. Figure 9-1 illustrates the proposed device schematically. In this device, moist air is sparged through a porous plate (or any other type of sparger [126]) to form bubbles in a pool of cold liquid. The upward motion of the air bubbles causes a wake to be formed underneath the bubble which entrains liquid from the pool, setting up a strong circulation current in the liquid pool [127]. Heat and mass are transferred from the air bubble to the liquid in the pool in a direct contact transport process. At steady state, the liquid, in turn, loses the energy it has gained to the coolant circulating through a coil placed in the pool for the purpose of holding the liquid pool at a steady temperature.

9.1 Predictive model for combined heat and mass transfer

In this section, we develop a thermal resistance model for the condensation of water from an air-vapor mixture in a bubble column heat exchanger. Figure 9-2 illustrates a local thermal resistance network describing the heat and mass transfer processes in the bubble column condenser. To draw this network, we define local energy-averaged 'bulk' temperatures for the condensing mixture, the liquid in the pool, and the coolant and also approximate the heat transfer to be locally one-dimensional.

The four temperature nodes in the network are: (1) the average local temperature of the air-vapor mixture in the bubbles ($T_{air}$), (2) the average temperature of the liquid in the pool ($T_{column}$), (3) the local temperature of the coil surface ($T_{coil}$), and
Figure 9-1: Schematic diagram of the bubble column dehumidifier.

Figure 9-2: A thermal resistance model for the bubble column dehumidifier.
the average local temperature of the coolant inside the coil \(T_{\text{coolant}}\). Between \(T_{\text{air}}\) and \(T_{\text{column}}\) there is direct contact heat and mass transfer. The heat transfer is via a thermal resistance represented by \(R_{\text{sensible}}\) and the mass transfer is represented as a (latent) heat source \(q_t = j \cdot h_{fg}\). The thermal resistance due to the coil wall itself will be very small and has been neglected. This is especially true in the cases considered in this thesis since copper tubing is used. In cases where stainless steel or a lower thermal conductivity metal is used, this resistance might not be negligible. Between the coil surface and the bubble there could be direct contact heat exchange and associated condensation of vapor on the coil surface. The heat transfer is via a heat transfer resistance \(R_{\text{impact}}\) and the mass transfer are represented by the heat source \(q_{t,\text{impact}}\). Several researchers [124, 125, 128–130] have previously studied the thermal resistance between the pool of liquid and the immersed surface in a bubble column reactor. In the current chapter, this resistance is represented as \(R_{bc}\) between \(T_{\text{coil}}\) and \(T_{\text{column}}\). Finally, there is a convective resistance inside the coil for the coolant flow represented by \(R_{\text{coil}}\).

In order to simplify the circuit, the direct impact of the bubble on the coil surface is approximated to have negligible effect on the heat and mass transfer (i.e. \(R_{\text{impact}}\) and \(q_{t,\text{impact}}\) are neglected). The experiments were designed and carried out such that this approximation was satisfied (see section 9.2) and the effect of direct impact was dealt with in a separate set of experiments (see section 9.3.6). Each of the remaining resistances depicted above will be modeled using reasonable simplifying assumptions in the following paragraphs.

9.1.1 Thermal resistance between the liquid in the column and the coil surface

In bubble column reactors used in the chemical industry, proper design of the heat transfer surfaces is vital to maintain catalytic activity, reaction integrity and product quality since the reactions typically involve very high heat release rates because of their highly exothermic or endothermic nature. In these scenarios, the temperature
of the liquid in the column is of utmost importance. Hence, several studies have been conducted over the last five decades on modeling and measuring the heat transfer coefficients between the liquid and the heat transfer surface.

In a pioneering effort, Konsetov [129] proposed a semi-analytical model based on the assumption that heat rates are determined by isotropic turbulent fluctuations in the liquid. He approximated the characteristic dimension for heat transfer from the liquid to the coil to be the coil diameter and used the Kutateladze model [131] for determining the gas holdup. Konsetov used a flexible constant to fit the data from the model to experimental data in literature. This correlation, however, has not been widely used for bubble column reactor design.

Kast [130] developed a model by considering that a fluid element in front of the rising bubble receives radial momentum and moves toward the wall. This was postulated to break up the boundary layer at the wall. The author proposed that below the bubble, liquid is sucked in at a radial velocity \( V_r \) and that this results in a capacitive heat transport given by \( V_r \cdot \rho \cdot c_p \). He further observed that \( V_r \) is proportional to the superficial gas velocity \( V_g \) and defined Stanton number as \( St = \frac{h}{\rho \cdot c_p \cdot V_g} \). Based on intuitive reasoning Kast proposed the following correlation.

\[
St = f(ReFrPr)^n
\]  

(9.1)

Deckwer [128] used the Kolmogorov theory of isotropic turbulence [132] and Higbie’s theory of surface renewal [133] to explain the form of the equation (Eq. 9.1) proposed by Kast. By observing that there is no experimental evidence of a physical length scale for the heat transfer, Deckwer postulated that the micro eddy scale of energy dissipation (see Eq. 9.2) proposed by Kolmogrov is an ideal characteristic length scale for the problem at hand. The author also proposed use of the Kolmogorov velocity (see Eq. 9.3) as the characteristic velocity.

\[
l = \left( \frac{\nu^3}{\varepsilon} \right)^{1/4}
\]  

(9.2)

\[
V = (\nu \varepsilon)^{1/4}
\]  

(9.3)
The heat transfer correlation (Eq. 9.4) thus derived by Deckwer had the same form as Kast’s model and used a flexible constant.

\[ St = c(ReFrPr)^{-0.25} \]  

(9.4)

In this section, the present authors have described an improved model for predicting the heat transfer rate between the liquid in the column and the coil surface. In this model we (like Deckwer [128]) used Higbie’s theory of surface renewal however, with a different length scale. In fluid elements adjacent to the surface, unsteady heat diffusion takes place and is described by the following equation:

\[ \frac{\partial T}{\partial t} = \alpha \frac{\partial^2 T}{\partial x^2} \]  

(9.5)

The appropriate boundary conditions to describe this problem are ones that describe the temperature \( T \) as the wall temperature \( T_{\text{coil}} \) at \( x = 0 \) and all times, the bubble temperature as the initial temperature at all \( x \), and \( T \) as the bubble temperature at \( x = \infty \) at all times. The final boundary condition is possible only when we approximate the fluid element to have infinite depth and the contact times to be short.

\[ T = T_{\text{coil}} \quad x = 0 \quad t \geq 0 \]  

(9.6)

\[ T = T_{\text{bubble}} \quad x > 0 \quad t = 0 \]  

(9.7)

\[ T = T_{\text{bubble}} \quad x = \infty \quad t > 0 \]  

(9.8)

Solving the above equations, we can obtain the following expression for the heat flux and the thermal resistance.

\[ q = 2 \frac{kpc_p}{t} \cdot (T_{\text{bubble}} - T_{\text{coil}}) \]  

(9.9)

\[ \frac{1}{R_{bc}} = 2 \frac{kpc_p}{\sqrt{\pi} t} \]  

(9.10)

218
From Eq. (9.10), it is clear that the resistance can be modeled by modeling the surface renewal time \( t \). For modeling \( t \), we need to model the characteristic length scale and velocity scale accurately. As stated earlier, Deckwer modeled the length and velocity using the Kolmogrov theory (Eqs. 9.2 & 9.3). These scales are indicative of the smallest eddies present in the flow and are the scales at which energy is dissipated. These are the scales that form the viscous sub-layer and are very small physically. Hence, this scale is unlikely to regulate the surface renewal mechanism and the physical mixing. We propose to use a more intuitive length scale which is the integral scale of turbulence.

The integral scale is the representative size of the largest energy bearing eddy. In some cases this scale can be defined by the physical constraints of the flow domain. For example, in pipe flow the diameter of the pipe is of the order of the largest eddies in the flow, and the ratio of the pipe diameter to mean velocity along the pipe is a good estimate of the time period required to describe the flow. In other cases where the integral scale is not obvious from the flow geometry, it can be defined using the autocorrelation of the velocity (i.e., the correlation of a velocity component with itself) as follows:

\[
l_{\text{int}} = \int_0^\infty \frac{u(x,t) \cdot u(x+r,t)}{u^2} dr
\]

where \( u \) is the root-mean-square velocity in the \( x \)-direction and \( r \) is the distance between two points in the flow. The determination of the integral scale using Eq. (9.11) is not straight-forward [134–136]. Direct numerical simulations or large scale visualization experiments using particle image velocimetry or other such techniques are normally used to obtain the autocorrelation of velocity in a 3D flow like in bubble columns [137]. Instead of going into these elaborate techniques, we propose to approximate the integral scale by the bubble diameter. Magaud et al. [138] have presented experimented data that supports this approximation. Similar results concluding that the integral length is of the order of the bubble diameter have been reported by other authors as well [139, 140].
There exist various expressions to calculate the bubble diameter (depending on flow regime) and most of them are empirical or semi-empirical. We use the following expression for the cases involved in this thesis [141]. This equation can also be derived from a simple force balance (with the gravitational force equaling the surface tension).

\[ D_b = \left( \frac{6 \sigma_d b}{\rho_l - \rho_g} \right)^{1/3} \]  \hspace{1cm} (9.12)

We also propose to use the liquid circulation velocity as the characteristic velocity. This is logical because the liquid that ‘renews’ the boundary layer formed on the heat transfer surface is at this velocity and when we aim to calculate the time between two ‘renewals’ we need to take this into account. Field and Rahimi [142] have proposed that the following expression, which is commonly used in literature [125], is appropriate to calculate liquid circulation velocity in bubble columns.

\[ V_c = 1.36 \left( gH(V_g - \epsilon \cdot V_b) \right)^{1/3} \]  \hspace{1cm} (9.13)

According to this expression, the circulation velocity is a function of the bubble velocity and several researchers have previously presented various expressions for calculating the same. An appropriate correlation from the wide selection needs to be picked based on the conditions in the bubble column. For the cases reported in this chapter we find it appropriate to use the following equation for evaluating bubble velocity developed based on Mendelson’s wave equation [143]. Our own experimental observations showed that this correlation works well for the cases reported in this thesis.

\[ V_b = \sqrt{\frac{2\sigma}{\rho_l D_b} + \frac{gD_b}{2}} \]  \hspace{1cm} (9.14)

To calculate circulation velocity based on Eq. (9.13) we also need to evaluate the volumetric gas holdup (\( \epsilon \)), for which we propose to use the following expression provided by Joshi and Sharma [127]. Several other correlations which have been reported by various researchers, but for the conditions under which we conducted the experiments in (described in Sec. 9.2) we find the following correlation to be the most appropriate.
Also, we find that this correlation (Eq. 9.15) is the most widely used by researchers in the field. Our own experimental observations showed that this correlation works well for the cases reported in this chapter.

\[ \epsilon = \frac{V_g}{0.3 + 2 \cdot V_g} \]  

(9.15)

Based on these expressions, we can evaluate the time between two ‘renewals’ as \( t = \frac{D_k}{v_c} \). By applying this contact time to Eq. (9.10), we evaluate the thermal resistance between the liquid in the column and the coil surface. This resistance is, however, a minor one in the network and the prediction of the same has little effect on the overall result for the cases reported in this thesis.

**9.1.2 Thermal resistance between the liquid in the column and the bubbles**

The high resistance to diffusion of vapor through a vapor-gas mixture is the reason that regular dehumidifiers have low heat transfer coefficients. In this section, we model the equivalent of the aforementioned diffusion resistance for the case of bubble column dehumidifiers. In Fig. 9-2, the total heat flux between the bubbles and the liquid was modeled as the sum of the heat flux due to condensation \( q_{\text{cond}} \) and the heat flux due to heat transfer through the resistance \( R_{\text{sensible}} \). We will evaluate the latent heat using a mass transfer resistance model and the sensible heat using a heat and mass transfer analogy. The mass transfer resistances associated with condensation are shown in Fig. 9-3. In drawing these resistances it is approximated that the condensation occurs at an interface just outside the bubble surface and mass averaged ‘bulk’ humidity ratios are defined for the vapor-gas mixture inside the bubble and at the bubble interior surface.

The mass transfer resistances depicted in Fig. 9-3 are: (1) the resistance to diffusion of vapor through the vapor-gas mixture in the bulk \( (\omega_{\text{bulk}}) \) to the bubble surface \( (\omega_{\text{bubble}}) \) and (2) the mass transfer resistance caused by bubble motion through the liquid. The first resistance is not easy to model without knowing the mechanism of
Figure 9-3: A mass transfer resistance model between the liquid in the column and the bubbles

Convective transport inside the bubble which could be augmented by a fluid circulation caused by rapid and asymmetric vertical motion of the bubble in the liquid pool. Since there are several complexities involved in evaluating the mechanism of transport inside the bubble, we assume a boundary layer is formed for diffusive transport and approximate the thickness of the boundary layer by the radius of the bubble itself. This is an upper limit for the size of the boundary layer and the associated thermal resistance and hence, in the succeeding sections (Sec. 9.3.5) it is shown that the heat transfer and condensation rates predicted by the model consistently underestimates those measured experimentally. The model equation is:

$$k_{l,1} = \frac{D_{AB}}{D_b/2}$$  \hspace{1cm} (9.16)

We model the resistance outside the bubble surface using surface renewal mechanism (similar to that presented in Eqs. (9.5-9.10)):

$$k_{l,2} = \frac{2}{\sqrt{\pi}} \sqrt{\frac{D_{AB}}{t}}$$  \hspace{1cm} (9.17)

The surface renewal time \( t \) in this case is modeled as the ratio of the bubble diameter and the bubble slip velocity. The bubble slip velocity is the relative velocity of the bubble with respect to the circulating liquid. The liquid circulation velocity and the bubble velocity are calculated using the expression presented in Eq. (9.13) & (9.14) respectively. The model equation is:

$$t = \frac{D_b}{V_b - V_c}$$  \hspace{1cm} (9.18)
The heat transfer resistance \( R_{\text{sensible}} \) can be modeled by defining Lewis factor (\( Lef \)) for the vapor-gas system. The Lewis factor appears in the governing equations of simultaneous heat and mass transfer processes (for example, in wet-cooling towers [144] and in cooling coils [145]). \( Lef \) is defined by Eq. (9.19) and is directly related to Lewis number which is a fluid property:

\[
Lef = \frac{h_t}{k l \rho c_{pg}} \quad (9.19)
\]

\[
Lef \approx L e^{2/3} [145]
\]

\[
\approx 0.89 - 0.92 \text{ for air-water systems [146]} \quad (9.21)
\]

\[
Le = \frac{\alpha}{D_{AB}} \quad (9.22)
\]

where \( h_t \) is the heat transfer coefficient associated with \( R_{\text{sensible}} \), \( k l \) is the mass transfer coefficient associated with the latent heat, and \( c_{pg} \) is the specific heat at constant pressure of the vapor-gas mixture:

\[
\frac{1}{h_t A} = R_{\text{sensible}} \quad (9.23)
\]

\[
\frac{1}{k l A} = \left( \frac{1}{k l,1} + \frac{1}{k l,2} \right)^{-1} \quad (9.24)
\]

Here, the heat and mass transfer coefficients are defined based on the heat transfer area of the coil surface \((A)\) instead of the bubble surface area. This is because from an engineering perspective, we need to evaluate the coil area required for a certain total heat duty in the bubble column dehumidifier.

Finally, the correlations for heat transfer coefficient for flow inside circular tubes are well known and documented in heat transfer text books [94]. Based on the flow regime inside the coil, we selected appropriate correlations to evaluate \( R_{\text{coil}} \).

### 9.1.3 Evaluation of total heat flux from the resistance model

In the preceding sections we presented discussed the models for the various thermal resistances in the bubble column dehumidifier (Fig. 9-2). In this section, we present the equations needed to solve for the total heat flux and all the temperatures in the
bubble column dehumidifier.

The heat flux through the network associated with the sum of the bubble column resistance \( R_{bc} \) and the convection resistance in the coil \( R_{coil} \) is defined as follows:

\[
q = \frac{\dot{Q}}{A} = \frac{\theta_1}{R_{bc} + R_{coil}} \tag{9.26}
\]

The associated log mean temperature difference \( \theta_1 \) is defined between the liquid column temperature \( T_{column} \) and the coolant inlet/exit temperatures. It is very important to note that experimental data in the literature and our own experimental data reported later in this chapter (See Sec. 9.2) have shown the liquid in the column is at a constant temperature because of rapid mixing induced by the bubbles. The LMTD is given as follows:

\[
\theta_1 = \frac{(T_{column} - T_{coolant,in}) - (T_{column} - T_{coolant,out})}{\ln \left( \frac{T_{column} - T_{coolant,in}}{T_{column} - T_{coolant,out}} \right)} \tag{9.27}
\]

The heat flux can also be expressed as sum of the latent heat of condensation of the vapor from the vapor-air bubbles into the liquid column and the associated sensible heat transfer.

\[
q = q_{latent} + q_{sensible} \tag{9.28}
\]

The sensible heat flux is the one associated with the resistance \( R_{sensible} \). The heat transfer coefficient associated with this resistance is evaluated using Eq. (9.19). It is important to note that the area is normalized using the specific interfacial area of the bubbles. We have

\[
q_{sensible} = \frac{\theta_2}{R_{sensible}} \tag{9.29}
\]

\[
\frac{1}{h_t A} = R_{sensible} \tag{9.30}
\]

\[
h_t = Le_f \cdot (\rho c_{p,g} k_t) \cdot \frac{a_{s,vol}}{A} \tag{9.31}
\]
The specific interfacial area is evaluated using the following widely used expression [141].

\[ a_s = \frac{6c}{D_b} \]  

(9.32)

The associated log mean temperature difference is defined between the column temperature and the air inlet/exit temperature:

\[ \theta_2 = \frac{(T_{\text{air,in}} - T_{\text{column}}) - (T_{\text{air,out}} - T_{\text{column}})}{\ln \left( \frac{T_{\text{air,in}} - T_{\text{column}}}{T_{\text{air,out}} - T_{\text{column}}} \right)} \]  

(9.33)

The latent heat transfer rate is calculated using the following expression based on mass flux:

\[ q_{\text{latent}} = j \cdot h_{fg} \]  

(9.34)

The mass flux is evaluated by using the mass conversion equation across the bubble column condenser:

\[ j = \frac{\dot{m}_{da}}{A} (\omega_{in} - \omega_{out}) \]  

(9.35)

The energy balance between the coolant and the air is written as follows

\[ q = \frac{\dot{m}_{\text{coolant}} c_{p,\text{coolant}}}{A} (T_{\text{coolant,out}} - T_{\text{coolant,in}}) = \frac{\dot{m}_{da}}{A} (h_{\text{air,in}} - h_{\text{air,out}}) \]  

(9.36)

(9.37)

By applying a mass balance on the vapor over a incremental time \( dt \) and integrating the same over a the residence time for the bubble in the liquid \( t_f \) we obtain the following expression.

\[ k_l \cdot a_s = \frac{1}{t_f} \ln \left[ \frac{\omega_{in} - \omega_{sat}}{\omega_{out} - \omega_{sat}} \right] \]  

(9.38)

Where the bubble residence time \( t_f \) is evaluated as the ratio of the liquid height and the bubble velocity:

\[ t_f = \frac{H}{V_b} \]  

(9.39)

By solving Eqs. (9.25-9.39) we can obtain the heat flux and the associated temperatures from the estimated thermal resistance [Eqs. (9.5-9.24)].
Solution technique

The equations presented in this section are solved simultaneously using Engineering Equation Solver (EES) [77]. The various fluid property packages used in EES have been previously been explained in Chapter 3.1.3.

9.2 Experimental details

A laboratory scale test rig was designed and built to study the condensation process from a vapor-air mixture in a bubble column condenser. Figure 9-4 shows a schematic diagram of the test apparatus used in the study. The apparatus consists of two bubble columns (4) and (9) with dimensions of 12" (304.8 mm) width x 12" (304.8 mm) length x 18" (457.2 mm) height made from transparent PVC sheets of 3/8" (9.52 mm) thickness. The first column (4) is used to produce moist air for the experiment by passing air through a sparger (3) into hot water. The water in this column is heated by a 1.5 kW submerged electric heater (5). The air is supplied from a compressor and the flow rate is controlled by valve (1) and measured by rotameter (2). The humidified air from the first bubble column flows to the test column (9) where the dehumidification measurements are carried out. Before entering the second column, the flow rate is measured by a rotameter (6), the pressure is measured by a pressure gauge (7), and the dry and wet bulb temperatures are measured with thermocouples T1 and T2 respectively. Air flows into the sparger of the test column (8) where it is cooled and dehumidified using the cold copper coil (10). The copper coil has a pipe diameter of 1/4" (6.35 mm), a coil height of 6" (152.4 mm) and a turn diameter of 9" (228.6 mm). Cold water acting as the coolant flows inside the coil and is pumped from the cooling tank (15) where chilled water coil (16) keeps the temperature inside this tank almost constant. The dry bulb temperature and wet bulb temperature of the outlet air from the second column are measured by thermocouples T3 and T4 respectively. The two columns are provided with a charging and emptying valve at the back side (not shown in figure). Cold water from the cooling water tank (15) is pumped into the copper coil (10). The flow rate of the water is adjusted by the
inline valve (11) and the bypass valve (12). The flow rate of water is measured by rotameter (13) and the fine temperature of the water can be adjusted by the inline electric water heater (14). The inlet and outlet water temperature from the copper coil are measured by thermocouples T5 and T6 respectively. The water temperature in the condenser bubble column is measured at two levels using thermocouples T7 and T8.

The sparger (3) of the humidifier bubble column (4) is a cartridge type sparger of 10" (254 mm) length. The sparger is from Mott corporation made of stainless steel (316LSS) porous pipe of 2" (50.8 mm) outside diameter and 1/16" (1.59 mm) thickness. This sparger generates uniform and fine bubble sizes and has a pressure drop less than 13.7 kPa (2 psi). The sparger (8) in the dehumidifier column is made of aluminium box 10" (254 mm) x 10" (254 mm) x 1" (25.4 mm). The top cover of this box is made of an acrylic sheet with a number of holes drilled in it to generate the air bubbles. There are 5 acrylic sheets; each one has different number of holes, hole diameter, and hole pitch as shown in Table 9.1. The thermocouples used in the apparatus are of K-type are connected to a data logger and a PC. The thermocouples and the data logging system have an uncertainty of ±0.1°C. The rotameters used for air flow measurements have a range of 0.8 - 8.2 ft³/min (378 - 3870 m³/s) with a least count of ±0.2 ft³/min (±94.4 m³/s). The rotameter used for water flow
measurement has a range of 0.01 - 0.85 L/min with a least count of 0.01 L/min.

<table>
<thead>
<tr>
<th>Design No.</th>
<th>Hole size (mm)</th>
<th>Pitch (mm)</th>
<th>Number of holes</th>
</tr>
</thead>
<tbody>
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<td>121</td>
</tr>
<tr>
<td>2</td>
<td>2.38</td>
<td>23</td>
<td>64</td>
</tr>
<tr>
<td>3</td>
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<td>26</td>
<td>36</td>
</tr>
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<td>4</td>
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<td>32</td>
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</tr>
<tr>
<td>5</td>
<td>4.76</td>
<td>40</td>
<td>16</td>
</tr>
</tbody>
</table>

In order to study the impact of bubble-on-coil, we designed a set of circular coils to avoid impact and a set of serpentine coils which will facilitate impact. The photographs of the two set of coils are shown in Fig. 9-5. It may be observed that the circular coil has a turn diameter of 9" (228.6 mm) and the sparger face area is 8" (203.2 mm) x 8" (203.2 mm) which brings the point of inception of the bubble to be vertically away from the coil. This helps minimize impact. In the serpentine coil, each pass of the coil is deliberately made to cross over the sparger holes maximizing impact. The results are markedly different for the two coils and are reported later on in this chapter (see Sec 9.3.6).

### 9.3 Results and discussion

In this section, we explain the importance of parameters such as superficial velocity, inlet mole fraction of vapor, bubble diameter, liquid height and effect of bubble-on-coil impact on the performance of a bubble column dehumidifier by varying these parameters independently.

The performance parameter of interest is the total heat flux exchanged between the coolant and the air-vapor mixture. Other alternative performance parameters, such as an 'equivalent' heat transfer coefficient, are not strictly correct, in contrast to the situation for a heat exchanger. This is because defining a global value for heat transfer coefficient will involve defining a log mean temperature difference (or another such global parameters for the device in its entirety) between the air and water.
Figure 9-5: Photographs showing design of sparger and coil for (a) non-impact and (b) impact cases
temperatures at the inlet and outlets. This would amount to associating the mass transfer (and the associated latent heat release, which is the major portion of the total heat exchanged between the fluid streams) with a temperature difference: and this is of course inappropriate because the mass transfer is associated with a concentration difference and not a temperature difference [20, 21, 88]. It is, hence, logical to use heat flux as the performance parameter since it captures all the important characteristics of the bubble column dehumidifier (including the condensation rate) but does not involve all of the aforementioned issues.

9.3.1 Effect of superficial velocity

Several researchers have studied the effect of superficial velocity on mass transfer in bubble columns [124, 147–150]. These studies, however, did not involve condensation from the bubble into the liquid column. A typical example of the mass transfer studies in literature would be absorption of isobutylene in aqueous solutions of H₂SO₄ [151, 152]. Researchers have also separately studied the effect of superficial velocity on heat transfer to immersed surfaces in bubble column reactors [128–130, 153–158]. However, we should note that the effect of superficial velocity on simultaneous heat and mass exchange with condensation has not been studied before (to the best knowledge of the authors), and it will be the focus of this section.

The general consensus in literature is that the heat and mass transfer coefficients are higher at higher superficial velocity [159]. Studies have also shown that the rate of increase of heat transfer coefficients with gas velocity is more pronounced at lower gas velocity, and more gradual at higher gas velocities. This is because of the change in flow regime from homogenous bubbly flow to the churn-turbulent regime. The effect of increase in superficial velocity is reported to be lower in the churn-turbulent regime.

A flow regime map reported by Shah et al. [124] predicts that the transition velocity for the experimental bubble column reported in the current chapter lies somewhere between 4.5 and 7 cm/s (based on an effective column diameter of 30.5 cm). During experimentation it was observed that between superficial velocities of 3 to 8 cm/s the
bubble flow was either perfect or imperfect bubbly flow (churn turbulence and slug flow was not observed).

Figure 9-6 illustrates experimental and calculated values of heat flux at various values of superficial velocity. These results are at fixed values for bubble diameter, inlet mole fraction and water column height. The trend and the slope of the curve presented in Fig. 9-6 is representative of the trend obtained at other values of the aforementioned fixed parameters. From Fig. 9-6, it may be observed that as the superficial velocity was increased so was the heat flux which is a conclusion consistent with other such studies in literature. In addition, it can be observed that the predictive model estimates the effect of the superficial velocity accurately.

![Graph](image)

Figure 9-6: Effect of superficial velocity on the total heat flux in the bubble column measured and evaluated at \(D_b = 4\) mm; \(\chi_{in} = 21\%\); \(H = 254\) mm.

The uncertainty of measurement on the superficial velocity is ± 0.11 cm/s and that on the heat flux is ± 5%.

231
9.3.2 Effect of bubble diameter

In the literature, no consensus is evident on the effect of bubble diameter on transport coefficients in bubble columns. While on the one hand some researchers have reported that bubble properties (including bubble diameter) affect the mass transfer coefficient greatly [160–162], on the other hand Deckwer [128] has suggested that there is no evidence of the effect of bubble diameter on heat transfer to immersed surfaces. Also, the effect of bubble diameter on simultaneous heat and mass transfer has not been investigated yet.

Our experiments and modeling show (see Fig. 9-7) that there is a relatively minor but discernible effect of bubble diameter on the total heat flux exchanged in a bubble column dehumidifier. The heat flux is found to decrease with an increase in bubble diameter. This result is found to be consistent at other values of the fixed parameters (superficial velocity, inlet mole fraction and liquid height) as well. It is to be noted that the predictive model proposed in Sec. 9.1 predicts the trend in Fig. 9-7 to a good degree of accuracy.

9.3.3 Effect of inlet mole fraction

In steam condensers with a small amount of non-condensable gas present (< 10% by mole) the inlet mole fraction of vapor has been reported to have a very sharp effect on the heat transfer coefficient [117, 121]. As mentioned earlier in this chapter, experimental data in the literature suggests that the effect is almost quadratic in nature. In this section, we investigate the effect of the same parameter in a bubble column dehumidifier.

The inlet mole fraction of vapor is varied from 10% to 25% (3.6 to 9 times lower than regular condensers) at fixed values of superficial velocity, bubble diameter and liquid height. Fig. 9-8 illustrates the experimental and modeling results for the same. A strong effect of the mole fraction is seen, as is also the case in steam condensers. From our experiments, we observe that the effect is more linear than quadratic (in the studied range). Hence, the presence of non-condensable gas is affecting the heat
Figure 9-7: Effect of bubble diameter on the total heat flux in the bubble column measured and evaluated at $V_g = 3.8 \text{ cm/s}; \chi_{in} = 21\%; H = 254 \text{ mm}$.

Figure 9-8: Effect of inlet mole fraction of the vapor on the total heat flux in the bubble column measured and evaluated at $V_g = 3.8 \text{ cm/s}; D_b = 4 \text{ mm}; H = 254 \text{ mm}$.
transfer to a much lesser degree than in the film condensation situations of a standard
dehumidifier. This demonstrates the superiority of the bubble column dehumidifier
[32]. This observation is further discussed in Sec. 9.4.2. Figure 9-8 also illustrates
that the predictive model predicts the effect of inlet mole fraction very accurately.

The uncertainty of measurement on the mole fraction is ± 1%.

9.3.4 Effect of liquid column height

Regular dehumidifiers and steam condensers can be designed to have minimal pressure
drop (as low as a few hundred Pa). In bubble columns, a large percentage of the
pressure drop that occurs on the vapor side is due to the hydrostatic head of the
liquid in the column that the air-vapor mixture has to overcome. Thus, it is desirable
to keep the liquid height to a minimum value. The cooling coils must remain fully
immersed in the liquid pool, and hence, the minimum liquid height should be that
which just immerses the coils.

Figure 9-9 illustrates that there is no effect of reducing the liquid height from 10"
(254 mm) to 6" (152.4 mm - the minimum height at which the coil was fully immersed).
This can be understood by considering the length scale of the liquid circulation, which
we have postulated as the integral length of turbulence. As explained earlier in
Sec. 9.1.1, the integral length is very close to the bubble diameter. Hence, the scale
at which the circulation happens in the liquid is of the order of a millimeter which
is two orders of magnitude lower than the liquid height. Therefore, unless the liquid
height is reduced to a few millimeters, it will not have an effect on the bubble column
performance. This is a very significant consideration when designing bubble column
dehumidifiers for systems which cannot take large gas side pressure drops, such as
the humidification dehumidification desalination (HDH) system [18, 19].

9.3.5 Comparison of model and experiments

We have seen that the predictive model estimates the effect of bubble diameter,
superficial velocity and inlet mole fraction of vapor on heat flux exchanged in a
Figure 9-9: Effect of liquid height on the total heat flux in the bubble column measured and evaluated at $V_g = 6.52 \text{ cm/s}$; $D_b = 4 \text{ mm}$; $\chi_m = 21\%$.

Figure 9-10: Parity plot of heat flux values evaluated by the model and that measured by experiments for various boundary conditions.
bubble column dehumidifier accurately. In Fig. 9-10 we present a comparison of the experimental data and the model for various boundary conditions in a parity plot. There is excellent agreement (within -20%), and as per our expectation, the model consistently underpredicts the heat flux. This is because we approximated the boundary layer inside the bubble to be of the order of the bubble radius itself (Sec. 9.1.2), which is clearly an overestimation of the associated thermal resistance.

9.3.6 Effect of bubble-on-coil impact

In consideration of the effect of all of the different parameters (described in the previous paragraphs), bubble-on-coil impact was avoided during experimentation (see Sec. 9.2) and neglected in the predictive model. In this section, we study the effect of impact. Figure 9-11 clearly shows the difference in bubble rise path in cases without and with bubble-coil impact. Predictably, this has a large effect on the heat rate in the device.

Figure 9-12 illustrates this effect. It may be observed that impact raises the heat transfer rates to significantly higher values. Thus, in case of the serpentine coils, a major portion of the heat communicated between the air-vapor bubbles and the coils is through direct impact between the two. Hence, to obtain higher heat transfer rates it is desirable to design coils to have maximum impact.

9.4 Effect of multi-staging

In a HDH system, the isothermal nature of the column liquid in the bubble column dehumidifier reduces the temperature to which seawater can be preheated to (in the coils). This limits the energy effectiveness of the device (for definition of energy effectiveness, see Chapter 3). A low effectiveness in the dehumidifier, reduces the HDH system performance significantly (see Chapter 4). In this section, we detail an innovation which increases the energy effectiveness of these devices.

A schematic diagram of a multi-stage bubble column is shown in Fig. 9-13. In this device, the moist air is sparged successively from the bottom-most (first) stage
Figure 9-11: Bubble rise path in circular (left) and serpentine (right) coils.

Figure 9-12: Effect of bubble-on-coil impact.
to the top-most (last) stage via a pool of liquid in each stage. The coolant enters the coil in the last stage and passes through the coil in each stage and leaves from the first stage. The condensate is collected directly from the column liquid in each stage.

Figure 9-13: Schematic diagram of multi-stage bubble column dehumidifier.

Figure 9-14 illustrates the temperature profiles in a single stage and multi-stage bubble column. In both cases, the moist air comes in fully saturated at a temperature of 353 K and leaves dehumidified at 310 K. In the process, the pool of liquid in the bubble column gets heated and in turn preheats the seawater going through the coil. In the single stage case, the coolant gets preheated to a temperature of 308 K only (limited by the air exit temperature of 310 K). This corresponds to a very low effectiveness of 30%. In the case of multi-stage bubble columns, the column liquid in each stage is at a different temperature limited by the temperature of the air passing through the respective stage. Hence, the outlet coolant temperature is only limited by the exit temperature of the air from the first stage. In this example, we see that
this reaches 348 K (40 K higher than the single stage case). This corresponds to an increase in effectiveness from 30% in the single stage to 92% in the multi-stage.

Figure 9-14: An illustration of the temperature profile in the bubble columns for (a) single stage and (b) multi-stage

9.4.1 Prototype

A prototype (shown in Fig. 9-15) has been constructed to experimentally demonstrate the increase in effectiveness. The main body of the column of the prototype is made of an acrylic cylinder 6.5" (16.5 cm) wide and 12" (30.5 cm) tall. Both ends of the
cylinder have a 9" (22.9 cm) by 9" (22.9 cm) x 2" (5 cm) thick polypropylene cover with all necessary features CNC milled on the blocks, ensuring there are no leaks through the sides of the column. Each block features a channel for the inlet and outlet of the condensing coil, a drain for collecting condensate, and a bolt circle for mounting the sparger plate. The system is held together by 17" (43.18 cm) lengths of 3/8" (0.953 cm) threaded rods. Air flows from the bottom end through the column while the coolant travels from the top condensing coil to the bottom. The coils are connected using half inch (nominal diameter) tubing that runs along the outside of the column.

![Prototype of multi-stage bubble column.](image)

Figure 9-15: Prototype of multi-stage bubble column.

The sparger plates are similar to the ones used in a single stage bubble column except that they are circular. Additionally the holes are kept within a 5" (12.7 cm) circle as to not interfere with the condensing coil fittings. Stages with smaller
condensing coils use sparger plates with greater number of holes because their column supports a lower liquid pool height. Copper coils were used to pass the coolant through. Table 9.2 shows the dimensions of coils. Setup A is a single stage, B is two stage, and C is three stage. In setup B and C, the first stage has the smallest coil because otherwise all heat transfer will occur in the first stage and make the device inefficient. Also, note that stage 2 and stage 3 coils use a spiral design (as seen in Fig. 9-15). Further details of the prototype construction is given by Lam [163]. The setup was instrumented using the same sensors used for the single stage experiments. The total uncertainty on the effectiveness of the device is small (< 1 %).

<table>
<thead>
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<td>1/4</td>
<td>50</td>
<td>Serpentine</td>
</tr>
<tr>
<td>B</td>
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<td>1/4</td>
<td>28</td>
<td>Serpentine</td>
</tr>
<tr>
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<tr>
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<td>3</td>
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<td>40</td>
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</tr>
</tbody>
</table>

Figure 9-16 illustrates the increase in effectiveness of the device with multistaging. The data presented here is for an air inlet temperature of 65°C, inlet relative humidity of 100%, a water inlet temperature of 25°C and a water-to-air mass flow rate ratio of 2.45. It can be seen that the energy effectiveness of the device is increased from around 54% at single stage to about 90% for the three stage device. This can be further increased with a fourth stage. Work is progress in this regard. Further, owing to the higher superficial velocity (because of smaller column diameter) than in the single stage experiment reported in previous sections, the heat fluxes were much higher (up to 25 kW/m²). Also, the total gas side pressure drop of this device was 800 Pa.
Figure 9-16: Effect of multistaging the bubble column on energy effectiveness of the device.
9.4.2 Comparison with existing devices

A state-of-the-art dehumidifier (which operates in the film condensation regime) procured from George Fischer LLC was found to yield a maximum heat flux of 1.8 kW/m² (as per the design specification) compared to a maximum of 25 kW/m² obtained in the bubble column dehumidifier, demonstrating the superior performance of the novel device. This comparison was carried out at the same inlet conditions for the vapor-air mixture and the coolant streams. Also, the streamwise temperature differences were similar in both the cases. Further, the energy effectiveness of a three-stage bubble column dehumidifier was found to be similar to the conventional dehumidifier mentioned here.

9.5 Chapter conclusions

This chapter has proposed a novel bubble column dehumidifier for high rates in condensation of moist air mixtures with up to 90% non-condensables in them. The main conclusions are as follows.

1. Bubble column dehumidifiers have an order of magnitude better performance than existing state-of-the-art dehumidifiers operating in the film condensation regime.

2. The bubble column should be designed for high superficial velocity, low bubble diameter, and maximum bubble-on-coil impact. In order to minimize pressure drop, the liquid height can be kept to a minimum such that the coil is entirely submerged in the liquid. This is possible because the height has no effect on the performance of the device if it is greater than the bubble diameter (≈4-6 mm).

3. The inlet mole fraction of the vapor is found to have a weaker effect on the performance of the device than in a regular dehumidifier (in which the performance deteriorates quadratically with the vapor mole fraction at low values of mole fraction).
4. A physics-based predictive heat transfer model based on a thermal resistance circuit to estimate heat flux and temperature profiles in the bubble column condenser has been developed. The experimental data is underpredicted by a maximum of 20%. The model accurately predicts the effects of the various parameters on heat flux without incorporating any adjustable parameters.

5. A three-stage bubble column with a manageable air side pressure drop of $< 1$ kPa, an high effectiveness of 92% and a very high heat rate of 25 kW/m$^2$ was constructed at a fraction of the cost of a regular dehumidifier operating in the film condensation regime.

Implementation of the novel dehumidifier described in this chapter for application in HDH systems will reduce the capital cost of the system leading to a reduced cost of water production. The volume is reduced to 1/18$^{th}$ of the regular dehumidifier.
Chapter 10

Conclusions

Small-scale desalination systems can contribute to solution of the global water scarcity problem by facilitating community-level, easily-scalable water supply. The humidification dehumidification (HDH) technology is a carrier-gas-based thermal desalination technique ideal as a small-scale system. However, existing state-of-the-art HDH systems have a high cost of water production (about 30 $/m^3 of pure water produced). This thesis has made fundamental contributions to the thermal design of simultaneous heat and mass exchange (HME) devices and the HDH systems which can lead to affordable HDH systems. These contributions are summarized into two classes: (1) component design; and (2) system design.

10.1 Component design

1. A new definition for energy effectiveness, which can be applied to all types of HMEs, has been developed. It is based on the total enthalpy change of each fluid stream participating in the transfer processes. This parameter has enabled control volume based modeling of the humidifier and the dehumidifier of the HDH system.

2. Further, a novel “enthalpy pinch” has been defined for combined heat and mass exchange devices, analogous to the temperature pinch traditionally defined for
heat exchangers. Enthalpy pinch ($\Psi$) combines stream-to-stream temperature and humidity ratio differences, and it is directly related to the effectiveness of the device. This parameter has enabled thermal designs of HME devices and HDH systems with continuous or discrete extractions and injections.

3. A new non-dimensional parameter called the ‘modified heat capacity rate ratio’ (HCR) has been shown to be the linchpin in thermal design of HME devices (including the humidifier and the dehumidifier). The thermally balanced state for HME devices (with minimum entropy generation) has been found to be at a HCR of unity for both on-design (fixed-hardware) and off-design (fixed-performance) conditions. The closed form equations for zero ‘remanent’ irreversibility designs with continuous mass extraction or injection in HME devices have been reported in this thesis.

4. A pilot scale packed-bed humidifier unit with an enthalpy pinch of 14.8 kJ/kg dry air and a temperature pinch of 2.8°C has been built, and used to experimentally validate the aforementioned theory for HME devices.

5. Existing dehumidifiers which operate in the film condensation regime are found to have extremely low heat and mass transfer rates. A novel multi-stage bubble column dehumidifier with a heat rate of 25 kW/m² (an order of magnitude higher than regular dehumidifiers), an effectiveness of 92% at a pressure drop of less of < 1 kPa has been built at MIT. These devices can be constructed at a fraction of the cost of existing dehumidifiers.

10.2 HDH system design

1. Various HDH systems were analyzed in detail, and it was found that the optimum energy performance (GOR) of these systems occurs at a modified heat capacity rate ratio of unity for either the humidifier or the dehumidifier depending on the mode of energy input. In water heated systems, the optimum occurred at $\text{HCR}=1$ for the dehumidifier. This was shown both experimentally
and using on-design simulations. Further, on-design models showed that for mechanical compression driven, varied pressure systems the optimum occurs at a HCR = 1 for the humidifier.

2. The aforementioned understanding was applied to develop thermal design algorithms for thermodynamic balancing in HDH systems with mass extractions and injections. For an HDH system with ‘infinitely’ sized ideal components (zero enthalpy pinch) and continuous mass extraction and injection, the GOR was found to be about 89% of the reversible limit. This is the upper limit of the HDH system performance. Also, it is found that thermodynamic balancing is effective in HDH only when the HME devices have an appropriately low enthalpy pinch ($\Psi \leq 27$ kJ/kg dry air).

3. A pilot-scale HDH system producing a maximum of 700 liters of water a day was constructed and used to validate the aforementioned algorithms. It was found that this system had a peak GOR of 4 with a single air extraction and injection from the humidifier to the dehumidifier.

4. Energy consumption in HDH systems can be further reduced by treating pressure as a variable parameter. The concept of mechanical compression driven, sub-atmospheric pressure HDH systems (with the humidifier at a lower pressure than the dehumidifier) which follow a reverse-Brayton-like desalination cycle was developed. This system has been theoretically demonstrated using effectiveness based on-design models to operate at a thermal energy consumption of less than $100 \text{ kWh}$ per cubic meter (GOR = 6). Pilot scale experiments are being designed for construction in Saudi Arabia based on the aforementioned findings.

5. Further, designs of thermal compression driven HDH systems hybridized with RO were developed. It was found that these systems have a high GOR of up to 20. More research needs to be done to investigate the applicability of these steam driven units to community level water treatment.
All the science developed in this thesis can make HDH systems affordable\textsuperscript{1} for application in small scale drinking water systems in the developing world.

\textsuperscript{1}The water cost was estimated at $< 5 \$/m^3$. See Appendix D.
References


254


Appendix A

Calculation of reversible GOR for HDH

The highest GOR achievable in a cycle of this type will be that for zero entropy production. We derive here the expression for this upper limit. Applying 1st and 2nd law to the system shown in Fig. A-1,

\[ \dot{Q}_{ht} + (\dot{m}_1) = (\dot{m}_2 + \dot{m}_3) \]  
\[ \frac{\dot{Q}_{ht}}{T_h} + (\dot{m}_1) + \dot{S}_{gen} = (\dot{m}_2 + \dot{m}_3) \]

Figure A-1: Schematic diagram for calculating Carnot GOR of HDH
Using these equations,
\[
\dot{Q}_{ht} - \frac{\dot{Q}_{ht} T_0}{T_h} - \dot{S}_{gen} \cdot T_0 = \dot{n}_2(h - T_0 \delta)_2 + \dot{n}_3(h - T_0 \delta)_3 - \dot{n}_1(h - T_0 \delta)_1
\]
\[
= \frac{\dot{n}_2 \cdot g_2 + \dot{n}_3 \cdot g_3 - \dot{n}_1 \cdot g_1}{\frac{T_0}{T_h}} \quad \text{(A.4)}
\]
\[
\dot{Q}_{ht} = \frac{\dot{n}_2 \cdot g_2 + \dot{n}_3 \cdot g_3 - \dot{n}_1 \cdot g_1 + T_0 \cdot \dot{S}_{gen}}{1 - \frac{T_0}{T_h}} \quad \text{(A.5)}
\]

The least heat of separation is for \( \dot{S}_{gen} = 0 \),
\[
(\dot{Q}_{ht,\text{least}}) = \frac{\dot{n}_2 \cdot g_2 + \dot{n}_3 \cdot g_3 - \dot{n}_1 \cdot g_1}{1 - \frac{T_0}{T_h}} \quad \text{(A.6)}
\]

The following conditions are assumed for calculating the least heat and maximum theoretical GOR for HDH:

1. For HDH the recovery ratio is typically <10%. Here for the sake of calculation it is taken as 10%.

2. The inlet feed stream salinity is taken as 35,000 ppm and is approximated by a 0.62 mol/kg NaCl solution.

3. Pressure is 1 bar.

4. Top and bottom temperatures (\( T_0 \) and \( T_h \) respectively) are taken as 90°C and 30°C respectively.

5. The calculation is performed for water production of 1 kg/s.

6. The air stream enters and leaves at the same temperature and humidity. State 4 and 5 in Fig. A-1 are the same. \( T_{a,\text{out}} = T_{a,\text{in}} \) and \( \omega_{\text{out}} = \omega_{\text{in}} \).

At these conditions,
\[
\text{GOR}_{rev} = \frac{h_{fg} \cdot \left(1 - \frac{T_0}{T_h}\right)}{RR \cdot g_2 + (1 - RR) \cdot g_3 - g_1} \quad \text{(A.7)}
\]
\[
= 122.5 \quad \text{(A.8)}
\]

264
Appendix B

Entropy generation in a counterflow cooling tower

Using the control volume in Fig. 3-3,

\[ \dot{S}_o = \dot{m}_{w,o} \cdot s_{w,o} + \dot{m}_{a,o} \cdot s_{a,o} \]  \hspace{1cm} (B.1)

\[ \dot{S}_i = \dot{m}_{w,i} \cdot s_{w,i} + \dot{m}_{a,i} \cdot s_{a,i} \]  \hspace{1cm} (B.2)

Applying the Second Law to the same isolated control volume,

\[ \dot{S}_{gen} = \dot{S}_o - \dot{S}_i \]  \hspace{1cm} (B.3)

\[ = \dot{m}_{w,i} \cdot (s_{w,o} - s_{w,i}) - (\dot{m}_{w,i} - \dot{m}_{w,o}) \cdot s_{w,o} \]

\[ + \dot{m}_{a,o} \cdot s_{a,o} - \dot{m}_{a,i} \cdot s_{a,i} \geq 0 \]  \hspace{1cm} (B.4)

Moist air entropy can be written as:

\[ \dot{m}_{a,o} \cdot s_{a,o} = \dot{m}_{da} \cdot s_{da,o} + \dot{m}_{da} \cdot \omega_o \cdot s_{w,o} \]  \hspace{1cm} (B.5)

\[ \dot{m}_{a,i} \cdot s_{a,i} = \dot{m}_{da} \cdot s_{da,i} + \dot{m}_{da} \cdot \omega_i \cdot s_{w,i} \]  \hspace{1cm} (B.6)
Now Eqn. B.4 can be written as:

\[
\dot{S}_{gen} = \dot{m}_{w,i} \cdot (s_{w,o} - s_{w,i}) - (\dot{m}_{w,i} - \dot{m}_{w,o}) \cdot s_{w,o} \\
+ \dot{m}_{da} \cdot (s_{da,o} - s_{da,i}) + \dot{m}_{da} \cdot \omega_i \cdot (s_{v,o} - s_{v,i}) \\
+ \dot{m}_{da} \cdot (\omega_o - \omega_i) \cdot s_{v,o} + \Delta \dot{S}_{mix,out} - \Delta \dot{S}_{mix,in}
\]  

(B.7)

where \(\Delta \dot{S}_{mix}\) is the entropy of mixing of dry air and water vapor mixture. It should also be noted that entropy of individual components \((s_{da,o}, s_{da,i}, s_{v,o}, \text{and } s_{v,i})\) are all evaluated at the respective moist air temperature and total pressure. A mass balance gives

\[
\dot{m}_{w,i} - \dot{m}_{w,o} = \dot{m}_{da} \cdot (\omega_o - \omega_i)
\]  

(B.8)

This further reduces Eqn. B.7 to give

\[
\dot{S}_{gen} = \dot{m}_{w,i} \cdot (s_{w,o} - s_{w,i}) - \dot{m}_{da} \cdot (\omega_o - \omega_i) \cdot (s_{w,o} - s_{v,o}) \\
+ \dot{m}_{da} \cdot (s_{da,o} - s_{da,i}) + \dot{m}_{da} \cdot \omega_i \cdot (s_{v,o} - s_{v,i}) \\
+ \Delta \dot{S}_{mix,out} - \Delta \dot{S}_{mix,in}
\]  

(B.9)

Water is assumed as an incompressible fluid, and the total pressure of the system is taken to be uniform. Assuming that dry air and water vapor exhibit ideal gas behavior, we have:

\[
\dot{S}_{gen} = \dot{m}_{w,i} \cdot \left[ c_{p,w} \cdot \ln \left( \frac{T_{w,o}}{T_{w,i}} \right) \right] \\
- \dot{m}_{da} \cdot (\omega_o - \omega_i) \cdot \left[ c_{p,v} \cdot \ln \left( \frac{T_{w,o}}{T_{a,o}} \right) \right] \\
+ \dot{m}_{da} \cdot \omega_i \cdot \left[ R_{w} \cdot \ln \left( \frac{P_{sat,w,o}}{P_{total}} \right) + s_{f,g,o} \right] \\
+ \dot{m}_{da} \cdot c_{p,da} \cdot \ln \left( \frac{T_{a,o}}{T_{a,i}} \right) \\
+ \dot{m}_{da} \cdot \omega_i \cdot c_{p,v} \cdot \ln \left( \frac{T_{a,o}}{T_{a,i}} \right) \\
+ \Delta \dot{S}_{mix,out} - \Delta \dot{S}_{mix,in}
\]  

(B.10)
where, $P_{\text{sat},w,o}$ is the saturation pressure of water at $T_{w,o}$ and $s_{fg,o}$ is the enthalpy change of vaporization at $T_{w,o}$. The entropy of mixing for an ideal mixture can be written as:

$$
\Delta \dot{S}_{\text{mix,out}} - \Delta \dot{S}_{\text{mix,in}} = -R_{da} \cdot \dot{m}_{da} \cdot \ln(x_{da,o})
- R_{w} \cdot \dot{m}_{v,o} \cdot \ln(x_{v,o})
+ R_{da} \cdot \dot{m}_{da} \cdot \ln(x_{da,i})
+ R_{w} \cdot \dot{m}_{v,i} \cdot \ln(x_{v,i})
$$

(B.11)

Here the mole fraction of dry air and water vapor can be expressed in terms of humidity ratio, $\omega$, as:

$$
\frac{1 - x_{da}}{x_{da}} = \frac{M_{da}}{M_{v}} \cdot \omega
= 1.608 \cdot \omega 
$$

(B.12)

$$
x_{da} = \frac{1}{1 + 1.608 \cdot \omega}
$$

(B.13)

$$
x_{v} = \frac{1}{1 + 1.608 \cdot \omega}
$$

(B.14)

Using these expressions for mole fraction in Eqn. B.10 we get the following.

$$
\Delta \dot{S}_{\text{mix,out}} - \Delta \dot{S}_{\text{mix,in}} = \dot{m}_{da} \left\{ R_{da} \cdot \ln \left( \frac{1 + 1.608 \cdot \omega_{o}}{1 + 1.608 \cdot \omega_{i}} \right)
- R_{w} \cdot \omega_{o} \cdot \ln \left( 1 + \frac{1}{1.608 \cdot \omega_{o}} \right)
+ R_{w} \cdot \omega_{i} \cdot \ln \left( 1 + \frac{1}{1.608 \cdot \omega_{i}} \right) \right\}
$$

(B.15)

Using Eqns. B.10 and B.15, we obtain Eqn. 3.45.
Appendix C

Algorithms for modeling HDH systems with and without thermodynamic balancing
Figure C-1: Flowchart of the overall HDH system design for the no extractions case.
Estimate the total heat duty $\Delta h^*$ by assuming parallel temperature profiles for dehumidifier and humidifier

Plot saturated air temperature profile $T_s(h^*)$

Plot dehumidifier temperature profile $T_d(h^*)$

Divide total enthalpy range into small equal intervals

Calculate $\Delta S$ (per kg of water)

Calculate $\Delta s$ (per kg of dry air)

Calculate mass flow rate ratio for humidifier stream

Calculate $\Delta m_w$

Calculate mass flow rate of water produced in interval

Calculate mass flow rate of water stream in humidifier in following interval

Calculate salinity of water stream in humidifier

Calculate specific heat of water stream in humidifier

Calculate slope of humidifier temperature profile

Calculate lower temperature of water stream in humidifier in the interval

Repeat process for all intervals and generate the humidifier temperature profile

Define state A with $h_a=h_{ma}(T_{ma})+\Psi$ and $T_A=T_{ma}(h_a)$

Define state B for water stream in humidifier with $h_b=h_{mb}(T_{mb})$ and $T_B$ determined from humidifier temperature profile at $h_b$

Shift entire humidifier temperature profile upwards by $\Delta T=T_c-T_B$

Calculate total entropy generation

Calculate heat input

Calculate gained output ratio

Figure C-2: Flowchart of the overall system design for the continuous air extractions case.
Figure C-3: Flowchart of the overall system design for the single air extraction case.
Appendix D

Cost of water production in HDH system proposed in this thesis

The cost of water production is calculated by a standard method used in the desalination industry [164, 165].

Figure D-1 illustrates a three-dimensional model of a trailer-mounted, single air extraction HDH system operating under sub atmospheric pressure with a 12 foot (3.6 m) tall packed bed humidifier and a four-stage bubble column dehumidifier. All the details of the layout design was performed by Jeffery Huang [166]. This unit is designed to produce 10 m$^3$ per day. We calculate the cost of this unit as cost of the state-of-the-art HDH system.

The total thermal energy consumed by this system is 156 kWh$_{th}$ per cubic meter of water produced and the electrical energy consumption is 1.2 kWh per cubic meter of water produced (see Chapters. 5 and 6). The thermal energy is provided from compressed natural gas tanks on the trailer at an assumed cost of $4 per 1000 cubic feet (this is the average price in India [167]). The electrical energy is provided using an diesel generator at the rate of $0.20 per kWh. The total energy cost per m$^3$ of water produced is $2.17.

The capital cost is the sum of the various costs listed in Table D.1. These costs are obtained from different manufacturers. The parts for the humidifier and dehumidifier are to be obtained from these manufacturers and the assembly and fabrication is to
Figure D-1: Three-dimensional model of a trailer-mounted, sub-atmospheric-pressure, natural-gas-fired, single air extraction HDH system with a 12 foot tall packed bed humidifier and a four-stage bubble column dehumidifier.

be done by sub-contractors.

The total capital cost is amortized over a life of 20 years. The amortization factor (CAF) is calculated as follows.

\[ CAF = \frac{I}{1 - (1 + I)^{-20}} \]  \hspace{1cm} (D.1)

where, \( I \) is the interest rate (taken to 6\% in this calculation). Annual amortization is the product of CAF and CAPEX. This comes to \$ 4,266.38. It is widely accepted in desalination industry that fixed O & M is a maximum of 5\% of the CAPEX. The total annual levelized cost is the sum of the annual amortization and annual fixed cost divided by the total amount of water produced in a year. This comes to 2.30 \$ per cubic meter of water produced. Further, we also assume that the plant is available for 90\% of the time only, which is reasonable for thermal desalination systems [164].
Hence the total cost of water is $4.91 per cubic meter of water produced\(^1\). This is much lower than the $15 to $60 per m\(^3\) that has been reported for Membrane Distillation Technology (which is the other promising small-scale thermal desalination system) \([168]\). It is also much lower than the costs reported for other HDH system (see Sec. 2.4).

Table D.1: Various components of capital expenditure (CAPEX) for a 10 m\(^3\) per day HDH system.

<table>
<thead>
<tr>
<th>Component</th>
<th>Cost [$]</th>
</tr>
</thead>
<tbody>
<tr>
<td>Vacuum pumps</td>
<td>1,250.00</td>
</tr>
<tr>
<td>Column containers</td>
<td>2,500.00</td>
</tr>
<tr>
<td>Pumps</td>
<td>1,600.00</td>
</tr>
<tr>
<td>Blowers</td>
<td>2,000.00</td>
</tr>
<tr>
<td>Dehumidifier</td>
<td>8,000.00</td>
</tr>
<tr>
<td>Humidifier</td>
<td>8,000.00</td>
</tr>
<tr>
<td>NG combustor</td>
<td>1,500.00</td>
</tr>
<tr>
<td>Generator</td>
<td>1,000.00</td>
</tr>
<tr>
<td>Sub contractor costs</td>
<td>15,000.00</td>
</tr>
<tr>
<td>Assembly</td>
<td>2,585.00</td>
</tr>
<tr>
<td>Controls</td>
<td>5,500.00</td>
</tr>
<tr>
<td><strong>Total</strong></td>
<td>48,935.00</td>
</tr>
</tbody>
</table>

\(^1\)This is based on conservative upper limit estimates for CAPEX and is bound to be lower as the advantages of economies of scale are realised.
Appendix E

Helium as carrier gas in HDH systems

In this appendix, we propose to solve the problems of low heat and mass transfer rates in the dehumidifier by changing the carrier gas from air to helium. The energy performance of HDH systems using helium relative to air is analysed in detail using novel on-design models for the components described in Chapter 3.

E.1 Rationale for selecting helium

In HDH systems, the properties of the carrier gas affect the overall cost of water production. While the psychrometric properties affect the thermodynamics and the energy consumption of the system, the thermophysical properties, like thermal conductivity, affect the heat transfer area required in the heat and mass exchange devices. In this section, we consider these properties for various gases which could possibly be used in HDH systems and outline the rationale for using helium in HDH systems.
Table E.1: Thermophysical properties of different carrier gases at STP and of dry saturated steam at atmospheric pressure

<table>
<thead>
<tr>
<th></th>
<th>M [g/mol]</th>
<th>k [W/m.K]</th>
<th>c_p [J/kg.K]</th>
<th>ρ [kg/m³]</th>
</tr>
</thead>
<tbody>
<tr>
<td>Air</td>
<td>28.97</td>
<td>0.02551</td>
<td>1.005</td>
<td>1.169</td>
</tr>
<tr>
<td>He</td>
<td>4.003</td>
<td>0.1502</td>
<td>5.193</td>
<td>0.1615</td>
</tr>
<tr>
<td>N₂</td>
<td>28.01</td>
<td>0.02568</td>
<td>1.038</td>
<td>1.13</td>
</tr>
<tr>
<td>Ar</td>
<td>39.95</td>
<td>0.01796</td>
<td>0.5203</td>
<td>1.611</td>
</tr>
<tr>
<td>CO₂</td>
<td>44.01</td>
<td>0.01657</td>
<td>0.8415</td>
<td>1.775</td>
</tr>
<tr>
<td>Steam</td>
<td>18.02</td>
<td>0.02503</td>
<td>2.043</td>
<td>0.5897</td>
</tr>
</tbody>
</table>

E.1.1 Thermophysical properties

Table E.1 shows all the important thermophysical properties for air, helium, nitrogen, argon and carbon dioxide at T=25°C and P=1 atm. Since these gases are to be used as a mixture with steam in the HDH systems, the properties of dry saturated steam at P=1 atm are also shown.

It is observed that helium has a much higher thermal conductivity than the other gases in consideration, about six times that of air. The thermal conductivity is especially important because a higher value will help reduce the gas side thermal resistance in the dehumidifiers.

The specific heat capacity is also much higher for helium than for the other gases. Specific heat affects the ratio of mass flow rate of seawater entering the system to that of the carrier gas circulated in the system. The performance of a HDH system is optimal when operated at a particular ratio of mass flow rates (Chapter 4). Hence, it is important to identify this ratio for each of the carrier gases at a given operating condition; this is done as part of the thermodynamic study later in the paper.

As can be seen from density of the gases, helium is by far the lightest gas and this may cause some operational difficulties. Dynamic viscosity (not shown) is about the same for the gases under consideration.

E.1.2 Psychrometric properties

Using Dalton's law and approximating the carrier gas and water vapor mixture as an ideal mixture we can write down the humidity ratio of the carrier gas as a function
Figure E-1: Psychrometeric chart for helium water vapor mixture and moist air.

of the mixture temperature ($T$), pressure ($P$), relative humidity ($\phi$) and molar mass ($M$) of the carrier gas.

$$\omega(T, P, \phi) = \frac{M_w}{M_{cg}} \cdot \frac{\phi \cdot p_{sat}(T)}{P - \phi \cdot p_{sat}(T)} \quad \text{(E.1)}$$

From this equation, it may be seen that the humidity ratio is higher for low molar mass gases. We know that helium has low molar mass compared to air (see Tab. E.1). Hence, the humidity ratio for helium is much higher than that for air (Fig. E-1). At any given temperature, total pressure, and relative humidity, the humidity ratio for helium is 7.23 times that of air. As we would expect based on Eq. E.1, this ratio is same as the ratio of the molar mass of helium to air.

The higher humidity ratio leads to a lower total mass flow of carrier gas per unit of water produced. This leads to a lower gas side pressure drop and a smaller auxiliary power consumption as explained in Sec. E.1.4.
E.1.3 Estimate of gas side heat transfer coefficient in the dehumidifier

In the dehumidifier of a HDH system the thermal resistance to heat transfer associated with the presence of non-condensables (air) is much higher than thermal resistance of the liquid film formed as a result of the condensation of steam from the moist air mixture [169]. In this section, an estimate of this thermal resistance using wealth of experimental data in literature is made [170, 171]. In particular, researchers from BARC (Bhabha Atmoic Research Center) in India [172] reviewed various experimental correlations that give the condensation heat transfer performance of steam in the presence of non-condensables (including air and helium) for simple geometries (e.g., a vertical tube). The correlations are generally of the form

\[ \text{Nu}_{\text{cgm}} = z_1 \cdot \text{Re}_{\text{cgm}}^{z_2} \cdot x_{\text{cgm}}^{z_3} \cdot \text{Ja}_{\text{cgm}}^{z_4} \]  \hspace{1cm} (E.2)

The Nusselt number is given as a function of the Reynolds number of the mixture, the bulk mole fraction of the carrier gas in the mixture and the Jakob number of the mixture.

\[ \text{Nu}_{\text{cgm}} = \frac{u \cdot D_h}{k_{\text{cgm}}} \]  \hspace{1cm} (E.3)

\[ \text{Re}_{\text{cgm}} = \frac{\rho_{\text{cgm}} \cdot v_{\text{mix}} \cdot D_h}{\mu_{\text{cgm}}} \]  \hspace{1cm} (E.4)

\[ \text{Ja}_{\text{cgm}} = \frac{c_{p,\text{cgm}} \cdot (T_{\text{bulk}} - T_{\text{wall}})}{h_f} \]  \hspace{1cm} (E.5)

The coefficients $z_1$ to $z_4$ will depend on the boundary conditions and the carrier gas. It is also common to normalize the Nusselt number for steam condensation in the presence of saturated helium by that in the presence of saturated air at a given temperature and absolute pressure (as shown in Eq. E.6).
\[ \Phi = \frac{Nu_{he}}{Nu_a} \] \hspace{1cm} (E.6)

To evaluate the Nusselt number and the heat transfer coefficient for gas side in the dehumidifier, the thermophysical properties of the carrier gas steam mixture are to be evaluated. The density, thermal conductivity and viscosity of gas mixtures at low pressures are evaluated using the following approximate expressions [173].

\[ \rho_{cgm} = \frac{\sum \rho_i x_i}{\sum x_i} \] \hspace{1cm} (E.7)

\[ k_{cgm} = \frac{\sum k_i x_i M_i^{1/3}}{\sum x_i M_i^{1/3}} \] \hspace{1cm} (E.8)

\[ \mu_{cgm} = \frac{\sum \mu_i x_i M_i^{1/2}}{\sum x_i M_i^{1/2}} \] \hspace{1cm} (E.9)

Using the appropriate correlations (internal flow with Re_{cgm} = 5000 \text{ - } 10000 \text{ and } Ja_{cgm} = 0.038) for Nusselt number [170, 172] and the property equations shown above, the ratio of heat transfer coefficients for condensation in the presence of helium and air are estimated in Table E.2. From this estimate it may be seen that the heat transfer coefficient can be increased by 5-6 times. This is a major advantage of using helium as the carrier gas in HDH systems.

Arabi & Reddy [174] had investigated the natural convection heat transfer coefficient in humidification dehumidification systems for various carrier gases for certain specific geometries. They had also observed that helium has the highest heat transfer coefficient among the gases considered (2 times higher than air for the cases they considered). The increase is lower compared to our estimate because they studied natural convection systems and the estimate presented earlier in this section is for forced convection systems.
Table E.2: Estimated improvement in gas side dehumidification heat transfer coefficients when helium is used as the carrier gas.

<table>
<thead>
<tr>
<th>T [°C]</th>
<th>$x_{cg}$ [-]</th>
<th>$k_{he}/k_a$ [-]</th>
<th>$\Phi$ [-]</th>
<th>$u_{he}/u_a$ [-]</th>
</tr>
</thead>
<tbody>
<tr>
<td>40</td>
<td>0.93</td>
<td>5.5</td>
<td>1.1</td>
<td>6.1</td>
</tr>
<tr>
<td>50</td>
<td>0.88</td>
<td>5.2</td>
<td>1.15</td>
<td>5.9</td>
</tr>
<tr>
<td>60</td>
<td>0.8</td>
<td>4.7</td>
<td>1.25</td>
<td>5.8</td>
</tr>
<tr>
<td>70</td>
<td>0.69</td>
<td>4.0</td>
<td>1.35</td>
<td>5.4</td>
</tr>
</tbody>
</table>

E.1.4 Gas side pressure drop in the dehumidifier

In this section, we estimate the gas side pressure drop for helium and air. The estimate is for a fixed flow geometry with a fixed hydraulic diameter and flow area. For these constraints, Shah [95] provides the following relationship of pressure drop on fluid properties and mass flow rate under turbulent flow conditions:

$$\Delta P_{cgm,D} \propto \frac{\mu_{cgm} \cdot \dot{m}_{cgm}^{1.8}}{\rho_{cgm}}$$  \hspace{1cm} (E.10)

The mass flow rate of carrier gas to be used in these equations is for the case to produce a fixed amount of water in the HDH systems. This is explained in the following sections. Using the mixture property equations (Eqs. E.7-E.9), the gas side pressure drop in the dehumidifier when using helium is found to be 1/5 to 1/8 times that when using air as a carrier gas. Along with the increase in heat transfer coefficient, the reduction in pressure drop clearly shows the potential of using helium as carrier gas. In Sec. E.3, we analyse the effect of reduction in pressure drop on the energy efficiency of the HDH systems.

E.2 Thermodynamic cycle for HDH desalination

The simplest HDH cycle is the water-heated, closed-air, open-water cycle. This cycle is described in great detail earlier in this thesis (see Chapters 2, 4 and 6). The analysis in this chapter is carried out for the same.
E.3 Relative thermodynamic performance of helium based cycles

The effect of changing the carrier gas from air to helium on the system performance is investigated in this section. HDH cycles are traditionally heat driven cycles run by using low grade energy to heat the seawater. The efficiency of the cycle itself is measured by the gained output ratio (GOR) defined in Eq. 2.2. The GOR for the water heated HDH cycle may be rewritten as follows

\[
\text{GOR} = \frac{\dot{m}_{pw} \cdot h_{fg}}{\dot{m}_{w} \cdot c_{pw} \cdot (T_{w,ht,out} - T_{w,ht,in})}
\]

From Chapters 3, 4, 5, and 7, we know that the modified heat capacity rate ratio (HCR) is the most important thermodynamic parameter for heat driven HDH cycles. Figure E-2 illustrates the GOR of cycles using air and helium as the carrier gas against the modified heat capacity ratio of the dehumidifier. For this example, the component effectivenesses are fixed along with the operating pressures and feed seawater conditions. The seawater temperature at the exit of the heater is fixed at 90°C.

It is observed that the change in carrier gas has very little impact on the performance of the system. To explain this trend let us rewrite Eq. E.11 as follows

\[
\text{GOR} = \frac{\Delta \omega \cdot h_{fg}}{\dot{m}_{r} c_{pw} \Delta T_{ht}}
\]

\[
= \frac{M_{w} \Delta x \cdot h_{fg}}{\dot{m}_{r} c_{pw} \Delta T_{ht}}
\]

where the mass flow rate ratio \( m_r = \frac{\dot{m}_{w}}{\dot{m}_{a}} \)

So, the ratio of GOR for system with helium and air can be written as follows:

\[
\frac{\text{GOR}_{he}}{\text{GOR}_{a}} = \left\{ \frac{M_{a} \Delta x_{D,he}}{M_{he} \Delta x_{D,a}} \right\} \cdot \left\{ \frac{m_{r,he} \Delta T_{ht,he}}{m_{r,a} \Delta T_{ht,a}} \right\}^{-1}
\]

283
Figure E-2: Relative performance of water heated HDH cycle with helium or air as carrier gas. $T_{\text{avg, in}} = 30^\circ\text{C}; T_{\text{cg, H, out}} = 90^\circ\text{C}; \varepsilon_H = 60\%; \varepsilon_D = 90\%; P = 100\text{ kPa}$.

Table E.3: Various system parameters and temperatures for cases shown in Fig. E-2

<table>
<thead>
<tr>
<th>$HCR_D$</th>
<th>$\Delta T_{ht, he}$</th>
<th>$\Delta T_{ht, a}$</th>
<th>$\frac{m_r, he}{m_r, a}$</th>
<th>$\Delta T_{D, he}$</th>
<th>$\Delta T_{D, a}$</th>
</tr>
</thead>
<tbody>
<tr>
<td>-</td>
<td>[°C]</td>
<td>[°C]</td>
<td>[-]</td>
<td>[°C]</td>
<td>[°C]</td>
</tr>
<tr>
<td>0.8889</td>
<td>31.73</td>
<td>31.81</td>
<td>7.041</td>
<td>20.3</td>
<td>20.49</td>
</tr>
<tr>
<td>1</td>
<td>29.56</td>
<td>29.65</td>
<td>7.049</td>
<td>26.59</td>
<td>26.8</td>
</tr>
<tr>
<td>1.333</td>
<td>31.87</td>
<td>31.97</td>
<td>7.074</td>
<td>30.66</td>
<td>30.94</td>
</tr>
<tr>
<td>1.667</td>
<td>34.26</td>
<td>34.35</td>
<td>7.084</td>
<td>32.67</td>
<td>32.98</td>
</tr>
<tr>
<td>2</td>
<td>36.55</td>
<td>36.63</td>
<td>7.087</td>
<td>33.23</td>
<td>33.52</td>
</tr>
</tbody>
</table>
From Tab.E.3, it can be seen that

\[
\frac{\Delta T_{ht,he}}{\Delta T_{ht,a}} \approx 1 \quad (E.15)
\]

\[
\frac{\Delta T_{D,he}}{\Delta T_{D,a}} \approx 1 \quad (E.16)
\]

\[
\frac{\Delta x_{D,he}}{\Delta x_{D,a}} \approx 1 \quad (E.17)
\]

\[
\frac{m_{r,he}}{m_{r,a}} \approx \frac{M_a}{M_{he}} \quad (E.18)
\]

Hence,

\[
\frac{\text{GOR}_{he}}{\text{GOR}_a} \approx 1 \quad (E.19)
\]

Thus the GOR is approximately the same for helium and air systems.

It was found from experiments described in Chapter 6 that specific work consumption was around 0.4 kWh/m³. For the same boundary conditions at a total pressure drop would be 1/5 to 1/8 of that of the experiment (see section E.1.4), a system with helium as the carrier gas will have an electricity consumption of < 0.1 kWh/m³.

### E.4 Chapter conclusions

1. Large pressure drops and low heat transfer coefficients in the gas side of the dehumidifier are significant problems for HDH systems. A possible solution to these problems is to use helium as the carrier gas instead of air. Owing to its superior thermophysical and psychrometric properties, helium as a carrier gas is estimated to significantly improve the heat transfer coefficient and lower the pressure drop compared to systems using air as a carrier gas.

2. The thermodynamic performance (GOR) of the water heated HDH system is not affected by changing the carrier gas.

3. However, the reduction of pressure drop will reduce the auxiliary electricity consumption by 5 to 8 times and increase in dehumidifier heat transfer coefficient will reduce dehumidifier size greatly.
4. Hence, given the advantages of lower heat exchanger size requirement and lower electricity consumption, it is concluded that using helium as the carrier gas has promise for HDH desalination systems. The current crisis of lack of availability of helium, however, poses a challenge to the commercial realization of such a system.