PERFORMANCE STUDY OF A HEAT PUMP ASSISTED DRYER SYSTEM FOR SPECIALTY CROPS

A Thesis Submitted to the College of Graduate Studies and Research in Partial Fulfillment of the Requirements for the Degree of Master of Science in the Department of Mechanical Engineering University of Saskatchewan Saskatoon

By
Phani Kumar Adapa
Spring 2001

© Copyright Phani Kumar Adapa, 2001. All rights reserved.
PERMISSION TO USE

In presenting this thesis in partial fulfillment of the requirements for a Postgraduate degree from the University of Saskatchewan, I agree that the Libraries of the University of Saskatchewan may make this thesis freely available for inspection. I further agree that permission for copying of this thesis in any manner, in whole or in part, for scholarly purpose may be granted by the professor or professors who supervised my thesis work or, in their absence, by the Head of the Department or Dean of the College in which my thesis work was done. It is understood that any copying or publication or use of this thesis or parts thereof for financial gain shall not be allowed without my written permission. It is also understood that due recognition shall be given to me and the University of Saskatchewan in any scholarly use which may be made of any material in my thesis.

Requests for permission to copy or to make other use of material in this thesis in whole or part should be addressed to:

Head of the Department of Mechanical Engineering
University of Saskatchewan
Saskatoon, Saskatchewan
S7N 5A9, Canada
ABSTRACT

This thesis is concerned with the technology of heat pump assisted drying of specialty crops, the benefits and potential of the process in the agriculture sector. The purpose of using a heat pump for drying of specialty crops is to dry them at lower temperature than would otherwise be possible using conventional drying techniques. Low temperature drying of specialty crops reduces the risk of loss in nutrient content and damage to physical properties which are important aspects considering their high commercial value. Heat pump drying is potentially more energy efficient since it is possible to recover latent heat from the humid dryer exhaust air.

In this research, cabinet and prototype continuous bed heat pump dryers were studied. Simulation models for the heat pump dryers were developed to predict the performance of the dryer, i.e. the drying rate of the material and psychrometric conditions of the air, and also to determine the refrigerant mass flow rate and corresponding temperatures at the condenser and evaporator coils of the heat pump system, based on the psychrometric conditions of process air inside the drying chamber. The accuracy of the predicted results is later verified with the experimental results.

Chopped alfalfa was dried in a cabinet dryer in batches and by emulating the continuous bed drying using two household dehumidifiers. The reason for using alfalfa instead of specialty crops like ginseng, herbs, echinacea, feverfew, etc. was that the material and its drying properties were readily available. Also the structure of alfalfa leaves and stems is similar to that of many herbs and specialty crops.
Results showed that alfalfa was dried from an initial moisture content of 70% (wb) to a final moisture content of 10% (wb). It was noticed that batch drying took about 4.5 h while continuous bed drying took 4 h to dry the material. The initial weight of alfalfa in each tray was 400 g. The average air velocity inside the dryer was 0.36 m/s. Low temperatures (30-45°C) for safe drying of specialty crops were achieved experimentally. Specific moisture extraction rate was maximum when relative humidity stayed above 40%. The household type dehumidifiers used in this study were about 50% more efficient in recovering the latent heat from the dryer exhaust compared to the conventional dryers. It was concluded that continuous bed drying is potentially a better option than batch drying because high process air humidity ratios at the entrance of the evaporator and constant moisture extraction rate and specific moisture extraction rate values can be maintained.

Simulation results for a prototype heat pump continuous bed dryer system suggested that the change in dryer inlet temperatures of the process air has an insignificant effect on drying of the material. Therefore, based on the results, it was concluded that the dryer inlet air temperature could be kept as low as 30°C, if required, to maintain product quality. The material was dried to a safe limit of 10% moisture content. The material mass flow rate was only 3-4.5 kg/h, indicating that it might be advisable to use the heat pump dryer in combination with some other technique.
ACKNOWLEDGEMENTS

I would like to express my deepest gratitude and sincere thanks to my supervisors, Dr. G.J. Schoenau and Dr. S. Sokhansanj for their excellent guidance, encouragement and support throughout my research work. I also appreciate their eagerness to help and the moral support they gave me during this period. Their advice and input during the preparation of this thesis are greatly appreciated.

I would also like to express my appreciation to my academic committee members Dr. R.W. Besant and Dr. D. Sumner for their advice and positive criticism during this project work. Likewise, I would like to thank Mr. W. Crerar, Mr. D. Deutscher, Mr. W.A. Morley and Mr. L.L. Roth for their technical support and assistance during the experimentation.

I also express my gratitude to Dr. L. Tabil Jr., Department of Agricultural and Bioresource Engineering, University of Saskatchewan, for agreeing to serve as my external examiner.

I acknowledge financial support from my supervisors and the Department of Mechanical Engineering during my studies and research.
I dedicate this work to my parents,

Dr. Prabhakara Gandhi Adapa and Mrs. Bala Adapa
TABLE OF CONTENTS

TITLE PAGE.................................i
PERMISSION TO USE............................ii
ABSTRACT..................................iii
ACKNOWLEDGEMENTS............................v
DEDICATION..................................vi
TABLE OF CONTENTS..........................vii
LIST OF TABLES................................x
LIST OF FIGURES............................xi
LIST OF ABBREVIATIONS.........................xiii

1. INTRODUCTION..............................1
   1.1 Background................................1
   1.2 Review of Heat Pump Drying..............3
   1.3 Research Objectives......................7
   1.4 Overview of the Thesis...................8

2. DRYER AND HEAT PUMP MODELS.............10
   2.1 Dryer Model – Thin Layer................10
      2.1.1 Drying Rate Equation................13
      2.1.2 Mass Balance Equation...............14
      2.1.3 Energy Balance Equation.............15
      2.1.4 Heat Transfer Rate Equation........16
   2.2 Psychrometric Conditions of Process Air in a Dryer.........18
   2.3 Thermodynamics of a Heat Pump...........20
   2.4 Heat Pump Model..........................22
      2.4.1 Overall Heat Transfer Coefficient at Condenser Coil.....25
         2.4.1.1 Condenser Coil Convective Heat Transfer Coefficient – Refrigerant Side..26
## LIST OF TABLES

<table>
<thead>
<tr>
<th>Table</th>
<th>Title</th>
<th>Page</th>
</tr>
</thead>
<tbody>
<tr>
<td>3.1</td>
<td>Heat Pump Dehumidifier Specifications</td>
<td>50</td>
</tr>
</tbody>
</table>
# LIST OF FIGURES

<table>
<thead>
<tr>
<th>Figure</th>
<th>Title</th>
<th>Page</th>
</tr>
</thead>
<tbody>
<tr>
<td>1.1</td>
<td>Basic Heat Pump Dryer System</td>
<td>2</td>
</tr>
<tr>
<td>2.1</td>
<td>Schematic Diagram of a Continuous Bed Cross Flow Dryer</td>
<td>11</td>
</tr>
<tr>
<td>2.2</td>
<td>Elemental Volume of the Continuous Bed Cross Flow Dryer</td>
<td>12</td>
</tr>
<tr>
<td>2.3</td>
<td>Schematic Diagram of Heat Pump Dryer System</td>
<td>18</td>
</tr>
<tr>
<td>2.4</td>
<td>Ideal Psychrometric Chart for the Dryer</td>
<td>19</td>
</tr>
<tr>
<td>2.5</td>
<td>Thermodynamic Heat Pump Cycle</td>
<td>21</td>
</tr>
<tr>
<td>2.6</td>
<td>Actual Heat Pump Cycle</td>
<td>23</td>
</tr>
<tr>
<td>2.7</td>
<td>Ideal Heat Pump Cycle</td>
<td>24</td>
</tr>
<tr>
<td>2.8</td>
<td>Modeled Pressure-Enthalpy Diagram for Refrigeration Cycle</td>
<td>39</td>
</tr>
<tr>
<td>2.9</td>
<td>Psychrometric Chart Showing Process Air Conditions Inside the Dryer</td>
<td>41</td>
</tr>
<tr>
<td>3.1</td>
<td>Block Diagram of the Experimental Heat Pump Dryer System</td>
<td>49</td>
</tr>
<tr>
<td>3.2</td>
<td>Overall View of Experimental Heat Pump Dryer System</td>
<td>49</td>
</tr>
<tr>
<td>3.3</td>
<td>Condenser and Evaporator Coils Setup in the Dehumidifier</td>
<td>50</td>
</tr>
<tr>
<td>4.1</td>
<td>Measured Temperature and Relative Humidity of Process Air at the Dryer Entrance and Exit</td>
<td>58</td>
</tr>
<tr>
<td>4.2</td>
<td>Measured Refrigerant Temperature at Condenser and Evaporator Coils Over Material Drying Time</td>
<td>60</td>
</tr>
<tr>
<td>4.3</td>
<td>Measured Moisture Condensation Rate for Batch and Continuous Bed Drying</td>
<td>61</td>
</tr>
<tr>
<td>4.4</td>
<td>Specific Moisture Extraction Rate and Power Consumed by Heat Pumps Over the Drying Process</td>
<td>63</td>
</tr>
<tr>
<td>4.5</td>
<td>Comparison of Measured and Simulation Temperatures of Refrigerant at Condenser and Evaporator Coils</td>
<td>64</td>
</tr>
<tr>
<td>Section</td>
<td>Title</td>
<td>Page</td>
</tr>
<tr>
<td>---------</td>
<td>-----------------------------------------------------------------------</td>
<td>------</td>
</tr>
<tr>
<td>4.6</td>
<td>Uncertainty Analysis for the Experimental Temperatures Obtained at the Condenser Coil</td>
<td>65</td>
</tr>
<tr>
<td>4.7</td>
<td>Uncertainty Analysis for the Experimental Temperatures Obtained at the Evaporator Coil</td>
<td>65</td>
</tr>
<tr>
<td>4.8</td>
<td>Moisture Extraction Rate for Experimental and Simulation Results Over Material Drying Time</td>
<td>67</td>
</tr>
<tr>
<td>4.9</td>
<td>Measured Psychrometric Condition of Process Air with air Bypass Around Evaporator Coils and Non-Adiabatic Drying (Figure not to Scale)</td>
<td>68</td>
</tr>
<tr>
<td>5.1</td>
<td>Overview of Prototype Cross-Flow Continuous Bed Heat Pump Dryer System</td>
<td>69</td>
</tr>
<tr>
<td>5.2</td>
<td>Schematic Sectioned View of the Counter Cross Flow Continuous Bed Drying Chamber</td>
<td>70</td>
</tr>
<tr>
<td>5.3</td>
<td>Psychrometric and Material Conditions at the two Elemental Volumes on Conveyors One and Two</td>
<td>71</td>
</tr>
<tr>
<td>5.4</td>
<td>Schematic Block Diagram of the Prototype Heat Pump System</td>
<td>73</td>
</tr>
<tr>
<td>5.5</td>
<td>Schematic Block Diagram of the Prototype Heat Pump Dryer System</td>
<td>74</td>
</tr>
<tr>
<td>5.6</td>
<td>Predicted Affect of Material Mass Flow Rate on Moisture Content of the Material at Dryer Exit</td>
<td>76</td>
</tr>
<tr>
<td>5.7</td>
<td>Predicted Affect of Material Mass Flow Rate on Humidity Ratio of Material at Dryer Exit</td>
<td>78</td>
</tr>
<tr>
<td>5.8</td>
<td>Change in Temperature and Moisture Content of Material Along Conveyors One and Two of the Dryer</td>
<td>79</td>
</tr>
</tbody>
</table>
**LIST OF ABBREVIATIONS**

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>(A_c)</td>
<td>total surface area of condenser coil, (m^2)</td>
</tr>
<tr>
<td>(A_{dsh})</td>
<td>area of desuperheating section of condenser coil, (m^2)</td>
</tr>
<tr>
<td>(A_f)</td>
<td>total condenser fin area, (m^2)</td>
</tr>
<tr>
<td>(A_F)</td>
<td>total evaporator fin area, (m^2)</td>
</tr>
<tr>
<td>(A_{oc})</td>
<td>surface area of the condenser coil, (m^2)</td>
</tr>
<tr>
<td>(A_{oe})</td>
<td>surface area of the evaporator coil, (m^2)</td>
</tr>
<tr>
<td>(A_{pie})</td>
<td>inside evaporator pipe area, (m^2)</td>
</tr>
<tr>
<td>(A_{po})</td>
<td>outside pipe evaporator area, (m^2)</td>
</tr>
<tr>
<td>(A_{ti})</td>
<td>inside condenser tube area, (m^2)</td>
</tr>
<tr>
<td>(A_{to})</td>
<td>outside condenser tube area, (m^2)</td>
</tr>
<tr>
<td>(AMTD)</td>
<td>arithmetic mean temperature difference</td>
</tr>
<tr>
<td>(ASHRAE)</td>
<td>American Society of Heating Refrigerating and Air-conditioning Engineers</td>
</tr>
<tr>
<td>(b_{wm})</td>
<td>coefficient evaluated at water film temperature, (kJ/kg\cdot{}^\circ{}C)</td>
</tr>
<tr>
<td>(C)</td>
<td>capacity rate ratio, (C_{\text{min}}/C_{\text{max}})</td>
</tr>
<tr>
<td>(C_{\text{min}})</td>
<td>smaller capacity rate ((m_r C_{pr}) or (m_o C_{pa}) whichever is smaller), (kJ/sK)</td>
</tr>
<tr>
<td>(\text{COP}_{\text{HP}})</td>
<td>Coefficient of performance of heat pump (dimensionless)</td>
</tr>
<tr>
<td>(\text{COP}_{\text{ref}})</td>
<td>Coefficient of performance of refrigerator (dimensionless)</td>
</tr>
<tr>
<td>(C_p)</td>
<td>specific heat, (J/kgK)</td>
</tr>
<tr>
<td>(C_{pa})</td>
<td>specific heat of air, (kJ/kgK)</td>
</tr>
<tr>
<td>(C_{pg})</td>
<td>specific heat of material, (kJ/kgK)</td>
</tr>
<tr>
<td>(C_{pl})</td>
<td>specific heat of liquid, (kJ/kgK)</td>
</tr>
</tbody>
</table>
$C_{pw}$  specific heat of water vapor, kJ/kgK

$C_{rdsh}$  capacity rate for refrigerant, $m, C_{pw}$, kJ/sK

$D_{eqv}$  equivalent fin diameter, m

$D_i$  inside coil diameter, m

$D_o$  coil outside diameter, m

$E$  enhancement factor

$G$  mass velocity, kg/m$^2$s

$G_a$  mass flow rate of air, kg/m$^2$s

$G_p$  mass flow rate of material, kg/m$^2$s

$H$  moist air enthalpy calculated at average air temperature, J/kg

$H$  enthalpy of saturated air, kJ/kg

$h_a$  convective heat transfer coefficient, W/m$^2$K

$h_{cow}$  convective heat transfer coefficient for outside surface, kW/m$^2$K

$h_{cv}$  heat transfer coefficient, kJ/m$^3$minK

$h_f$  forced convective heat transfer coefficient, W/m$^2$-K

$h_i$  inside (refrigerant) heat transfer coefficient, W/m$^2$-K

$h_{NcB}$  nucleate pool boiling coefficient, W/m$^2$-K

$h_o$  outside (air) heat transfer coefficient, W/m$^2$-K

$h_{ow}$  outside wet convective heat transfer coefficient, W/m$^2$-K

$H_{sR}$  moist air enthalpy calculated at average refrigerant temperature, J/kg

$H_{st}$  the assumed enthalpy of saturated moist air evaluated at tube temperature, kJ/kg

$i_{fg}$  latent heat of vaporization, J/kg
<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>$k$</td>
<td>drying rate constant, min&lt;sup&gt;-1&lt;/sup&gt;/h&lt;sup&gt;-1&lt;/sup&gt;</td>
</tr>
<tr>
<td>$k$</td>
<td>thermal conductivity of air, W/mK</td>
</tr>
<tr>
<td>$k_f$</td>
<td>thermal conductivity of fin, W/mK</td>
</tr>
<tr>
<td>$k_w$</td>
<td>thermal conductivity of water, W/mK</td>
</tr>
<tr>
<td>$L$</td>
<td>total length of coil, m</td>
</tr>
<tr>
<td>$L_g$</td>
<td>latent heat of vaporization of moisture from material, kJ/kg</td>
</tr>
<tr>
<td>LMTD</td>
<td>logarithmic mean temperature difference, °C</td>
</tr>
<tr>
<td>MR</td>
<td>moisture ratio (fraction)</td>
</tr>
<tr>
<td>$M$</td>
<td>moisture content at time $t$</td>
</tr>
<tr>
<td>$M$</td>
<td>molecular weight</td>
</tr>
<tr>
<td>$M_e$</td>
<td>equilibrium moisture content</td>
</tr>
<tr>
<td>$M_o$</td>
<td>initial moisture content</td>
</tr>
<tr>
<td>MER</td>
<td>moisture extraction rate, kg/h</td>
</tr>
<tr>
<td>$m_a$</td>
<td>mass flow rate of air, kg/s</td>
</tr>
<tr>
<td>$m_r$</td>
<td>mass flow rate of refrigerant, kg/s</td>
</tr>
<tr>
<td>$N_r$</td>
<td>number of rows</td>
</tr>
<tr>
<td>$Nu_{anr}$</td>
<td>Nusselt number for $N_r$ rows at AMTD</td>
</tr>
<tr>
<td>$Nu_{a3r}$</td>
<td>Nusselt number for 3 rows at AMTD</td>
</tr>
<tr>
<td>NTU</td>
<td>number of transfer units</td>
</tr>
<tr>
<td>$P$</td>
<td>atmospheric pressure, Pa</td>
</tr>
<tr>
<td>$P_{crit}$</td>
<td>critical pressure, Pa</td>
</tr>
<tr>
<td>$P_l$</td>
<td>longitudinal distance between rows, m</td>
</tr>
<tr>
<td>Symbol</td>
<td>Description</td>
</tr>
<tr>
<td>--------</td>
<td>-------------</td>
</tr>
<tr>
<td>Pr</td>
<td>Prandtl number for air</td>
</tr>
<tr>
<td>$P_{rp}$</td>
<td>reduced pressure = $P/P_{crit}$</td>
</tr>
<tr>
<td>$P_w$</td>
<td>vapor pressure, Pa</td>
</tr>
<tr>
<td>$P_{ws}$</td>
<td>saturated vapor pressure, Pa</td>
</tr>
<tr>
<td>$q_1$</td>
<td>energy supplied to maintain body B at higher temperature $T_1$, J</td>
</tr>
<tr>
<td>$q_2$</td>
<td>energy extracted from the surroundings at temperature $T_2$, J</td>
</tr>
<tr>
<td>RH</td>
<td>relative humidity, %</td>
</tr>
<tr>
<td>S</td>
<td>suppression factor</td>
</tr>
<tr>
<td>SMER</td>
<td>specific moisture extraction rate, kg/kWh</td>
</tr>
<tr>
<td>t</td>
<td>time, min</td>
</tr>
<tr>
<td>T</td>
<td>temperature, K</td>
</tr>
<tr>
<td>$T_a$</td>
<td>temperature of air, °C</td>
</tr>
<tr>
<td>$t_{aci}$</td>
<td>temperature of air before entering the condenser coil, °C</td>
</tr>
<tr>
<td>$T_{ao}$</td>
<td>initial temperature of air, °C</td>
</tr>
<tr>
<td>$t_{as}$</td>
<td>temperature of saturated air, °C</td>
</tr>
<tr>
<td>$T_g$</td>
<td>temperature of agricultural material, °C</td>
</tr>
<tr>
<td>$T_{go}$</td>
<td>initial temperature of agricultural material, °C</td>
</tr>
<tr>
<td>$t_R$</td>
<td>refrigerant temperature at saturation, °C</td>
</tr>
<tr>
<td>$t_{rci}$</td>
<td>temperature of refrigerant at condenser inlet (or at compressor outlet), °C</td>
</tr>
<tr>
<td>$t_{rs}$</td>
<td>temperature of refrigerant at vapor saturation curve, °C</td>
</tr>
<tr>
<td>$t_t$</td>
<td>total time, min</td>
</tr>
<tr>
<td>$t_i$</td>
<td>tube temperature, °C</td>
</tr>
<tr>
<td>TPC</td>
<td>first predicted value of refrigerant temperature at the condenser, °C</td>
</tr>
</tbody>
</table>
TPE  first predicted value of refrigerant temperature at the evaporator, °C

$\Delta T_{LMTCond}$  log mean temperature difference taken at the condenser coil, °C

$T_1 \ & T_4$  temperature of process air at the condenser outlet and inlet (refer to Figures 2.3 and 2.4), °C

$U_{dsh}$  overall heat transfer coefficient for desuperheating section of condenser, W/m²K

$U_{oe}$  overall heat transfer coefficient at the evaporator coil, kg/m²/s

$U_{oc}$  overall heat transfer coefficient at the condenser coil, W/m²/K

$v$  velocity of refrigerant, m/s

$V_{fin}$  air velocity between fins for a tubeless heat exchanger, m/s

$W$  humidity ratio, kg/kg dry air

$W_e$  work input to the heat pump (compressor), W

$W_f$  fin spacing, m

$W_o$  initial humidity ratio, kg/kg

$W_1$  humidity ratio of air before entering the evaporator, kg/kg dry air

$W_2$  humidity ratio of air after passing through the evaporator, kg/kg dry air

$x$  thickness of the dryer bed, m

$X$  liquid mass fraction

$X_L$  longitudinal distance between coil rows, m

$X_T$  transverse distance between coil rows, m

$y$  material movement direction along the dryer bed

$Y_f$  fin thickness, m

$y_w$  thickness of water film, m

$z$  direction along the width of dryer bed
<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>$Z_{eq}$</td>
<td>equivalent fin height, m</td>
</tr>
<tr>
<td>$\beta$</td>
<td>ratio of tubeless cross-sectional area to fin surface area including tube area</td>
</tr>
<tr>
<td>$\varepsilon$</td>
<td>effectiveness</td>
</tr>
<tr>
<td>$\mu$</td>
<td>viscosity, Ns/m$^2$</td>
</tr>
<tr>
<td>$\mu_a$</td>
<td>air viscosity, Ns/m$^2$</td>
</tr>
<tr>
<td>$\rho$</td>
<td>density of refrigerant, kg/m$^3$</td>
</tr>
<tr>
<td>$\rho_a$</td>
<td>air density, kg/m$^3$</td>
</tr>
<tr>
<td>$\phi$</td>
<td>surface heat flux, W/m$^2$</td>
</tr>
<tr>
<td>$\phi_{eff}$</td>
<td>fin efficiency</td>
</tr>
</tbody>
</table>
Chapter 1

INTRODUCTION

1.1 Background

Specialty crops such as ginseng, herbs, etc., need to be dried at low temperatures for product optimization. This is an important consideration as they have a relatively high commercial value. Heating ambient air to use for drying, although a simple cost-effective procedure, is of limited application, particularly at higher ambient air relative humidities, because of the low allowable maximum temperature conditions. High temperature drying deteriorates the material structure and can render it unsuitable for further use. Heat pump dryers operating under closed loop conditions have the potential to dry these crops independent of ambient conditions. Under high ambient moist air conditions, it may not even be possible to dry the material by conventional means.

A heat pump drying system is also potentially more energy efficient than conventional drying methods [1]. In conventional hot-air dryers, exhaust air is vented to the atmosphere and the energy is lost. In heat pump assisted dryers, both sensible and latent heat can be recovered from the dryer exhaust air, improving the overall thermal performance. Latent heat is recovered by dehumidification of process air at the evaporator. Dehumidification is achieved by cooling the vapor-air mixture below
the dew point temperature. The heat pump drying system is a combination of two sub-systems: a heat pump and a dryer.

The heat pump operates according to a basic air conditioning cycle involving four main components: the evaporator, the compressor, the condenser and the expansion valve, as shown in Figure 1.1.

![Figure 1.1: Basic heat pump dryer system](image)

The working fluid (refrigerant) at low pressure is vaporized in the evaporator by heat drawn from the dryer exhaust air. The compressor raises the enthalpy of the working fluid of the heat pump and discharges it as superheated vapor at high-pressure. Heat is removed from the working fluid and returned to the process air at the condenser. The working fluid is then throttled to the low-pressure line (using an expansion valve) and enters the evaporator to complete the cycle. It is the energy removed at the condenser that reduces the energy consumption of drying process.
The heat pump is unique in that it can recover heat from a low temperature stream and deliver that heat to a stream at a higher temperature while using energy from a different source (e.g. electricity) to perform the operation. A significant feature of a heat pump is that the amount of energy delivered at the higher temperature is greater than the energy input to drive the process. However, in a drying system this excess energy cannot be used for drying but would be available for other uses through heat exchangers located immediately upstream or downstream of the condenser.

1.2 Review of Heat Pump Drying

Conventionally, materials are dried either in the field (sun drying) or using high temperature dryers (electric, gas fired, etc.). Successful outdoor drying depends upon good weather and indeterminate weather can render a product worthless. High temperature drying can damage the nutrient content and impart an unpleasant smell to the dried product. Dehumidification is the process of removing water vapor from air. It can be done by cooling the vapor-air mixture below its dew point temperature. A mechanical refrigeration cycle (heat pump) is often used for dehumidification. Hodgett [2] studied the heat pump assisted batch drying of agricultural material and expressed the performance of a heat pump dryer in terms of specific moisture extraction rate (SMER), which is the mass of water removed per kilowatt hour (kWh) of energy spent. According to Hodgett [2], the SMER for heat pump drying is in the range of 1.0-4.0 kg/kWh, whereas the SMER for a conventional dryer is in the range of 0.2-0.6 kg/kWh. He also demonstrated that a heat pump dryer with an average
SMER of 3 kg/kWh would use less energy than a steam-heated dryer with a thermal efficiency of 75% or a direct-fired dryer of 58% efficiency.

Direct batch dryers are the type more commonly fitted with heat pumps because of ease in design. It is simple to re-circulate the process air in a batch dryer and this enhances the energy efficiency of the heat pump dryer compared to a conventional dryer. The hygrothermal conditions in the drying chamber are varied throughout the operation when drying batchwise. Tai et al. [3], [4] and [5] studied five different batch drying/dehumidification systems, including three with heat pumps and compared them on the basis of specific power consumption and primary energy consumption. They observed that the efficiency of each system improved as the relative humidity of the air leaving the dryer increased. The systems were most efficient when run at a minimum relative humidity of 30%. A closed cycle heat pump dryer proved to be the most advantageous of the five systems.

Oliver [6] did case study on process drying of wool, clay and timber with a dehumidifying heat pump. He suggested that the heat pump based dryer was more efficient when the drying temperatures required are no more than 50°C with an average relative humidity during drying at the dryer exit of well above 30%. This statement agrees with the results obtained by Tai et al. [3] and Baines [7]. According to Baines [7], the maximum temperature limit varies depending on the refrigerant and equipment used. Baines [7] concluded from his studies that using heat pumps in dryers reduces the energy loss incurred by the heating and evaporation processes that are energy inefficient in the simple heat and vent dryer. He obtained SMER in the range of 1 kg/kWh - 6 kg/kWh compared to that of 1 kg/kWh - 4 kg/kWh obtained by
Hodgett [2]. According to Strommen and Kramer [8], heat pump dryers produce a higher quality product compared to traditional oil fired dryers. Filho and Strommen [1] used a pilot scale heat pump dehumidifier for drying biomaterial, which demonstrated a reduced energy consumption of up to 60-80%. Prasertsan and Saensaby [9] dried sawed wood and bananas using a heat pump dryer. The maximum SMER was 0.572 kg/kWh to dry sawn wood to less than 10% moisture content (mc).

Heat pump dryers have been studied for both batch and continuous dryers. Although heat pump-assisted batch dryers have proven to be economical compared to conventional heated air dryers, their acceptance by the agricultural industry has been slow. The reason is difficulty in controlling the process to achieve a high efficiency. In batch drying the moisture content of material decreases with time and hence the relative humidity (RH) of the air inside the drying chamber. After a certain time period, the RH usually goes below 25 to 30%. The amount of power consumed remains the same even though the moisture condensed at the evaporator decreases. Tai et al. [3] showed that the power consumption per kilogram of moisture removed (the reverse of SMER) in a heat pump assisted dryer was a minimum when the RH of the drying chamber was maintained above 30%.

Sosle et al. [10] batch dried apple slices from 70% mc to 13% mc at 30°C using a heat pump. They found that the final quality of the product in terms of consumer preference was excellent. The chewability and the mouthfeel of the product were much appreciated in case of the low temperature processed product. Microscopic examination of freehand sections showed that the cell wall structure remains in an
extremely good shape and the cells regain their shape and size to a remarkably good extent upon re-hydration.

Achariyaviriya et al. [11] batch dried papaya glace at 50°C using a heat pump. The specific airflow rate of 40.2 kg/h-kg dry product was used. The fraction of evaporator bypass air was 0.63 and the initial and final moisture content were 40.4% and 23.2% (db), respectively. MER and SMER obtained from the experiment were 0.171 kg-water/h and 0.091 kg-water/kWh, respectively.

Continuous dryers, where the material is fed and discharged at a uniform rate are also configured with heat pumps. Unlike batch dryers, the continuous dryers are much easier to control since the energy flows and temperatures are constant, as the material is uniform and flows through at a constant rate. Very few studies have been done on heat pump assisted continuous dryers compared to heat pump assisted batch dryers. The most significant work was by Clements et al. [12]. They used wetted foam rubber as the test material in a continuous heat pump dryer. The initial moisture content of the foam was 64%. They showed that when the RH of the air at the dryer exit was increased from 32% to 80%, the SMER of the dryer doubled from 1.25 to 2.5 kg/kWh. To maintain a minimum relative humidity of 30% inside the dryer, the foam feed rate was kept at 11.5 kg/h. The relative humidity of the air entering the evaporator increased to 80% when the product feed speed was increased to 28.5 kg/h. In spite of the increased SMER, they did not dry the material to below 18% moisture content, which is quite high for safe storage of most agricultural crops. Another drawback of their study was that they performed drying tests on foam rubber instead of on hygroscopic agricultural materials. Because there have been few studies, more
detailed study is required to understand and explore the operating characteristics of heat pump assisted continuous dryers using crop material.

To understand the performance characteristics of a heat pump, it is important to find the overall heat transfer coefficients and refrigerant temperatures at the condenser and evaporator coils as well as the refrigerant mass flow rate. There is no straightforward and simple procedure available to calculate these. An accurate and a simplified heat pump model developed for simulating the performance of a heat pump assisted continuous dryer would be an asset in system design for growers of specialty crops. The advantages of heat pump assisted dryers, compared with conventional hot air drying, have been studied by Baines [7], Geeraert [13], Hodgett [1], and Tai et al. [3]. These studies were based on simple models, which did not take account of the detailed heat and mass transfer phenomena taking place in the evaporator and the condenser [12]. Clements et al. [12] developed a detailed heat pump assisted continuous drying model for simulating performance. The model developed by Clements et al. [12] was too complicated and lacked the versatility required to use in heat pump dryer design. There is a need to develop a simplified heat pump dryer model, which can give results for drying of hygroscopic agricultural material accurate enough for heat pump dryer design.
1.3 Research Objectives

The objectives of this research project are as follows:

i) Develop a model, which can be used for heat pump based dryer design using average (as opposed to local) evaporator and condenser coil conditions;

ii) Establish the accuracy of the computer model by comparison of experimental and predicted results using uncertainty analysis over a wide range of system operating conditions;

iii) Conduct a simulation to establish operating parameters for a prototype heat pump drying system.

1.4 Overview of the Thesis

The procedure to calculate detailed heat and mass transfer taking place at the dryer and heat pump system is presented in Chapter 2. This procedure was used to develop the simulation program for a heat pump assisted cross flow (thin layer) continuous bed dryer. In the dryer and heat pump models, all the governing equations are discussed in addition to the methodology and techniques that were used by these models. A description of the experimental heat pump drying system the instrumentation, calibration of the instrumentation and the experimental procedure are found in Chapter 3. A cabinet dryer was used to dry material (alfalfa) in batches and for tests emulating continuous bed drying. In Chapter 4, the results obtained from the experiments along with the comparison of the measured and predicted data in-order to verify the accuracy of the heat pump dryer model, are predicted. In Chapter 5, the
developed model is used to predict the performance of a prototype heat pump assisted continuous bed drying system currently under construction. A simulation was also conducted to establish operating parameters for the prototype. The actual prototype system will be available for experimentation in the summer of 2001. Finally, the general conclusions that result from this research project and recommendations for future work are presented in Chapter 6.
Chapter 2

DRYER AND HEAT PUMP MODELS

The modeling strategy adopted in this research involves the development of separate simulation models for the heat pump and dryer, and combining them using the laws of conservation of mass and energy.

2.1 Dryer Model – Thin Layer

During drying, agricultural material undergoes a process of simultaneous heat and mass transfer. Heat is required to evaporate the moisture, which is removed from the drying product surface by the external drying medium, usually, air. Most materials, when dried under constant external conditions, exhibit a constant rate of moisture loss, followed by a falling rate drying period. The constant rate period is not a characteristic of normal agricultural drying. It is reasonably well established that most of the drying of agricultural products takes place in the falling rate period. Several models have been proposed for describing the transfer of moisture in capillary porous products during the falling rate period. The most accepted equations were developed by Luikov [14]. A simplification of the solution to the equations developed by Luikov [14] was done by Crank [15]. This simplified solution is frequently used to describe the thin layer drying of agricultural material [16]. The basic equation is given by
\[ MR = \frac{M - M_e}{M_o - M_e} = \exp(-kt) \]  

where,

- \( t \) = time, min,
- \( k \) = drying rate constant, min\(^{-1}\),
- \( MR \) = moisture ratio (fraction),
- \( M \) = moisture content at time \( t \),
- \( M_e \) = equilibrium moisture content, and
- \( M_o \) = initial moisture content.

The condition of the material and air changes with time during continuous drying. Continuous dryers are categorized by the relative direction of flow of the agricultural material and the air movement through drier. The three basic designs are cross flow, concurrent flow and counter flow dryers. In the present research, the cross flow drier is considered. This type of dryer is shown schematically in Figure 2.1.

\[ \text{Figure 2.1: Schematic diagram of a continuous bed cross flow dryer} \]
Figure 2.2 shows an elemental volume (dxdy) of unit length taken from a continuous bed cross flow dryer of length, L, and at an arbitrary location within the moving bed of the material.

![Diagram of elemental volume in a continuous bed cross flow dryer]

**Figure 2.2: Elemental volume of the continuous bed cross flow dryer**

Wet material enters the dryer at one end and leaves at the other end. Hot and dry air is assumed to be flowing across the bed in the x direction (cross flow). Material is moving in the y direction and its properties are assumed to be constant along the z direction, i.e. along the width of the bed. There are four variables; the temperature of air, $T_a$, the temperature of agricultural material, $T_g$, the humidity ratio, $W$, and the moisture content of material, $M$. $G_p$ and $G_a$ are the mass flow rates of the material and air, respectively. To find these variables, four independent equations are required. What follows is a modified version of a modeling approach developed by Bala [16].
It is assumed that the dryer takes a total time, \( t_t \), to dry the material. This time can be divided into an equal number of small intervals, \( N \), in such a way that the change in material property is assumed to be negligible in that interval.

An iterative procedure is adopted to find the four unknowns; \( T_a, T_g, W \) and \( M \) at the different time intervals.

### 2.1.1 Drying Rate Equation

The drying rate can be determined directly from the thin layer drying equation. This equation is used to find the moisture content of material at any time \( t \) \((t \leq t_f)\) while drying. The rate of change in moisture content is equal to an appropriate thin layer drying equation, which can be expressed as

\[
\frac{\partial M}{\partial t} = -k(M - M_e).
\]  

(2)

Moisture content and time are integrated from their initial values of \( M_o \) and \( t_o \) to the final values of \( M \) and \( t \). \( M_o \) is the initial moisture content of the material at time \( t_o \), and \( M \) is the moisture content of the material at any time, \( t \), on the drying bed. After integration, the moisture content becomes

\[
M = M_e + (M_o - M_e) \cdot a \cdot e^{-kt}.
\]  

(3)

where,

\[
a = e^{kt_o}.
\]

The drying constant, \( k \), and the equilibrium moisture content, \( M_e \), vary for different agricultural materials. Usually they are determined experimentally. The
following equations for the drying constant and equilibrium moisture content are for alfalfa, taken from Sokhansanj et al. [17],

\[ M_e = \left[ \frac{1}{cT} \ln \left( \frac{1}{1 - RH} \right) \right]^{\frac{1}{m}} \]  

(4)

where,

\[ T = \text{temperature, K}, \]
\[ RH = \text{relative humidity of air (fraction), and} \]
\[ c \text{ and } m \text{ are dimensionless material constants which for alfalfa are } 8.51 \times 10^{-5} \]
\[ \text{and } 1.013, \text{ respectively.} \]

The drying constant is not actually a constant but depends on the air temperature,

\[ k = 0.2865 \exp(0.179 T_a) \]  

(5)

where,

\[ T_a = \text{air temperature, } ^\circ\text{F, and} \]
\[ k \text{ is expressed in } h^{-1}. \]

From equation (3), the moisture content of the material to be dried can be found at any time, t.

### 2.1.2 Mass Balance Equation

When a material undergoes drying, there exists a mass balance between the material and the process air. This implies that the change in moisture content of the air while passing through the material will be equal to the change in material moisture content given by
\[
\frac{\partial W}{\partial x} = -\frac{G_p}{G_a} \frac{\partial M}{\partial y} \tag{6}
\]

The humidity ratio, \( W \), and the thickness of the bed in the \( x \) direction are integrated from their initial values of \( W_0 \) and \( x_0 \) to final values of \( W \) and \( x \). In equation (6), the moisture content, \( M \), at any point, \( y \), along the drying bed can be determined using equation (3). After integration the humidity ratio becomes

\[
W = W_o + bx \tag{7}
\]

where,

\[
b = \frac{G_p}{G_a} \frac{\partial M}{\partial y}, \quad \text{and}
\]

\( x_o = 0 \) at all points on the drying bed, assuming no change in material properties in the \( x \) direction.

Equation (7) can be used to calculate the change in humidity ratio as the air passes through the bed.

### 2.1.3 Energy Balance Equation

Energy balance exists in a dryer, when there are no unknown energy losses or air leaks to the surroundings. According to the energy balance equation, a change in enthalpy of air is equal to the heat transferred convectively to the material and that supplied to air in the evaporated moisture. The energy balance equation was rearranged and solved in terms of the air temperature gradient across the drying bed thickness, \( x \).
The temperature of the air, $T_a$, and thickness of the bed, $x$, are integrated from initial values of $T_{a0}$ and $x_0$ to final values of $T_a$ and $x$. The change in the humidity ratio of the air, while passing across the bed in the $y$ direction, is obtained from equation (7). The final form of the equation will come out to be

$$
\frac{\partial T_a}{\partial x} = - \left( \frac{h_{cv} + G_a C_{pw} \left( \frac{\partial W}{\partial x} \right)}{G_a \left( C_{pa} + C_{pw} W \right)} \right) (T_a - T_g).
$$

(8)

The temperature of the air at the exit for the bed is calculated using equation (9).

$$
T_a = T_g + e^{\frac{d}{x_0}} (T_{a0} - T_g).
$$

(9)

where,

$$
d = h_{cv} + G_a C_{pw} \left( \frac{\partial W}{\partial x} \right),
$$

$$
e = G_a \left( C_{pa} + C_{pw} W \right),
$$

$x_0 = 0$ at all points on the drying bed,

$C_{pw} =$ specific heat of water vapor, kJ/kgK,

$C_{pa} =$ specific heat of air, kJ/kgK, and

$h_{cv} =$ heat transfer coefficient, kJ/m$^2$minK.

The temperature of the air at the exit for the bed is calculated using equation (9).

2.1.4 Heat Transfer Rate Equation

When the material undergoes drying, both sensible and latent heat transfer takes place between the material and the process air. The heat transfer rate between the process air and the material is equal to the sum of change in sensible heat of the
material and the net enthalpy change of air due to the moisture evaporation which can be expressed as a temperature gradient in the y direction as

\[
\frac{\partial T_g}{\partial y} = \left( \frac{h_{cv}(T_a - T_g) - \left(G_a(L_g + (C_{pw} - C_{pl})T_g)\left(\frac{\partial W}{\partial x}\right)\right)}{-G_p(C_{pg} + MC_{pl})} \right)
\]

The temperature of the material, \( T_g \), and the distance along the bed, \( y \), are integrated from their initial values of \( T_{go} \) and \( y_0 \). \( T_{go} \) is the temperature of the material at distance, \( y_0 \), and \( T_g \) is the temperature of the material after traveling a distance, \( y \), along the bed. The final value of material temperature, \( T_g \), and its position, \( y \), in the present iteration are the initial values \( T_{go} \) and \( y_0 \) in the next step of the iteration process. Equation (10) is solved to obtain

\[
T_g = \frac{1}{g} \left( f + gT_{go} \right) e^{\left( \frac{y - y_0}{h} \right)} - f
\]

where,

\[
f = h_{cv}T_a - G_aL_g \left( \frac{\partial W}{\partial x} \right),
\]

\[
g = \left( -h_{cv} + \left(C_{pw} - C_{pl}\right)G_a \left( \frac{\partial W}{\partial x} \right) \right),
\]

\[
h = -G_p\left(C_{pg} + MC_{pl}\right),
\]

\(C_{pl} = \) specific heat of liquid, kJ/kgK,

\(C_{pg} = \) specific heat of material, kJ/kgK, and

\(L_g = \) latent heat of vaporization of moisture from material, kJ/kg.

Equation (11) can be used to find the variation in the material temperature while it is passing through the dryer.
2.2 Psychrometric Conditions of Process Air in a Dryer

To design and match a heat pump to the dryer, it is necessary to know the psychrometric properties of the process air circulating inside the heat pump dryer. Figure 2.3 is a schematic diagram of the experimental drying system used in this study. The experimental system is described in more detail later in Chapter 4. However, it basically represents a closed chamber with recirculated air. This air passes through a small heat pump system and from there up through racks containing the material to be dried.

![Schematic diagram of heat pump dryer system](image)

Figure 2.3: Schematic diagram of heat pump dryer system

Figure 2.4 shows the drying process on a psychrometric chart. In this case, it was assumed that the drying process (1a-2a) was isenthalpic/adiabatic, i.e. no heat and mass transfer between the material and the surroundings [10]. This is a good assumption where the moisture is mostly on the surface of the material. The
temperature of process air at the evaporator coil exit (point 4a) was restricted to 5°C to avoid frosting at the coil.

![Ideal psychrometric chart for the dryer](image)

**Figure 2.4:** Ideal psychrometric chart for the dryer

In Figures 2.3 and 2.4, point 1a is the condition of air at the dryer inlet, point 2a is at the evaporator inlet and point 4a is at the condenser inlet. To find the psychrometric properties of the process air, the following equations were used:

Saturated Vapor Pressure, $P_{ws}$: (Taken from ASHRAE Handbook, Fundamentals [18])

$$\ln(P_{ws}) = \frac{C_1}{T} + C_2 + C_3T + C_4T^2 + C_5T^3 + C_6T^4 + C_7\ln T$$  \(12\)

where,

- $C_1 = -5.674359E+03$
- $C_2 = 6.3925247E+00$
- $C_3 = -9.6778430E+03$
- $C_5 = 2.0747825E-09$
- $C_6 = -9.4840240E-13$
- $C_7 = 4.1635019E+00$
\[ C_4 = 6.2215701 \times 10^{-7} \]

\[ T = \text{Absolute temperature, K;} \]

Humidity Ratio, W:

\[ W = 0.622 \left( \frac{P_w}{P - P_w} \right) \]  \hspace{1cm} (13)

where,

\[ P_w = \text{vapor pressure, Pa, and} \]

\[ P = \text{atmospheric pressure, Pa.} \]

Enthalpy, H:

\[ H = 1.006(T) + W(2501 + 1.805(T)) \]  \hspace{1cm} (14)

where,

\[ T = \text{temperature, } ^\circ \text{C} \]

### 2.3 Thermodynamics of a Heat Pump

A heat pump is a device which, operating in a cycle, maintains a body, B (Figure 2.5), at a temperature higher than the temperature of the surroundings. By virtue of the temperature difference, there will be heat leakage, \( q_1 \), from the body to the surroundings. The body will be maintained at the constant temperature, \( t_1 \), if heat is discharged into the body at the same rate at which heat leaks out of the body. The heat is extracted from the low temperature reservoir, which is nothing but the atmosphere (or the process air condition inside the chamber at the dryer exit) and discharged into the high temperature body, B, with the expenditure of work, \( W_e \), in a cyclic device called a heat pump. In a heat pump, attention is confined to the high temperature body.
B. Here \( q_1 \) and \( W_e \) are of primary interest, and the performance of a heat pump is defined in terms of the coefficient of performance (COP) [19], where

\[
\text{Figure 2.5: Thermodynamic heat pump cycle}
\]

\[
COP_{HP} = \frac{q_1}{q_1 - q_2} = \frac{q_1}{W_e} \tag{15}
\]

It is shown in the literature [19, 20] that

\[
COP_{HP} = COP_{ref} + 1 \tag{16}
\]

where,

\[
\begin{align*}
\text{COP}_{HP} &= \text{Coefficient of performance of heat pump (dimensionless)}, \\
\text{COP}_{ref} &= \text{Coefficient of performance of refrigerator (dimensionless)}, \\
q_1 &= \text{energy supplied to maintain body } B \text{ at higher temperature } t_1, \text{ W,} \\
q_2 &= \text{energy extracted from the surroundings at temperature } t_2, \text{ W, and} \\
W_e &= \text{work input to the heat pump (compressor), W.}
\end{align*}
\]

At steady state, the electrical energy \( W_e \), supplied to an electric heater is dissipated as heat to the space, but when supplied to a heat pump dissipates \( q_1 (>W_e) \) giving a thermal advantage. For a heat to flow from a cooler to a hotter body, \( W_e \)
cannot be zero, and hence, the $\text{COP}_{HP}$ cannot be infinite. For air-conditioning applications, $\text{COP}_{HP}$ usually ranges between 3 and 5 [20].

2.4 Heat Pump Model

The basic purpose of a heat pump in a dryer is to supply hot air to the drying chamber at the desired temperature and to dehumidify the exhaust air coming from the drying chamber. To achieve the desired drying conditions, it is important that the heat pump should be matched properly with the dryer. There is no standard procedure to match a heat pump and a dryer. However, there are individual approaches for modeling processes of heat transfer or fluid flows involving direct application of partial differential equations for mass, momentum and energy with appropriate boundary conditions. According to Theerakulpisut [21], in the usual approach to modeling heat pumps or other complex thermal systems, attempts were made to characterize the numerous processes by means of simpler algebraic equations or possibly ordinary differential equations.

Generally researchers analyze the performance of a heat pump, and are not concerned with the design. They usually have the information (both the physical and thermal properties) for the heat pump and have to determine the conditions of the air at the exit of the condenser or evaporator coils. In design, one has to find the energy transfer, mass flow rate and temperature of refrigerant at the condenser and evaporator coils, given the air conditions required at the exit of the condenser and evaporator coils.
The main purpose of developing the present heat pump model is to provide information for predicting performance, which can be of assistance in the design of heat pump dryer systems for specialty crops. In this research, it is assumed that the heat pump will be subjected to the constant operating conditions associated with continuous drying. The spatial average, and not the local conditions of the heat pump were used to find the overall heat transfer coefficients at the condenser and evaporator coils.

It was decided to modify the actual refrigeration cycle through some assumptions before modeling the heat pump system. Figure 2.6 represents an actual refrigeration cycle in which different pressure and energy losses of refrigerant in the condenser and evaporator coils are depicted.

![Figure 2.6: Actual heat pump cycle](image)

Figure 2.6 was simplified to get an ideal refrigeration cycle (Figure 2.7).
To get the ideal refrigeration cycle, the following assumptions were considered in the actual cycle (Figure 2.6):

a) No suction line heating (b-5r');

b) No suction valve pressure drop (5r'-c);

c) No cylinder heating of vapor (c-d);

d) Iso-entropic compression (d-e);

e) No discharge valve pressure drop (e-f);

f) No discharge line heat loss (f-6r');

g) No pressure drop in condenser and evaporator coils (g-7r' and 8r'-b);

h) No liquid line heat loss or sub-cooling (7r'-a).

The energy transfer between the refrigerant and process air at the evaporator and condenser coil is given by the following equations:

\[ q_{\text{evp}} = U_{oe} A_{oe} \left( H - H_{sR} \right); \]  \hspace{1cm} (17)

\[ q_{\text{cond}} = U_{oc} A_{oc} \Delta T_{\text{LMTDcond}}. \]  \hspace{1cm} (18)
The energy transfer at the evaporator depends on the overall heat transfer coefficient, $U_{oe}$, the surface area of the evaporator coil, $A_{oe}$, and the enthalpy difference of moist air calculated at the average air temperature, $H$, and average refrigerant temperature, $H_{sR}$. It is important to note here that although $U_{oe}$ is conventionally called the overall heat transfer coefficient [12, 21, 36], its units are different, i.e. kg/m²/s instead of W/m²/K.

Similarly, the energy transfer at the condenser depends on the overall heat transfer coefficient, $U_{oc}$, the surface area of the condenser coil, $A_{oc}$, and the logarithmic mean temperature difference between the air and refrigerant, $\Delta T_{LMTDcond}$.

### 2.4.1 Overall Heat Transfer Coefficient at the Condenser Coil

The condenser is easier to model than the evaporator because of the absence of liquid at the outer surface of the condenser coil. The principal task of the condenser model in the heat pump dryer is to predict the energy transfer, the mass flow rate and the temperature of the refrigerant at the condenser coil. For finding the above parameters, the overall heat transfer coefficient for the condenser, $U_{oc}$, must be known. It can be computed by using equation (19), which assumes negligible tube wall and fouling resistance. Equation (19) is taken from Jolly et al. [22]

$$U_{oc} = \frac{1}{\frac{A_c}{A_{hi}h_i} + \frac{1 - \phi_{eff}}{h_o + \frac{A_{ie} + \phi_{eff}}{A_f} + \frac{1}{h_o}}}$$

(19)
where,

\[ h_i = \text{inside (refrigerant) heat transfer coefficient, } W/m^2K, \]
\[ h_o = \text{outside (air) heat transfer coefficient, } W/m^2K, \]
\[ A_c = \text{total condenser outside area, } m^2, \]
\[ A_{ti} = \text{inside condenser tube area, } m^2, \]
\[ A_{to} = \text{outside condenser tube area, } m^2, \]
\[ A_f = \text{total condenser fin area, } m^2, \text{ and} \]
\[ \phi_{eff} = \text{fin efficiency.} \]

For coil specifications and fin efficiency, refer to Appendix A.

### 2.4.1.1 Condenser Coil Convective Heat Transfer Coefficient, \( h_i \) – Refrigerant Side

There are different correlations available to find the average forced convective heat transfer coefficient. Each correlation has been improved subsequently by different researchers over time (Chen [23], Forster and Zuber [24], Shah [25], Gungor and Winterton [26], Eckels and Unruh [27] and Mehendale Jacobi [28]).

The Gungor-Winterton correlation [26] was used in this research. This correlation developed by Gungor-Winterton [26] is based on the correlation of Chen [23]. The proposed correlation covers both the “saturated nucleate boiling region and the two-phase forced convection region”. It was assumed that both mechanisms occur to some degree over the entire range of the correlation and that the contributions made by the two mechanisms are additive. A large data base (3693 points) was used covering both the nucleate and convection dominated boiling regimes for water,
refrigerants (R11, R12, R22, R113, and R114) and ethylene glycol [29]. The basic equation for convective heat transfer coefficient is

\[ h_f = E h_f + S h_{\text{NeB}} \]  

(20)

where,

- \( h_f \) = forced convective heat transfer coefficient, W/m\(^2\)K,
- \( h_{\text{NeB}} \) = nucleate pool boiling coefficient, W/m\(^2\)K,
- \( E \) = enhancement factor, and
- \( S \) = suppression factor.

a) Heat transfer coefficient, \( h_f \)

The heat transfer coefficient, \( h_f \), is calculated from the Dittus-Boelter correlation (Chaddock and Noerager [30] and Theerakulpisut [21]), using the local liquid fraction of flow, i.e. \( m_r (1 - X) \), where \( m_r \) is mass flow rate of refrigerant and \( X \) is mass fraction or quality of refrigerant.

\[ \text{Nu} = 0.023 (\text{Re})^{0.8} (\text{Pr})^{n}, \quad \text{Re} > 5000, \text{Pr} > 0.6 \text{ and } L/D_i > 60 \]  

(21)

where,

- \( n = 0.3 \) for cooling (condenser) and 0.4 for heating (evaporator),

\[ \text{Nu} = \frac{h_f D_i}{k} \]  

(Nusselt Number),  

(22)

\[ \text{Re} = \frac{\rho v D_i}{\mu} \]  

(Reynolds Number),  

(23)

\[ \text{Pr} = \frac{\mu C_p}{k} \]  

(Prandtl Number),  

(24)
\[ D = \text{inside coil diameter, m}, \]
\[ \rho = \text{density of refrigerant, kg/m}^3, \]
\[ \mu = \text{viscosity, Pa.s}, \]
\[ C_p = \text{specific heat, J/kg-K}, \]
\[ L = \text{total length of coil, m}, \text{and} \]
\[ v = \text{velocity of refrigerant, m/s}. \]

b) Convection Enhancement Factor, \( E \)

Chen [23] first developed the correlation for the saturated nucleate boiling region and the two-phase forced convection region. He assumed that both mechanisms occur to some degree over the entire range of the correlation and that the contributions made by the two mechanisms are additive. Gungor-Winterton [26] modified Chen’s correlation and suggested that the heat transfer coefficient, \( h_i \), is not be simply the addition of the values for nucleate and two-phase forced convection region rather it will be given by equation (20) and there will be an enhancement factor, \( E \), multiplied to heat transfer coefficient calculated using Dittus-Boelter correlation, \( h_f \). The correlation for finding \( E \) is

\[
E = 1 + 24000(Bo)^{1.16} + 1.37 \left( \frac{1}{X_f} \right)^{0.86}
\]  \( (25) \)

where,

\[
Bo = \text{boiling Number} = \frac{\Phi}{G_{i_{fg}}},
\]  \( (26) \)
\( X_{\mu} = \text{lockhart-Martinelli parameter} = \left( \frac{\mu_l}{\mu_v} \right)^{0.1} \left( \frac{\rho_v}{\rho_l} \right)^{0.5} \left( 1 - \frac{X}{X} \right)^{0.9}, \) \hspace{1cm} (27)

\( \phi = \text{surface heat flux, W/m}^2, \)

\( G = \text{mass velocity, kg/m}^2 \cdot \text{s}, \)

\( i_{fg} = \text{latent heat of vaporization, J/kg,} \)

\( X = \text{liquid mass fraction,} \)

\( \mu = \text{viscosity, Pa } \cdot \text{s and} \)

\( \rho = \text{density, kg/m}^3. \)

Subscripts \( l \) and \( v \) are for liquid and vapor phase of refrigerant, respectively.

c) **Suppression Factor, S**

Suppression factor, \( S \), was first introduced by Chen [23]. He defined it as the ratio of the mean superheat \( (\Delta T_e) \) of refrigerant to the wall superheat \( (\Delta T_{sat}) \). He also quoted that the suppression factor might be expected to approach unity at low flows and zero at high flows. Chen [23] suggested that \( S \) can be represented as a function of the local two-phase Reynolds number, \( \text{Re} \), and enhancement factor, \( E \), as follows:

\[ S = \left( 1 + 0.00000115(E)^2(\text{Re})^{1.17} \right)^{-1}. \] \hspace{1cm} (28)

d) **Nucleate Pool Boiling Coefficient, \( h_{ncB} \)**

Looking at any boiling curve for a fluid, different boiling regimes can be found. Initially the heat transfer takes place due to natural convection and after a certain temperature it goes into the nucleate boiling regime where heat transfer takes place because of bubble formation and their coalescence. The heat transfer coefficient in the nucleate boiling regime is given by
\[ h_{nc,\beta} = 55 \left( \frac{P_{rp}}{P_{crit}} \right)^{0.12} \left( -0.4343 \ln \left( \frac{P_{rp}}{P_{crit}} \right) \right)^{0.55} M^{-0.5} \phi^{0.67} \]  

Where,

- \( P_{rp} \) = reduced pressure = \( P/P_{crit} \),
- \( P_{crit} \) = critical pressure, Pa,
- \( P \) = pressure at temperature \( T \), Pa,
- \( \phi \) = heat flux, W/m², and
- \( M \) = molecular weight.

### 2.4.1.2 Condenser Coil Heat Transfer Coefficient, \( h_o \) – Air Side

The condenser coil inside heat transfer coefficient, \( h_o \), has been found to generally be much larger than the airside heat transfer coefficient [31] and [12]. This implies that the thermal convection on the airside is dominant, and therefore \( h_o \) is the most important factor in equation (19). Correlations for airside convective heat transfer coefficients for tubes with flat fins have been published by McQuiston and Tree [31], Rich [32], McQuiston [33] and Gentry et al. [34]. However, in most applications, some type of fin pattern is embossed into the fin surface to increase the airside heat transfer coefficient by promoting shedding and restarting of the laminar boundary layer. Beecher and Fagan [35] first investigated the effect of fin pattern on airside heat transfer coefficient. They correlated the measured data in terms of the dimensionless heat transfer coefficient (Nusselt number) based on the arithmetic mean temperature difference (AMTD), \( N_u_o \), and the Graetz number, \( G_z \), a dimensionless measure of the level of flow development.
According to Beecher and Fagan [35], the effect of number of tube rows, \( N_r \), for flat fins, on heat transfer is given by

\[
\frac{Nu_{a_{nr}}}{Nu_{a_{3r}}} = \left( \frac{N_r}{3} \right)^{0.22 \left( \frac{Gz}{30} \right)^{0.6278}} \quad (2 \leq N_r \leq 6)
\]

(30)

where,

\( Nu_{a_{nr}} = \) Nusselt number for \( N_r \) rows at AMTD,

\( Nu_{a_{3r}} = \) Nusselt number for 3 rows at AMTD,

\( N_r = \) Number of rows,

\( Gz = \) Graetz number =

\[
\frac{4 \rho_a W_f^2 V_{\text{fin}} \text{ Pr}(1 - \beta)}{\mu_a N_r P_l \left( \sec \theta (1 - \beta) + \frac{2W_f \beta}{D_i} \right)}
\]

(31)

\( \rho_a = \) air density, kg/m\(^3\),

\( W_f = \) fin spacing, m,

\( V_{\text{fin}} = \) air velocity between fins for a tubeless heat exchanger, m/s,

\( \text{Pr} = \) Prandtl number for air,

\( \beta = \) ratio of tubeless cross-sectional area to fin surface area including tube area,

\( \mu_a = \) air viscosity, Ns/m\(^2\),

\( k = \) thermal conductivity of air, W/m-K,

\( P_l = \) longitudinal distance between rows, m,

\( \sec \theta = 1 \) for plain fins. (The ratio of the actual fin surface area to the projected fin surface area), and

\( D_i = \) inside tube diameter, m.
$\text{Nu}_{a_3 r}$, was found by curve-fitting the data given by Beecher and Fagan [35].

The effect of number of tube rows $\text{Nr}$ for fins on heat transfer is given by

\[
\text{Nu}_{a_3 r} = a_1 + a_2 G\zeta + a_3 G\zeta^2 + a_4 G\zeta^3 + a_5 G\zeta^4 + a_6 G\zeta^5
\]

(32)

where,

\[
\begin{align*}
  a_1 &= -6.051 & a_4 &= 0.005368 \\
  a_2 &= 2.15 & a_5 &= -0.00008312 \\
  a_3 &= -0.1541 & a_6 &= 4.687E-07 
\end{align*}
\]

The dimensionless Graetz number can be obtained from equation (31). From equations (30), (31) and (32), the Nusselt number based on arithmetic mean temperature difference (AMTD), $\text{Nu}_{\text{amr}}$, for $N$ rows of tubes can be found. The Nusselt number at the logarithmic mean temperature difference (LMTD), $\text{Nu}_l$, is generally preferred, at low air velocities where the heat exchanger effectiveness is high and the temperature difference between the fin surface and the exit air is very small. However, small errors in temperature measurement can lead to large errors in the calculated LMTD. Therefore, Beecher and Fagan [35] correlated the LMTD data in terms of AMTD, which, due to the uniform fin surface temperature, was simply the difference between the fin surface temperature and the mean air temperature. $\text{Nu}_l$ is related to $\text{Nu}_{\text{amr}}$ by the equation

\[
\text{Nu}_l = 0.25 G\zeta \ln \left( \frac{1 + \frac{2 \text{Nu}_{\text{amr}}}{G\zeta}}{1 - \frac{2 \text{Nu}_{\text{amr}}}{G\zeta}} \right)
\]

(33)
\[ Nu_t = \frac{h_o D_h}{k} \quad (34) \]

where,

The hydraulic diameter, \( D_h \), is calculated by

\[ D_h = \frac{2W_f (1 - \beta)}{\left( \sec \theta (1 - \beta) + 2W_f \frac{\beta}{D_i} \right)} \quad (35) \]

where,

\[ \beta = \frac{\pi D_i}{4(P_i P_f)} \],

\[ \sec \theta = \left( X_f^2 + P_d^2 \right)^{0.5} \]

\[ X_f = \frac{P_i}{2N_p} \]

For unpatterned fins, \( P_d \) is zero and \( \sec \theta \) is unity.

### 2.4.2 Overall Heat Transfer Coefficient at Evaporator Coil, with Dehumidification

An important function of the evaporator is to remove moisture from the system by dehumidification of the exhaust air from the dryer. Sensible and latent heat are also recovered which would have otherwise been lost to the atmosphere. With dehumidification, the airside surface can be wetted with liquid or frost. Condensation of moisture at the evaporator coil makes the evaporator difficult to model. Water-vapor transfer does not depend on temperature difference alone; instead, it depends
more on the difference between moist air enthalpy, $H$, and the enthalpy of moist air calculated at the refrigerant temperature, $H_{sR}$. The task of the evaporator model is to predict the total energy transfer from the refrigerant to the process air and the temperature and mass flow rate of refrigerant. It is assumed that dehumidification occurs only on the two-phase section of the evaporator coil. The overall heat transfer coefficient at evaporator is required,

$$q_{evp} = U_{oe} A_e (H - H_{sR}).$$  \hspace{1cm} (36)$$

where,

$U_{oe} =$ overall heat transfer coefficient for the wet surface based on enthalpy difference, kg/m$^2$/s,

$H_a =$ enthalpy of air, kJ/kg, and

$H_{sar} =$ enthalpy of saturated air evaluated at refrigerant temperature, kJ/kg.

$U_{oe}$ can be computed by using equation (37), which assumes negligible tube wall and fouling resistance. This equation is taken from Jolly et al. [22], which is the modified form of equation given by Threlkeld [36] and Turaga et al. [37]

$$U_{o,e} = \frac{1}{b_R A_{oe} + b_{sm} (1 - \Phi_{eff}) + b_{sm} \frac{A_{pi} \Phi_{eff}}{h_{ow} h_{ow}}}$$  \hspace{1cm} (37)$$

where,

$A_{oe} =$ total evaporator area, m$^2$,

$A_{pi} =$ inside evaporator pipe area, m$^2$,

$A_{po} =$ outside pipe evaporator area, m$^2$,

$A_F =$ total evaporator fin area, m$^2$, and
\[ h_{ow} = \text{outside wet convective heat transfer coefficient, W/m}^2\text{-K.} \]

In order to calculate \( U_{oe} \) from equation (37), values of the mean water film surface temperature, \( t_{wm} \), and the pipe temperature, \( t_i \), are assumed. These assumptions allow initial approximations to be made for \( b_{wm} \) and \( b_R \) (see details at the end of this page), respectively. After calculation of \( U_{oe} \), the assumptions can be checked. Jolly et al. [22] developed a procedure for this, which is represented here.

The local rate of heat transfer is given by

\[ q = h_i A_{pi} (t_i - t_R), \tag{38} \]

where, \( t_i \) and \( t_R \) are tube and refrigerant temperatures.

By definition \( U_{oe} \) [18], gives

\[ q = U_{oe} A_{oe} (H - H_{SR}). \tag{39} \]

Combining equations (38) and (39) gives

\[ t_p = t_R + \frac{U_{oe} A_{oe} (H - H_{SR})}{h_i A_{pi}} \tag{40} \]

As given by Threlkeld [29], \( t_{wm} \), can be determined by

\[ H_{swm} = H - \frac{C_{pa} h_{ow} \Phi_{ef}}{b_{wm} h_{co}} \left( 1 - \frac{b_R U_{oe} A_{op}}{h_i A_{pi}} \right) (H - H_{SR}) \tag{41} \]

Equation (41) allows the determination of \( t_{wm} \) through calculation of the enthalpy of saturated air, \( H_{swm} \), at the same temperature.

Some of the terms in equation (37) are worthy of further discussion. The term \( b_R \) includes enthalpy and temperature terms in it. It is derived by the concept that moisture transfer does not depend on temperature difference alone; instead, it depends on the enthalpy difference,
where,

\[ b_R = \frac{H_{st} - H_{sR}}{t_t - t_R}, \]  

(42)

\[ H_{st} = \text{the assumed enthalpy of saturated moist air evaluated at tube temperature, kJ/kg,} \]

\[ H_{sR} = \text{the assumed enthalpy of saturated moist air evaluated at refrigerant temperature, kJ/kg,} \]

\[ t_t = \text{tube temperature, } ^{\circ}\text{C, and} \]

\[ t_R = \text{refrigerant temperature at saturation, } ^{\circ}\text{C.} \]

The term, \( b_{wm} \), is evaluated at the mean surface temperature of the water film on the fin. It is the slope in equation (43), which represents saturated enthalpy of air, \( H_{as} \), as a function of temperature over a small range (5°C),

\[ H_{as} = a + b_{wm} t_{as} \]

\[ \frac{dH_{as}}{dt_{as}} = b_{wm} \]  

(43)

where,

\[ a = \text{constant, kJ/kg,} \]

\[ b_{wm} = \text{constant, kJ/kg-}^{\circ}\text{C, and} \]

\[ t_{as} = \text{temperature of saturated air, } ^{\circ}\text{C.} \]

For a larger temperature range, the saturation enthalpy of moist air is better represented by a polynomial of higher order. For the purpose of this study, the equation derived in Theerakulpisut [21] based on ASHRAE 1985 [38] data over the temperature range of 0°C to 50°C is used:

\[ H_{as} = 9.3839 + 1.71137 t_{as} + 0.02222 t_{as}^2 + 0.00063 t_{as}^3. \]  

(44)
Differentiation of equation (44) yields

\[
\frac{dH_{as}}{dt_{as}} = 1.71137 + 0.0444t_{as} + 0.00189t_{as}^2.
\]

Comparing equation (45) with the slope of equation (43) it follows that

\[
b_{wm} = 1.71137 + 0.0444t_{as} + 0.00189t_{as}^2.
\]

The value of \(b_{wm}\) given by equation (46) is the actual value of the slope of equation (44) corresponding to a particular saturation temperature, \(t_{as}\), while, the value of \(b_{wm}\) obtained by equation (43) is an average value over a small temperature range of 5°C. Theerakulpisut [21] averaged the values of \(b_{wm}\) over 5°C-ranges, starting from 0°C until the whole range was completed at 50°C. The average values of \(b_{wm}\) were assumed to correspond to the values of \(t_{as}\) at the mid-points of every step of 5°C. The average values of \(b_{wm}\) and the midpoints \(t_{as}\) were curve-fitted to give

\[
b_{wm} = 0.0026(t_{as} + 10)^{0.93463} + 1.48609.
\]

The error in calculating \(b_{wm}\) from equation (47) was less than 0.5% of actual value of \(b_{wm}\) in the range of 0°C to 50°C.

The next term in equation (37) to consider is the heat transfer coefficient, \(h_{cow}\), for the outer surface or airside of the evaporator coil,

\[
h_{cow} = \frac{1}{Cpa} \left( \frac{1}{b_{wm}} + \frac{y_w}{h_{cow}} \right),
\]

where,

\(Cpa = \) specific heat of air, kJ/kgK,

\(b_{wm} = \) coefficient evaluated at water film temperature, kJ/kg-°C,

\(h_{cow} = \) convective heat transfer coefficient for outside surface, kW/m²K,
\( y_w \) = thickness of water film, m, and \\
\( k_w \) = thermal conductivity of water, W/mK.

In this study, a water film thickness of 0.1016mm (0.004 inches), as suggested by Myers [39] and Theerakulpisut [21] was assumed. Myers [39] derived a simplified equation showing \( h_{cow} \) as a function of saturated-air face velocity, \( V_{sf} \), both for dry-surface and for wet-surface operation. Myers equation is given by

\[
\frac{h_{cow}}{h_{cod}} = 0.626V_{sf}^{0.101}.
\]  (49)

For the calculation of convective heat transfer coefficient (dry), \( h_{cod} \), at the outside surface of the evaporator coil, the procedure given in Section 2.4.1.2 is used. The inside heat transfer coefficient, \( h_i \), can be calculated following the procedure outlined in Section 2.4.1.1. Calculation of fin efficiency, \( \phi_{eff} \), is given in Appendix A.

In the ideal cycle (Figure 2.7) 5r-5r' and 6r'-6r are superheating and de-superheating sections for the refrigerant fluid, respectively. The evaporator coil is divided into two sections, superheat and two-phase. In the superheat section, a superheat temperature increase of 5°C has been assumed, to avoid liquid refrigerant entering the compressor. The refrigerant is then compressed iso-entropically before entering the condenser. Using the effectiveness NTU (Number of Transfer Units) method on the condenser and the evaporator coils, it was found that the fractions of superheating and de-superheating areas were very small (0.001% of total coil area). Hence, the amount of heat transfer through these sections was neglected. The final refrigerant cycle left was 5r-6r-7r-8r (Figure 2.8) with only two-phase sections of refrigerant at the condenser and evaporator coils.
Figure 2.8 shows the modeled pressure – enthalpy diagram for refrigerant circulating inside the heat pump.

![Figure 2.8: Modeled pressure-enthalpy diagram for refrigeration cycle](image)

The effectiveness-NTU method that was used to find the de-superheating and superheating fractions for condenser and evaporator coils, as follows:

\[
\varepsilon = 1 - \exp \left( \frac{(NTU)^{0.22}}{C} \left( \exp \left( -C(NTU)^{0.73} \right) - 1 \right) \right); \\
\varepsilon_{dsh} = \frac{C_{rdsh} (t_{rci} - t_{rs})}{C_{min} (t_{rci} - t_{aci})} \\
\]

where,

\( \varepsilon \) = effectiveness,

\( C_{rdsh} \) = capacity rate for refrigerant, \( m_r C_{pr} \), kJ/sK

\( C_{min} \) = smaller capacity rate (\( m_r C_{pr} \) or \( m_a C_{pa} \) whichever is smaller), kJ/sK,

\( C \) = capacity rate ratio, \( C_{min}/C_{max} \),

\( t_{rci} \) = temperature of refrigerant at condenser inlet (or at compressor outlet), °C,

\( t_{rs} \) = temperature of refrigerant at vapor saturation curve, °C, and
\( t_{aci} \) = temperature of air before entering the condenser coil, °C.

Equations (50) and (51), can be used to find NTU, which, in turn, is used to calculate the de-superheating fraction of the condenser, \( f_{dsh} \), from equation (52),

\[
NTU = \frac{U_{dsh} A_{dsh}}{C_{min}} = \frac{U_{dsh} f_{dsh} A_c}{C_{min}},
\]  

(52)

where,

\( U_{dsh} \) = overall heat transfer coefficient for desuperheating section of condenser, W/m\(^2\)-K

\( A_{dsh} \) = area of desuperheating section of condenser coil, m\(^2\), and

\( A_c \) = total area of condenser coil, m\(^2\).

A similar procedure was adapted to find the superheating fraction of area of the evaporator coil.

Figure 2.9 shows the psychrometric chart, depicting the change in process air conditions inside the heat pump dryer system.
The energy transfer between the refrigerant and the air at the evaporator, $q_{\text{evp}}$, and condenser, $q_{\text{cond}}$, can be summarized in the following equations

$$q_{\text{evp}} = \dot{m}_r (H_{5r} - H_{8r}) = \dot{m}_a (H_{2a} - H_{4a}) = U_{oe} A_{oe} (H - H_{sh});$$  \hspace{5cm} (53)

$$q_{\text{cond}} = \dot{m}_r (H_{6r} - H_{7r}) = \dot{m}_a C_{pa} (t_{1a} - t_{4a}) = U_{oc} A_{oc} \Delta T_{LMTDcond}.$$  \hspace{5cm} (54)

In the above equations, the unknowns are $\dot{m}_r, \dot{m}_a, U_{oe}$ and $U_{oc}$

where,

$\dot{m}_r$ = mass flow rate of refrigerant, kg/s,

$\dot{m}_a$ = mass flow rate of air, kg/s,

$U_{oc}$ = overall heat transfer coefficient at the condenser coil, W/m$^2$K,

$U_{oe}$ = overall heat transfer coefficient at the evaporator coil, kg/m$^2$/s,

$\Delta T_{LMTDcond}$ = log mean temperature difference taken at the condenser coil, °C,
\[ A_{oe} \& A_{oc} = \text{total evaporator and condenser coil areas, m}^2, \]
\[ H_{sr} = \text{enthalpy of saturated air at refrigerant temperature, kJ/kg, and} \]
\[ H = \text{enthalpy of saturated air, kJ/kg.} \]

\( H \) with subscripts 1a, 2a and 4a represents the enthalpies for air at various locations inside the dryer (Figure 2.9) and \( H \) with subscripts 5r, 6r, 7r and 8r represents enthalpies for refrigerant at various locations inside heat pump loop (Figure 2.8).

The moisture removal rate, \( m_w \), can be calculated by applying mass balance theory at the evaporator. If the mass flow rate of air, \( m_a \), is established then the moisture removal rate at the evaporator coil is given by

\[ m_w = m_a (W_1 - W_2), \quad (55) \]

where,

\( W_1 = \text{humidity ratio of air before entering the evaporator, kg/kg dry air, and} \)
\( W_2 = \text{humidity ratio of air after passing through the evaporator, kg/kg dry air.} \)

The initial values of temperature at condenser and evaporator are predicted to find refrigerant mass flow rate and the overall heat transfer coefficients. Mass flow rate of refrigerant can be calculated from either equation (53) or (54). Using equation (53), the mass flow rate becomes

\[ m_r = \frac{m_a (H_{2a} - H_{4a})}{(H_5 - H_8)}. \quad (56) \]
The next step in energy calculation is to find the overall heat transfer coefficient at the condenser and evaporator coils.

2.4.3 Compressor and Expansion Valve Model

The compressor is by far the most complex component in the heat pump. The major difficulties in mathematical modeling of compressors arise from the lack of understanding of basic knowledge of fluid flow, heat transfer and thermodynamics of the processes occurring within the compressor. According to Theerakulpisut [21] and as discussed by Qvale et al. [40], a very complex compressor model accounting for various processes within the compressor could be embedded in the heat pump simulation model. Unfortunately the size and complexity of such a compressor model would tend to overshadow the rest of the heat pump simulation. To simplify the calculations; constant volumetric efficiency and isentropic compression of the refrigerant are assumed. A more detailed model could be used, as described by Fischer and Rice [41], although this would require detailed manufacturer performance data, which can be difficult to obtain.

In the expansion valve model, a capillary tube is used. It controls the amount of superheat at the evaporator outlet. The expansion process is assumed to be isenthalpic. If the manufacturer performance data are available, the refrigerant mass flow rate through the capillary tube can be calculated. Otherwise, the capacity of the expansion valve is assumed large enough so that the refrigerant mass flow rate passing through the capillary tube is same as that through the compressor.
2.5 Combined Heat Pump and Dryer Model

The heat pump and the dryer models are combined to get a single heat pump dryer system. The exit conditions of air from the dryer are taken as inlet conditions to the heat pump and the exit conditions of air from the heat pump are taken as inlet conditions to the dryer. Thus the heat pump or dryer model can be changed without the need to change the program structure, and it is a simplified model compared to that of Clements et al. [12] and Theerakulpisut [21] models. Unlike the Baines and Carrington [42] model, which assumes that the air conditions leaving the product are fixed, this model calculates all the air and refrigerant conditions iteratively according to the mass and energy balances for a specified air mass flow rate.

The first step in the model is to find the condition of process air at important points in the process as indicated in Figures 2.3 and 2.4. The inputs to the model are the airflow rate inside the drying chamber, the temperature of process air at the exit of evaporator and condenser, the relative humidity of air at the exit of drier and the initial values of refrigerant temperatures at the evaporator and condenser coils. To find the first predicted values of temperatures the following method was adopted. The refrigerant at the evaporator must be able to cool the air from about 28°C to about 10°C. This implies that the refrigerant has to be at a lower temperature than 10°C. Therefore, a good guess for the first predicted value of temperature for the refrigerant at the evaporator would be about 0°C. Similarly, the refrigerant at the condenser has to heat the air from about 10°C to about 60°C. Hence, the refrigerant has to be at higher temperature than 60°C. So, the first predicted value or a good guess for the temperature of refrigerant at the condenser would be 75°C.
The dryer model is used to determine moisture removal rate, \( m_w \), at the evaporator from air, the temperature of process air, \( T_a \), the temperature of agricultural material, \( T_g \), the humidity ratio, \( W \), and the moisture content of material, \( M \). The mass flow rate of the air is used to find the mass flow rate of the refrigerant inside the heat pump cycle. The heat pump cycle uses refrigerant –134A (tetrafluoroethane) as the heat transfer medium. This refrigerant is a hydrofluorocarbon (HFC) and is relatively common in new environmental friendly refrigeration systems, compared to previously used chlorofluorocarbon (CFC) like R-12, R-22, etc. Before the calculations are conducted, the thermo-physical properties (enthalpy, pressure, viscosity, thermal conductivity, specific heat and latent heat of vaporization) of refrigerant – 134A given in ASHRAE handbook, Fundamentals [43] were curve fitted using a fifth-order Binomial expansion. These equations are given in Appendix B.

The heat transfer rate for the condenser is obtained from

\[
q_{\text{cond}} = U_{oc} A_{oc} \Delta T_{LMTD}
\]

and for the evaporator from

\[
q_{\text{evp}} = U_{oe} A_{oe} (H - H_{sr}),
\]

where,

\[
\Delta T_{LMTD} = \log \text{mean temperature difference} = \frac{(T_{PC} - T_1) - (T_{PC} - T_4)}{\ln \left( \frac{T_{PC} - T_1}{T_{PC} - T_4} \right)}, \degree C,
\]

\( T_1 \) & \( T_4 \) = temperature of process air at the condenser outlet and inlet (refer to Figures 2.3 and 2.4), \degree C, and

\( T_{PC} \) = first predicted value of refrigerant temperature at the condenser, \degree C.
The calculated values of $\Delta T_{LMTD}$ and $H_{SR}$ are used to find the new values of the refrigerant temperatures at the condenser and evaporator. From the condenser heat transfer equation, a new TPC is determined and from the evaporator equation, a new $H_{SR}$ is calculated, which will help in finding a new TPE (predicted value of the refrigerant temperature at the evaporator). The calculated new values of TPC and TPE are used in the next iteration. Iterations were performed until the difference between the new and the old refrigerant temperature were within $\pm 0.25^\circ C$. The condenser and evaporator coil specifications are given in Appendix A.
Chapter 3

EXPERIMENTAL SYSTEM AND TEST PROCEDURE

A small-scale experimental setup was established in order to verify the accuracy of the heat pump dryer model developed. Two household type dehumidifiers were used to heat and dehumidify the process air. These were located in a closed cabinet, which constituted the recirculating dryer. The main purpose of this Chapter is to describe the heat pump drying system, the instrumentation, the calibration of the instrumentation, and the experimental procedure.

3.1 Recirculating Cabinet Dryer

A cabinet dryer consisting of a chamber and two dehumidifiers is shown schematically in Figure 3.1 and in a photograph in Figure 3.2. The chamber was divided into left and right side compartments with air circulation through the compartments in a clockwise direction. The walls of the chamber were insulated with a layer of 0.051-m thick fiberglass insulation. The doors of the cabinet were well sealed to avoid air leaks using rubber strips. Two heat pump dehumidifiers, manufactured by Canadian Tire Corporation™ Model-43-5404-8 D1 (Refrigerant – 134A, Power rating = 424 W), were used for the study. These units have condenser and evaporator coils, which provided the heating and cooling/dehumidification of the process air in the dryer. These are shown in the photograph of Figure 3.3. The
condenser and evaporator coils in each of the units were located in close proximity. This arrangement is acceptable for room air conditioning but not ideal for drying, as the introduction of atmospheric air or the removal of process air at this point cannot be easily achieved. The systems are not likely as efficient as large-scale commercial systems considering that the evaporator coils have no fins. However, these units were relatively inexpensive and were capable of demonstrating the concept and validating the model. A fan (diameter 25 cm, model no. IS-4420WCA, 1125 rpm) is used in the dehumidifier to drive the air through the coils. These provide the airflow for the recirculating loop. The heat pump dryer system specifications are given in Table 3.1.

In the left compartment, the material was placed in the stacked trays in the process air path of the drying chamber. The base of each tray consisted of nylon mesh. As shown in Figure 3.1, sensors were located to measure the temperature and relative humidity of the air at the entrance and exit from the stacked trays, atmospheric air temperature and relative humidity (calculated using the wet bulb and dry bulb temperatures measured by the thermocouples), and the average velocity of process air across the trays. The moisture condensed at the evaporator was drained using a plastic tube into a container placed outside the chamber. A load cell continuously monitored the mass of container and contents. At the bottom of the left compartment a baffle was placed in order to help develop uniform flow rate of process air through the material trays.
Figure 3.1: Block diagram of the experimental heat pump dryer system

Figure 3.2: Overall view of experimental heat pump dryer system
Table 3.1: Heat pump dehumidifier specifications

<table>
<thead>
<tr>
<th>Specification</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Number of heat pump dehumidifiers</td>
<td>2</td>
</tr>
<tr>
<td>Refrigerant used</td>
<td>R134a</td>
</tr>
<tr>
<td>Power consumed by heat pump dehumidifiers</td>
<td>424 Watts</td>
</tr>
<tr>
<td>High side pressure (condenser)</td>
<td>295 psi</td>
</tr>
<tr>
<td>Low side pressure (evaporator)</td>
<td>140 psi</td>
</tr>
<tr>
<td>Dehumidifying capacity of each heat pump</td>
<td>9.5 L of water per day</td>
</tr>
</tbody>
</table>

The Condenser and evaporator coil specifications are given in Appendix A.

3.2 Instrumentation and Parameters Measured

This section provides more detailed information on the sensors described in the Section 3.1.

T-type thermocouples were used to measure the temperature at different places in the heat pump dryer system. The alloy combination of the thermocouples was copper
and constantan. They were the extension type with a maximum useful temperature range of -60 to 100°C. The error in temperature measurement (±1°C) was determined by placing the thermocouples in hot (100°C) and cold (0°C) water baths. Thirty thermocouples were installed. Four thermocouples were located at the entrance and four at the outlet of the trays in the left compartment of the drying chamber (Figure 3.1). To measure the refrigerant temperatures (Figure 3.3) four thermocouples were attached on each surface of the condenser and evaporator coils of the two dehumidifiers using a heat conducting glue. These thermocouples were isolated from the process air using a thick rubber insulation wrap. They were located at approximately four equidistant points on the coils to measure average temperature. Of these four thermocouples, two were located at the refrigerant entrance and exit of the condenser and evaporator coils. Two pairs of thermocouples, where each placed in the air gap between the condenser and evaporator coils of the two dehumidifiers. Two thermocouples were located outside the drying chamber. One of them measured the dry bulb temperature and the other measured the wet bulb temperature of ambient air. The wet bulb temperature was measured by wrapping a thermocouple with a wet wick (the wick was maintained wet using a small reservoir), over which the air was blown using a small fan. These thermocouples were used to provide a measurement of the ambient air temperature and relative humidity. The thermocouple used for measuring the wet bulb temperature was not very sensitive compared to the electronic relative humidity sensors (described in next paragraph). However, it was considered adequate for measuring ambient air relative humidity, which was almost constant over time.
Two Vaisala HMP233 relative humidity (RH) sensors were installed at the entrance and exit of the trays. These were electronic sensors, which give an electrical output signal based on the relative humidity of air. These sensors were calibrated against a chilled mirror (manufactured by General Eastern model no. 1211H) having maximum pressure and temperature of 300 Psi and 100°C. The calibration was done in the range of 5% to 95% RH. The error obtained from the calibration test results was ± 0.95% RH.

A hot wire anemometer (Model TA2, Air Flow Development Ltd., Georgetown, ON) was used to measure the velocity of process air between the trays. The calibrated accuracy of the anemometer was 0.01 m/s. The hot wire anemometer had an antenna to which the velocity measuring probe was fixed. Two small holes were made at the left side and at the back of the left compartment of the cabinet for inserting the antenna through them. Measurements for finding average air velocity were done at five different locations by traversing the hotwire anemometer across the trays through the holes. These measurements were done in between tray 7 and tray 8 (the bottom tray was counted as tray 1). It was assumed that the air velocity would be relatively uniform at that point. The fan speed for the heat pumps was not changed during the test. Therefore, the air velocities obtained in the recirculating loop depends on the flow resistance in the circuit. The power consumed by the two dehumidifiers was measured by a wattmeter (manufactured by Valhalla Scientific, digital power analyzer, model no. 2101/3-8113). The weight of the moisture condensed at the evaporator and crop material dried was measured using a load cell and an electronic weighing scale respectively. The opening and closing of the inlet and outlet ports at the top of the
drying chamber were controlled manually by using butterfly valves. The mass flow rate of air at the inlet port was measured using a Pitot tube (manufactured by Air Flow Development, ON). The Pitot tube was calibrated to measure airflow rates between 51 cfm and 255 cfm. The temperatures, relative humidities and the weight of material measured by a load cell were monitored continuously using an 8082A data logger (mfd. by Sciemetric Instruments, ON). The data logger was connected to a computer and its data was stored for later use.

Due to the practical difficulties in installing any instrument in the heat pump, the measurement of the refrigerant mass flow rate was not done. The refrigerant mass flow rate was calculated as part of the simulation.

3.3 Experimental Procedure

Two types of experiments were performed. One was the static bed or batch drying and the other one was an emulation of continuous bed drying. Alfalfa was the material used in all of the experiments. The reason for using alfalfa instead of specialty crops like ginseng, herbs, echinacea, feverfew, etc., was that crop drying properties like the equilibrium moisture content (Mₑ), drying rate constant (k), specific heat of material (Cₚₑ) and volumetric heat transfer coefficient (hᵥₑ) are available. Also the structure of alfalfa leaves and stems are similar to that of many herbs and specialty crops. Finally, alfalfa is abundant in the prairies, inexpensive and releases moisture quite well when exposed to relatively low temperatures (40-50°C), which helps in validating the heat pump model. Before drying, it was chopped to a size of 5–8 cm in order to increase surface area for drying. Using an electronic balance, 400 g of alfalfa
was weighed and spread uniformly over the trays. The thickness of the layer was 2-3 cm. The initial moisture content of alfalfa was measured to be 70% (wet basis) according to the procedure of ASAE standard 75-3517, 1997.

In batch drying, all nine trays were placed in the drying chamber at the beginning of the experiment and the doors were closed. The material in each tray was weighed at 30-min time intervals by rapidly removing the trays from the cabinet and weighing them before returning them back to the cabinet. Estimation of the material moisture content was done after weighing the trays. The batch drying was continued until the moisture content of the material in the bottom tray (tray 1) reached 10% (wet basis).

In continuous bed drying, the drying chamber was maintained at a steady state condition in terms of temperature and relative humidity before starting the experiment. It was found that the variation in temperature and relative humidity of process air at the end of batch drying was negligible. For this reason it was decided to perform continuous bed drying immediately following the batch drying. At the end of batch drying, when the bottom tray 1 was removed (once it reached 10% mc) from the drying chamber, all other trays (tray 2 to 9) were moved down on the racks allowing fresh tray of material (tray 10) to be kept on the top rack (rack 9). The doors were closed and the material in all the trays was dried for another 30 min. After that all the trays were removed and weighed again. By this time the bottom tray (tray 2) reached to 10% moisture content and was removed from the chamber. All other trays (tray 3 to 10) were moved down on the racks allowing next fresh tray of material (tray 11) to be kept on the top rack (rack 9). The same procedure was followed until the eighteenth tray was removed from the drying chamber.
As there was no heat sink/water cooled condenser, all the heat rejected by the air and the energy input at the compressor was recovered by condenser and was than supplied to the process air inside the drying chamber. The total energy available at the condenser was more than was required (evident from the basic thermodynamics of the heat pump as explained in section 2.3 of Chapter 2). There was no outlet for this extra energy available at the condenser, as the cabinet was well sealed. Due to this, the overall average air temperature inside the chamber continuously increased. To maintain a constant temperature inside the chamber, it was important to introduce some outside air into the chamber. This was done by using inlet and outlet ports at the top of the cabinet dryer, as shown in Figure 3.1. These ports were manually operated to control the temperature in the chamber through the amount of outside air introduced.
Chapter 4

EXPERIMENTAL RESULTS AND VERIFICATION OF SIMULATION MODEL

4.1 Experimental Results

The comparison of the measured and simulation data is important to verify the accuracy of the model. Experiments were performed using alfalfa for both batch drying and emulating continuous bed drying. Alfalfa was chosen rather than specialty crops, as it was readily available and the parameters for the drying model for alfalfa are relatively well established. The drying characteristics likely closely resemble those of herbs and likely some leafy medicinal plants, as well.

The overall objective of the experiments was to predict the performance of the dryer, i.e. the drying rate of the material and psychrometric conditions of the air, and also to determine the refrigerant mass flow rate and corresponding temperatures at the condenser and evaporator coils of the heat pump system, based on the psychrometric conditions of process air inside the drying chamber. The batch and continuous bed drying experiments were performed consecutively and parameters such as temperature, relative humidity, weight of material and moisture condensed at evaporator were monitored continuously.
Figure 4.1 depicts changes in temperature and relative humidity of process air as the material is dried. The temperatures of process air increased with time at the entrance and exit of the trays and ultimately reached constant values. Initially, the temperature was low because the material was wet and releasing latent heat. As time progressed, the material started to dry and the release of latent heat reduced, therefore increasing the process air temperature inside the cabinet. The maximum temperature attained by the air at the entrance to the trays was 45°C. The temperature was maintained at 45°C by manual opening and closing of inlet and outlet butterfly valves located at the top of the cabinet (See Figure 3.1). There were two reasons for not allowing the temperature to exceed 45°C. The first reason was that for specialty crops the optimum drying temperatures (at which no structural damage and nutrient losses occurs) lies between 30-45°C. The second reason was that, when temperatures of process air increased above 45°C, the psychrometric process moved towards the right on the chart increasing the energy required to condense moisture at the evaporator.

The relative humidity of process air at the entrance and exit of the trays decreased almost exponentially from an initial value of 79% down to 19% by the end of batch drying (Fig 4.1). The initial high moisture content of the material and low temperature of process air resulted in high relative humidity values. The initial evaporation of moisture from the material takes place at the surface. As the drying progresses, the moisture left in the material will be at its core. Therefore, the material will require more energy and time to diffuse the core moisture into the process air and hence this decreases the amount of moisture evaporated into the process air.
The emulation of continuous bed drying began at the end of batch drying (beyond 270 minutes). The temperature and relative humidity were constant (steady state) at that time because, on an average, the moisture within the material was also constant. As the continuous bed drying started, the relative humidity at the exit of trays, which was below 20%, increased and reached 30% and remained constant thereafter. Fresh material was introduced into the dryer every 30 min (Chapter 3) and dry material was removed maintaining a near steady rate of moisture evaporation into the process air. Controlling the mass flow rate of material into the drying chamber through continuous bed drying the relative humidity of process air can be adjusted to keep it at or above 30% before entering the evaporator coil, a necessary condition for removing moisture from the air at the evaporator.

Figure 4.1: Measured temperature and relative humidity of process air at the dryer entrance and exit
The variations in refrigerant temperatures at the condenser and evaporator coils of the two dehumidifiers over time are shown in Figure 4.2. The two units were purchased at the same time and were of the same model and capacity, yet still they showed a small difference in temperatures recorded (3°C) at the condenser and evaporator coils. The slight difference between the coil temperatures at the heat pump units could be due to a small difference in air mass flow rate. This would also explain why they have a very similar pattern in the variation of refrigerant temperatures. As the drying proceeded, the temperature of refrigerant at the condenser and evaporator coils increased and reached a constant value after about 150 min. The pattern in variation of refrigerant temperatures at the condenser coils (Figure 4.2) is observed to be similar to that of the variation in process air temperatures at the entrance of trays (Figure 4.1). The difference in the refrigerant temperatures at the condenser and the temperature of the process air at any given instant was about 15°C, which is quite high. This suggests that the condenser coils were not very effective in increasing air temperature.

The reason for the increase in temperature of refrigerant with time is the same as that given for the air temperatures in Figure 4.1.
Figure 4.2: Measured refrigerant temperature at condenser and evaporator coils over material drying time

Figure 4.3 depicts the changes in the moisture condensation rate for batch and continuous bed drying. The initial moisture condensation rate was high for batch drying and decreased as the drying progressed, reaching almost zero at about 270 min (end of batch drying). The high relative humidity of process air at exit from the dryer (Figure 4.1) resulted in a high initial moisture condensation rate. As the relative humidity of air decreased, the saturation temperature for process air increased, thus reducing the condensation rate which eventually reached almost zero at relative humidity values below 20%. In continuous bed drying, fresh material was introduced every 30 min, which increased the relative humidity of the air (to about 30%), increasing the condensation rate. The condensation rate remained constant, as the material introduction rate into the dryer was constant. The variation pattern of curves for predicted and measured data was similar during batch and continuous bed drying. The amount of moisture condensed at the evaporator was approximately 50% of the
total moisture evaporated from the material (alfalfa). The remaining 50% of the moisture was mainly lost to the ambient air through the inlet and outlet ports used to prevent overheating, with some additional loss when the chamber doors were opened to introduce and remove material during the continuous drying cycle. The heat pump dryer system used in this study was able to recover about 50% of the latent heat compared to a conventional dryer, which recovers none.

![Figure 4.3: Measured moisture condensation during batch and continuous bed drying with time](image)

Figure 4.3: Measured moisture condensation during batch and continuous bed drying with time

Figure 4.4 represents changes measured in the specific moisture extraction rate (SMER) and power consumed over drying time. The high SMER in the first 40 min was due to the high moisture condensation rate (0.477 kg/h) and the low power consumption of the compressors. After 40 min, a similar decreasing trend was observed for SMER and the relative humidity of the process air. Since the condensation rate was almost zero, the SMER was a minimum at the end of batch
drying. The reason for the low condensation rate at the end of batch drying was that the amount of moisture released by the material into the process air was negligible and also the temperature of the process air at the dryer entrance was high (45°C). This in turn led to an increase in energy required to condense moisture at the evaporator. As the continuous bed drying started, SMER increased to an average of about 0.004 kg/kWh and remained steady at this value throughout the drying period. In the continuous bed drying, the relative humidity of process air was about 30%. Previous research [3, 9, 12, 21] suggests that the heat pump system is more efficient when it is run at 30% or higher relative humidity of process air, enabling a high degree of condensation at the evaporator coil. As the material mass flow rate through the dryer increases, the relative humidity of the process air is increased thus increasing the value of SMER [12].

The initial power consumed by the dehumidifiers was low due to the low temperature of the process air at the exit of the condenser. This is because the power consumption by compressor of a heat pump depends on temperature of the refrigerant. If the temperature of the refrigerant required is low then the power consumed by compressor will be low. Therefore, power consumption increased over time as drying progressed because the latent heat released by the material decreased and the temperature of refrigerant increased.
4.2 Comparison of Experimental and Simulation Results

In order to determine the accuracy of the heat pump dryer model, the model was simulated using the psychrometric conditions of the batch and continuous bed dryer. The input properties required for the simulation include: the mass flow rate of air through the drying chamber, the temperature of the process air at the entrance and exit of the condenser coil, and the relative humidity of the process air at the dryer inlet. Utilizing a predetermined set of experimental conditions, the heat pump model was used to predict the refrigerant temperatures at the condenser and evaporator coils, the moisture removal rate at the evaporator, and the refrigerant mass flow rate.

Figure 4.5 depicts the variation in temperature measured at the condenser and evaporator coils. An error and/or difference between the predicted and measured temperatures of ±4°C at the condenser coil and ±5°C at the evaporator coil were
observed. All the curves generally showed good agreement in the temperature variation pattern.

![Figure 4.5: Comparison of measured and simulation temperatures of refrigerant at condenser and evaporator coils (during batch drying)](image)

It is of paramount importance to do uncertainty analysis of the experimental temperatures obtained, in order to verify whether the simulation results are within the permissible error limit or not. Figures 4.6 and 4.7 show the uncertainty analysis results in graphical form for the condenser and evaporator coils respectively. The uncertainty analysis results predict an error limit of ±5°C at the condenser and ±6°C at the evaporator coils. The differences observed in the temperatures obtained from simulation and experimental results (Figure 4.5) are within the uncertainty limits. This endorses the fact that the heat pump model accurately predicts the refrigerant temperature at the condenser and evaporator coils.
Figure 4.6: Uncertainty analysis for the experimental temperatures obtained at the condenser coil

Figure 4.7: Uncertainty analysis for the experimental temperatures obtained at the evaporator coil
Figure 4.8 represents the change in moisture extraction rate (MER) with material drying time. The moisture loss through the inlet and outlet ports and process air bypass around the evaporator coil was taken into consideration in calculating the predicted results (computer program is given in Section C.2.2, Appendix C). The MER increased initially with an increase in moisture condensation rate. The moisture condensation rate was high because of high relative humidity at the dryer exit. Also, as the drying continued, the relative humidity of process air decreased thus reducing the MER. A good agreement between general shape of the curves for the simulation and experimental results was observed. At any given time interval, the MER obtained experimentally was approximately 19% lower than that obtained for simulation results. This was expected. In weighing material, the drying chamber door was opened for couple of minutes before removing and replacing all the trays. It was not considered possible to determine the amount of process air that was leaked to the outside through the open door. Another problem was that the droplets of condensed moisture at evaporator took some time to form a stream and flow through the pipes before entering the collection container.
4.3 Measured Psychrometric Conditions of Process Air

Figure 4.9 shows the psychrometric process for the drying for one air cycle through the dryer and heat pump. This Figure is drawn for a particular set of drying conditions inside the chamber at the beginning of batch drying. In Figure 4.9, 1 - 2 represents the drying process. Points 1 and 2 are the conditions of process air at the dryer entrance and exit, respectively. The drying of agricultural material is often considered to be an adiabatic process [10]. The experimental results suggest that the drying process 1 - 2 was not adiabatic, even though the chamber was well sealed and insulated from the surroundings. For the drying process to be adiabatic, it is essential to have availability of free moisture in the drying chamber. However, crop materials don’t often have most of their moisture content on the surface. Therefore it takes extra energy to release the moisture from the core of crop material.
As represented by the process 2 – 3 – 4 – 4’, it was concluded that the evaporator coil was ineffective in cooling all of the process air near to the coil temperature. A reason for the ineffectiveness of coils was the absence of fins thus increasing the “effective” bypass of air not coming into contact with the coils. The fraction of air bypassed through the evaporator ‘b’ can be represented by [45]

\[ b = \frac{T_4 - T_4}{T_2 - T_4}. \]

This bypass factor provides an indirect measure of the efficiency of heat transfer at the coils. From the psychrometrics of Figure 4.9, b was calculated to be 0.71. The value of b for an efficient process at the evaporator should be close to 0.1 or 0.15.
Chapter 5

SIMULATED PERFORMANCE OF A PROTOTYPE CONTINUOUS BED DRYER

In this Chapter, the performance of a prototype heat pump drying system is evaluated using the model developed and validated in the previous Chapters.

5.1 Prototype Heat Pump Dryer System

A full-scale prototype heat pump drying system is under construction and will be available for field-testing in the summer of 2001. The system shown in Figure 5.1, uses re-circulating air, as was the case for the cabinet dryer. However, a continuous bed dryer is incorporated into the unit.

Figure 5.1: Overview of prototype cross-flow continuous bed heat pump dryer system
The system consists of three conveyors as shown schematically in Figure 5.2. Two conveyors (C1 and C2) are used for material drying and the third conveyor (C3) carries the dried material out of the drying chamber. The two conveyors carrying material (C1 and C2) move in opposite directions. The dryer inlet air flows from the bottom to top of the drying chamber in a cross-flow direction. The basic drying model developed in Chapter 2 can be used for the prototype heat pump dryer, except that there will be two conveyors instead of one. The dryer dimensions are shown in Figure 5.2. The airflow rate in the prototype heat pump dryer is assumed constant at 86 kg/min-m$^2$ (1000 cfm) by fixing the blower speed.

Figure 5.2: Schematic sectioned view of the counter cross flow continuous bed drying chamber

All dimensions are in meters
Figure 5.3 shows two elemental volumes (dx dy) taken from C1 and C2, each of length, L, and at an arbitrary location, but exactly in line with the cross-flow air stream.

Figure 5.3: Psychrometric and material conditions at the two elemental volumes on conveyors one and two

It was assumed that the dryer takes a total time, t, to dry the material. This time can be divided into an equal number of small intervals, N, in such a way that the change in material property can be assumed to be negligible in that interval. The material will take equal time to pass through C1 and C2. An iterative procedure is used to find Ta, Tg, W and M at different time intervals.

To find Ta, Tg, W and M, drying rate, mass balance, energy balance and heat transfer rate equations are required. These equations were derived in Section 2.1, Chapter 2. The only difference here will be in the elemental volume model used
(Figure 5.3) for determining the material and process air properties, along and across the dryer bed, respectively.

A computer program was written in FORTRAN to find:

1) The change in material temperature and moisture content along C1 and C2; and
2) the average temperature, humidity ratio and relative humidity of air at the exit from C1 (across the conveyor).

The inputs for the dryer simulation and program code are given in Sections D.3 and D.4 of Appendix D, respectively.

5.2 Heat Pump Model

The procedure for predicting the performance of the prototype heat pump system is the same as that developed in Section 2.4, Chapter 2. The energy transfer, mass flow rate and temperature of refrigerant at the condenser and evaporator coils were predicted based on the psychometric condition of air inside the drying chamber. However, the prototype heat pump system uses refrigerant 404A as the heat transfer medium. This refrigerant is a mixture of hydrofluorocarbons (R-125 (Pentafluoroethane) / R-143A (Trifluoroethane) / R-134A (Tetrafluoroethane) and their percentage by mass is 44/52/4) and is relatively common in commercial systems, along with R-134A. The thermophysical properties of R-404A from ASHRAE handbook, Fundamentals [43] were curve fitted using a sixth order binomial expansion. These equations are given in Section D.2 of Appendix D. Code was written to predict the prototype heat pump system performance. The program code is given in Section D.3 of Appendix D.
5.2.1 Heat Pump Specifications

The heat pump system is manufactured by Blanchard-ness, Montreal, QC. The compressor motor has the horse power of $\frac{1}{2}$ and run at 1750 rpm. The refrigerant used in the heat pump system is 404A. This system is designed to operate between high and low refrigerant pressures of 2.86 MPa and 1.48 MPa. The condenser and evaporator coils are made of copper, while the fins attached over these coils are of aluminum. The dimensions for both the coils are given in Section D.1 of Appendix D. Figure 5.4 is a schematic diagram of the prototype heat pump system.

Figure 5.4: Schematic block diagram of the prototype heat pump system
5.3 Combined Heat Pump and Dryer Model

The dryer and heat pump models developed in Sections 5.1 and 5.2 are combined to get a single heat pump dryer system model as was done for the cabinet dryer system. The exit conditions of air from the dryer are taken as inlet conditions to the heat pump and the exit conditions of air from the heat pump are taken as inlet conditions to the dryer. The procedure to evaluate the model is similar to that explained in Section 2.5, Chapter 2. Figure 5.5 is a schematic diagram of the prototype heat pump dryer system.

![Schematic diagram of the prototype heat pump dryer system](image)

**Figure 5.5: Schematic block diagram of the prototype heat pump dryer system**

5.4 Simulation Results

In this section, predicted results for the cross flow continuous bed dryer system are presented and discussed. Alfalfa is the material used for predicting the performance characteristics of the heat pump drying system. The reasons for using alfalfa instead of specialty crops are same as those given in Section 3.3, Chapter-3.
Figure 5.6 shows the variation in moisture content of the material at dryer exit as the material mass flow rate through the drying chamber is increased. The temperature and relative humidity of air at the dryer inlet are varied, keeping the air humidity ratio constant. Simulation results are obtained at three different temperatures of 50°C, 40°C and 30°C with respective relative humidity values of 8%, 12% and 22%. The dryer inlet air humidity ratio was kept constant at 5.5 g/kg of dry air. All the three curves showed the same general behavior. The lowest moisture content is obtained for an inlet air temperature of 50°C because, with the increase in inlet air temperature, vapor pressure inside the material increases and hence, the material moisture can be diffused more readily into the process air. However, there was not a significant difference in the amount of moisture removed over the three air temperatures studied.

In Figure 5.6, there is a sharp drop in moisture content of the material at dryer exit for the material mass flow rates below about 15 kg/h. Therefore, for the prototype drying system, the amount of moisture removed from the material will be less if the material mass flow rate is above 15 kg/h under the given psychrometric conditions. This likely occurs because the initial moisture removed from the material is at the surface and can be readily removed. It takes extra energy and time to remove the moisture present at the core of the material. Therefore, at mass flow rates of material below 15 kg/h, the material gets enough time and exposure to the process air to diffuse moisture from the core to the surface.

The safe limit of 10% mc at dryer exit was obtained when the material mass flow rate was only 3 - 4.5 kg/h. It may be worthwhile to investigate developing a hybrid
drying system. In a hybrid drying system, a conventional low temperature heat pump dryer could be combined with other means of high temperature drying techniques such as, microwave drying, electric heater drying, fluidized bed drying, etc. Hybrid drying can be of importance, because of the inherent transport properties of moisture within the material particles, it takes a much longer time to remove the last 20-30% of moisture entrapped within the material than it does to remove the initial 70-80% of moisture. Therefore, high temperature drying techniques coupled with low temperature drying can be of importance in removing the last 20-30% of moisture much faster from the material than the low temperature drying alone, provided this could be done without a detrimental loss of product quality.

![Figure 5.6: Predicted affect of material mass flow rate on moisture content of the material at dryer exit](image)

Figure 5.6: Predicted affect of material mass flow rate on moisture content of the material at dryer exit
Figure 5.7 depicts the change in average humidity ratio of air at the dryer exit (across the conveyors) as the material flow rate increased. The inlet air humidity ratio is held constant at 5.5 g/kg of dry air. It is observed that the exit air humidity ratio increased with increase in the material mass flow rate through the dryer. The reason is that more wet material comes in contact with the process air and hence, the air can gain more moisture.

It was also observed that the exit air humidity ratio was high for higher temperatures. As mentioned previously, as air temperature increases, vapor pressure inside the material increases, which in turn increases the humidity ratio of air by diffusing more moisture. The humidity ratio of air at the dryer exit increases with air temperature and material flow rate. However, the amount of change in humidity ratio of air between the inlet and outlet is very small. This is because of the inherent property of agricultural material, which diffuses moisture into the process air from its surface and core, very slowly (at low temperatures).

It can be implied from the above discussion that as the gain in humidity ratio of air from the dryer entrance to exit is small and is almost the same at 30°C, 40°C and 50°C, therefore the material will reach to the safe limit of 10% mc, almost at the same time, irrespective of the entrance air temperatures (Figure 5.6).
Figure 5.7: Predicted affect of material mass flow rate on humidity ratio of air at dryer exit

Figure 5.8 shows the variation in material temperature and moisture content along the length of both the conveyors with material drying time. These results are for dryer inlet air temperature, relative humidity and humidity ratio of 40°C, 12% and 5.5 g/kg of dry air, respectively. Similar curves can be obtained at temperatures of 30°C and 50°C. Since C1 is downstream from C2, it will be in contact with the air at a lower temperature. There is a sharp rise in the temperature of the material at the entering stage of each conveyor. This is due to the thermal capacitance effect as the material suddenly encounters a warmer air stream.

The curve for moisture content of material decreases from an initial high value of 70% (wb) to a low of 20% (wb) at the end of 300 min (or at the dryer exit). It is observed that the slope of moisture content curve for material on C1 is slightly
different than that of material on C2. The reason is that the conveyors are in contact with the air at different temperatures.

Figure 5.8: Change in temperature and moisture content of material along conveyors one and two of the dryer
Chapter 6

CONCLUSIONS AND FUTURE WORK

6.1 General Conclusions

The general objective of this investigation was to develop a simulation model for the heat pump based dryer design and establish the accuracy by comparison of the experimental and predicted results. A simulation was also conducted to determine the operating parameters for a prototype heat pump drying system under construction.

The results indicated that the required experimental low temperature drying conditions for specialty crops were achieved and also that all the objectives of the present investigation were accomplished. Heat pumps have their own advantages over other drying techniques. They provide energy savings (recovering latent and sensible heat at the evaporator) and environmental benefits through reduced air pollution. However, better quality of dried product is the major reason for using heat pump dryers. The current research led to the following general conclusions:

1) The drying of agricultural material in a heat pump dryer is not an adiabatic process as often assumed [10]. It was found that the drying process does not follow an isenthalpic line, even though the drying chamber was well sealed and insulated from the surroundings. The reason for non-isenthalpic drying process is the diffusion characteristics of agricultural material, which make the
process is the diffusion characteristics of agricultural material, which make the availability of free moisture difficult.

2) Low temperatures (30-45°C) for safe drying of specialty crops were achieved experimentally, which will help in maintaining better quality and conserve all the nutrients of the product.

3) The condenser and evaporator coils of the household dehumidifiers used in the present investigation were not effective in reducing the process air temperature close to the coil temperatures. Commercial quality heat pump systems are required.

4) The household dehumidifiers used in this study were about 50% more efficient in recovering latent heat from the dryer exhaust air compared to the conventional dryers, even with the ineffective condenser and evaporator coils.

5) In batch drying, heat pumps will be more efficient when they are run at or above 30% relative humidity of process air at the entrance of evaporator coil.

6) Continuous drying is potentially a better option than batch drying because high process air humidity ratios at the entrance of evaporator and, constant MER and SMER values can be maintained. During the diffusion controlled falling rate period (the period in which the drying process is controlled by the diffusion of water from inside of the product being dried), the moisture at the core of material takes a longer time to diffuse into the process air. Hence, the efficiency (SMER) of the batch dryers decreases drastically during this period of drying, resulting in a lower average SMER over the entire drying period.

7) The predicted maximum mass flow rate of material for the cross flow
continuous bed dryer is about 15 kg/h (Figure 5.6). Above this mass flow rate, a significant reduction in the amount of moisture removal occurs.

8) A change in inlet temperatures of the process air to the dryer has insignificant effect on drying of material. Therefore, the drying of material should be done at low temperatures (about 30°C), which will improve the product quality and will save power consumed by the compressor.

9) A safe limit of 10% mc at dryer exit was obtained, when the material mass flow rate was about 3 – 4.5 kg/h. This figure suggests that only a small amount of material will be dried by the prototype dryer per hour.

10) In a low temperature dryer, the material drying temperatures are near ambient temperatures and hence the energy losses to the surroundings will be less than high temperature dryers (temperatures above 80°C).

11) Closed circuit heat pump dryer systems would be more advantageous when operated in locations where atmospheric humidity level is high (Coastal region).

6.2 Future Work

The present research covered a small area in the field of heat pump assisted drying of specialty crops. There is still a lot of scope for researchers to investigate low temperature drying. The following are few areas that could be of interest for future studies:

1) Energy analysis, capital cost and economic feasibility of low temperature heat pump dryers compared to other drying techniques should be explored.
2) Accuracy of the predicted results of a prototype heat pump dryer system in the present research should be validated with the measured experimental data.

3) Investigation of a new approach, such as developing a hybrid drying system, is recommended. In a hybrid drying system, a conventional low temperature heat pump dryer could be combined with other means of high temperature drying techniques, such as microwave drying, electric heater drying, fluidized bed drying, etc. Hybrid drying can be of importance, because of the inherent transport properties of moisture within the material particles, it takes a much longer time to remove the last 20-30% of moisture entrapped within the material than it does to remove the initial 70-80% of moisture. Therefore, high temperature drying techniques coupled with low temperature drying can be of importance in removing the last 20-30% of moisture much faster from the material than the low temperature drying alone.
REFERENCES


38. ASHRAE. Psychrometrics. Fundamentals Handbook, ASHRAE, Chapter 6, 1985


A.1 Condenser and Evaporator Coil Specifications

It is important to know the coil specifications for the calculation of the overall heat transfer coefficients at the evaporator, $U_{oe}$ and the condenser, $U_{oc}$ as given in Chapter 2. The coil specifications given in this section are that of the household dehumidifier (manufactured by Canadian Tire Corporation Model-43-5404-8D1) used in the present experiments to verify the simulation model developed for the heat pump dryer system. The evaporator coil used in the household dehumidifier for the present experiments does not have any fins. While this is not likely a very efficient system, it was the only one available for experimental purposes.

**Condenser coil Specifications:**

- Number of longitudinal rows = 1
- Number of transverse rows = 10
- Transverse distance between rows = 0.0255 m
- Tube material = Brass
- Tube wall thickness = 0.00071 m
- Tube radius, outside = 0.005 m
- Fin material = Aluminum
- Fin radius, outside = 0.01275 m
Fin thickness = 0.00013 m
Fin spacing = 0.0019 m
Fin density = 12 / inch or 473 fins / m
Length of each tube = 0.27 m
Cross-sectional area of duct = 0.27m x 0.255m
Total length of coil = 2.7 m
Total air side condenser area (including fin area) = 0.632 m²
Total inside condenser tube area = 0.073 m²
Total outside condenser tube area = 0.08 m²
Total fin area = 0.552 m²

Evaporator coil Specifications:
Number of longitudinal rows = 2
Number of transverse rows = 16
Transverse distance between rows = 0.0169 m
Longitudinal distance between rows = 0.0169 m
Tube material = Aluminum
Tube wall thickness = 0.00071 m
Length of each tube = 0.24 m
Total length of coil = 7.68 m
Total air side evaporator area = 0.24 m²
Total inside evaporator tube area = 0.21 m²
Total outside evaporator tube area = 0.24 m²
A.2 Fin Efficiency

The fin efficiency is a function of the fin geometry and the airside heat transfer coefficient. Green and Roberts [44] evaluated the fin efficiency for circular fins. Most of the heat exchanger coils consist of a tube-plate-finned arrangement. Therefore, considering the plate fins to consists of a series of rectangular fins, the results of Green and Roberts [44] can be used by considering equivalent circular fins, where the equivalent fin diameter is

\[ D_{eq} = 2\left(\frac{X_{l} \cdot X_{T}}{\pi}\right)^{\frac{1}{2}} \] \hspace{1cm} (A1)

and the equivalent fin height is

\[ Z_{eq} = \frac{(D_{eq} - D_{o})}{2}, \text{ where} \]

\[ D_{o} = \text{tube outside diameter}. \]

Their data were formulated into equations for fin efficiency as a function of the parameters \( X \) and \( R \) where

\[ X = Z_{eq} \left(\frac{2h_{a}}{k_{f}Y_{f}}\right)^{\frac{1}{2}} \] \hspace{1cm} (A3)

and

\[ R = \frac{D_{eq}}{D_{o}}. \] \hspace{1cm} (A4)

The fin efficiency \( \phi \) is then defined using the following expressions:

- For \( R = 1.0 \),

\[ \phi = 1.03552448 - 0.31837607X + 0.02589744X^2 - 0.00101X^3; \] \hspace{1cm} (A5)
for $R = 1.5$,

$$\phi = 1.04188811 - 0.3713209X + 0.04282051X^2 - 0.000637X^3; \quad (A6)$$

for $R = 2.0$,

$$\phi = 1.03552448 - 0.45592852X + 0.08051282X^2 - 0.005004X^3; \quad (A7)$$

for $R = 3.0$,

$$\phi = 1.03881119 - 0.5195338X + 0.10275058X^2 - 0.0070863X^3; \quad (A8)$$

for $R = 4.0$,

$$\phi = 1.03230769 - 0.5540948X + 0.117016X^2 - 0.0087024X^3; \quad (A9)$$

where,

- $D_{eqv}$ = equivalent fin diameter, m,
- $X_T$ = transverse distance between coil rows, m,
- $X_L$ = longitudinal distance between coil rows, m,
- $Z_{eqv}$ = equivalent fin height, m,
- $D_o$ = coil outside diameter, m,
- $k_f$ = thermal conductivity of fin,
- $Y_f$ = fin thickness, m, and
- $h_a$ = heat transfer coefficient, W/m$^2$°K.
APPENDIX B

THERMOPHYSICAL PROPERTIES OF R-134A

Appendix B presents the curve fitted data for R-134A (tetrafluoroethane), to find the thermodynamic and transport properties. The curves are fitted to data for saturated liquid and vapor lines. These lines are as shown below in Figure B.1

ASHRAE [43] data are curve fitted in order to simplify and speed up the calculation process, as it is very cumbersome to solve the non-linear differential equations to get the thermo-physical properties of a refrigerant. The error obtained between the curve-fitted data and the value given in ASHRAE [43] is about ±1%. The following equations are for R-134A
B.1 Temperature (T, °C) - Enthalpy (H, J/kg)

B.1.1 Liquid saturation line

\[ H = a_1 + a_2(T) + a_3(T)^2 + a_4(T)^3 + a_5(T)^4 + a_6(T)^5 \]  \hspace{1cm} (B1)

\[ a_1 = 200002 \quad a_4 = 0.005944 \]
\[ a_2 = 1337 \quad a_5 = -0.00003226 \]
\[ a_3 = 1.53 \quad a_6 = 0.00001202 \]

B.1.2 Vapor saturation line

\[ H = a_1 + a_2(T) + a_3(T)^2 + a_4(T)^3 + a_5(T)^4 + a_6(T)^5 \]  \hspace{1cm} (B2)

\[ a_1 = 398685 \quad a_4 = -0.008933 \]
\[ a_2 = 583.4 \quad a_5 = 0.00004048 \]
\[ a_3 = -1.102 \quad a_6 = -0.00001944 \]

B.2 Temperature (T, °C) - Viscosity (\(\mu\), Pa.s)

B.2.1 Liquid saturation line

\[ \mu = a_1 + a_2(T) + a_3(T)^2 + a_4(T)^3 + a_5(T)^4 + a_6(T)^5 \]  \hspace{1cm} (B3)

\[ a_1 = 0.0002873 \quad a_4 = -2.724E-10 \]
\[ a_2 = -0.00003573 \quad a_5 = 1.842E-12 \]
\[ a_3 = 2.956E-8 \quad a_6 = -7.487E-15 \]

B.2.2 Vapor saturation line

\[ \mu = a_1 + a_2(T) + a_3(T)^2 + a_4(T)^3 + a_5(T)^4 + a_6(T)^5 \]  \hspace{1cm} (B4)

\[ a_1 = 0.00001094 \quad a_4 = 1.232E-12 \]
\[ a_2 = 4.637E-8 \quad a_5 = -5.159E-15 \]
\[ a_3 = 1.156E-11 \quad a_6 = 5.026E-16 \]
B.3 Temperature (T, °C) - Thermal Conductivity (K, W/m-K)

B.3.1 Liquid saturation line

\[ K = a_1 + a_2 T + a_3 T^2 + a_4 T^3 + a_5 T^4 + a_6 T^5 \]  \hspace{1cm} (B5)

\[ a_1 = 0.09339 \]
\[ a_2 = -0.0004596 \]
\[ a_3 = 4.952 \times 10^{-8} \]
\[ a_4 = -3.714 \times 10^{-9} \]
\[ a_5 = 7.619 \times 10^{-11} \]
\[ a_6 = -4.762 \times 10^{-13} \]

B.3.2 Vapor saturation line

\[ K = a_1 + a_2 T + a_3 T^2 + a_4 T^3 + a_5 T^4 + a_6 T^5 \]  \hspace{1cm} (B6)

\[ a_1 = 0.01179 \]
\[ a_2 = 0.00008611 \]
\[ a_3 = 8.913 \times 10^{-8} \]
\[ a_4 = 3.858 \times 10^{-9} \]
\[ a_5 = 8.69 \times 10^{-12} \]
\[ a_6 = -3.571 \times 10^{-14} \]

B.4 Temperature (T, °C) - Specific Heat (Cp, J/kg-K)

B.4.1 Liquid saturation line

\[ Cp = a_1 + a_2 T + a_3 T^2 + a_4 T^3 + a_5 T^4 + a_6 T^5 \]  \hspace{1cm} (B7)

\[ a_1 = 1336 \]
\[ a_2 = 3.078 \]
\[ a_3 = 0.00711 \]
\[ a_4 = -0.0002756 \]
\[ a_5 = 1.732 \times 10^{-5} \]
\[ a_6 = -3.862 \times 10^{-8} \]

B.4.2 Vapor saturation line

\[ Cp = a_1 + a_2 T + a_3 T^2 + a_4 T^3 + a_5 T^4 + a_6 T^5 \]  \hspace{1cm} (B8)

\[ a_1 = 878.4 \]
\[ a_2 = 4.373 \]
\[ a_3 = 0.08492 \]
\[ a_4 = 0.000155 \]
\[ a_5 = -0.00009301 \]
\[ a_6 = 11.98 \times 10^{-7} \]

B.5 Temperature (T, °C) - Latent Heat of Vaporization (\(i_g\), J/kg)

\[ i_g = a_1 + a_2 T + a_3 T^2 + a_4 T^3 + a_5 T^4 + a_6 T^5 \]  \hspace{1cm} (B9)
\[ ROH = a_1 + a_2 (T) + a_3 (T)^2 + a_4 (T)^3 + a_5 (T)^4 + a_6 (T)^5 \]  
\[ P = a_1 + a_2 (T) + a_3 (T)^2 + a_4 (T)^3 + a_5 (T)^4 + a_6 (T)^5 \]
APPENDIX C

DRYER SPECIFICATIONS AND COMPUTER PROGRAM CODE

C.1 Continuous Cross Flow Bed Dryer

Important parameters influencing the drying rate of the product include the inlet drying air temperature ($T_{1a}$), the relative humidity (RH%), the mass flow rate of air ($G_a$), and the mass flow rate of material ($G_p$). In order to study the effects of these parameters on the drying behavior of the material, the continuous cross flow bed dryer was simulated by subjecting the material bed to certain drying conditions. In every simulation one of the parameters was varied and the other three were kept constant. The unknowns of the dryer that we were trying to determine are the variation in the moisture content and temperature of the material along the dryer bed length, and the variation in the humidity ratio and temperature of the air across the dryer bed along the dryer bed length.

The constants that were used in the simulation of continuous cross flow bed dryer are:

a) Initial moisture content of material, $M_0 = 0.7$ (fraction);
b) Inlet temperature of material to be dried, $T_s = 15^\circ C$;
c) Time required by the material to pass through the dryer, $t = 300$ min;
d) Total length of dryer bed, $Y = 6$ m;
e) Thickness of layer of material, $X = 0.025$ m;
f) Specific heat of air, $C_{pa} = 1.008 \text{ kJ/kg-K};$

g) Specific heat of water, $C_{pw} = 1.89 \text{ kJ/kg-K};$

h) Specific heat of alfalfa, $C_{pg} = 4.52 \text{ kJ/kg-K};$

i) Specific heat of liquid, $C_{pl} = 4.186 \text{ kJ/kg-K};$

j) Latent heat of vaporization of free water, $L_w = 2501.64; \text{ and}$

k) Volumetric heat transfer coefficient, $h_{cv} = 175.058*(G_a)^{0.6906}.$

C.1.1 Program Code for Continuous Cross Flow Bed Dryer

The program code for continuous cross flow bed dryer is given in the attached diskette under the file name of "contbeddryer." More details about the model and theory behind the simulation can be found in Chapter 2.

C.2 Heat Pump Dryer Model

The main purpose of the heat pump dryer model is to predict the moisture removal rate at the evaporator coil ($m_w$), the mass flow rate of the refrigerant ($m_r$), and the temperature of the refrigerant at the evaporator and the condenser coils based on the psychrometric properties of air inside the dryer. Important parameters influencing the above four properties of a heat pump include the inlet drying air temperature ($T_{1a}$), the air temperature at the exit of the evaporator coil ($T_{4a}$), the relative humidity of air at the dryer exit (RH), and the mass flow rate of the air circulating inside the drying chamber ($m_a$). The air temperature at the exit of the evaporator coil was assumed constant at $5^\circ C$ to avoid frost formation. In order to study the effects of $T_{1a}$, $T_{4a}$, RH and $m_a$ on the heat pump properties, the heat pump
dryer was simulated by subjecting the heat pump to a certain psychrometric conditions. In every simulation one of the above parameters was varied and the other three were kept constant. The refrigerant used in the present simulation is R-134a (Tetrafluoroethane).

C.2.1 Program Code for Heat Pump Dryer – With no Bypass

A program was written for heat pump dryer with no bypass of air around the evaporator coil. Refer to the attached diskette for the program code. The program is given under the file name \textit{hpdnobypass}. This program will not work if the temperature of the refrigerant exceeds its critical limit. For more details refer to Chapter 2. The process is shown on the psychrometric chart below (Figure C.1)

\begin{figure}[h]
\centering
\includegraphics[width=0.5\textwidth]{psychrometric_chart.png}
\caption{Psychrometric chart for heat pump dryer with no bypass}
\end{figure}

C.2.2 Program Code for Heat Pump Dryer – With Bypass

A program was written for the heat pump dryer with bypass of the air around the evaporator coil. Refer to the attached diskette for the program code. The program is given under the file name \textit{hpdbypass}. This program will not work if the temperature of the refrigerant exceeds it critical limit. The different parameters that are used in the
present simulation model are already described in the previous program for the heat pump with no bypass. The psychrometric chart for the heat pump dryer with bypass is given below, in Figure C.2.

Figure C.2: Psychrometric chart for heat pump with bypass
APPENDIX D

PROTOTYPE HEAT PUMP DRYER SPECIFICATIONS AND PROGRAM CODES

D.1 Condenser and Evaporator Coil Specifications

The coil specifications are required for the calculation of overall heat transfer coefficients at the evaporator, \( U_{oe} \), and condenser, \( U_{oc} \), as given in Chapter 2. The coil specifications given in this section are that of the commercial dehumidifier manufactured by Blanchard-ness, model no. KOD-075-M4K and serial no. 2000020448.

Condenser coil Specifications:

- Number of longitudinal rows, \( N_r \) = 6
- Number of transverse rows, \( N_t \) = 12
- Longitudinal distance between rows, \( P_l \) = 0.032 m
- Transverse distance between rows, \( P_t \) = 0.038 m
- Tube material = Copper
- Tube wall thickness, \( X_p \) = 0.00071 m
- Tube radius, \( r_1 \) (outside) = 0.0063 m
- Fin material = Aluminum
- Fin radius, \( r_2 \) (outside) = 0.019 m
- Fin thickness, \( y \) = 0.00013 m
Fin spacing, $W_f$ = 0.0025 m
Fin density, $f_d$ = 10 / inch or 394 fins / m
Length of each tube, $L_t$ = 0.61 m
Crossectional area of the air duct = 2 ft x 2 ft or 0.3721 m$^2$
Total length of coil, $L$ = 43.92 m
Total air side condenser area (including fins area), $A_{oc}$ = 19.12 m$^2$
Total inside condenser tube area, $A_{tic}$ = 1.543 m$^2$
Total outside condenser tube area, $A_{toc}$ = 1.649 m$^2$
Total fin area, $A_{fc}$ = 17.47 m$^2$

**Evaporator coil Specifications:**

Number of longitudinal rows, $N_r$ = 8
Number of transverse rows, $N_t$ = 15
Transverse distance between rows, $P_t$ = 0.038 m
Longitudinal distance between rows, $P_l$ = 0.038 m
Tube material = Copper
Tube wall thickness = 0.00071 m
Tube radius, $r_1$ (outside) = 0.0063 m
Fin material = Aluminum
Fin radius, $r_2$ (outside) = 0.0214 m
Fin thickness, $y$ = 0.00013 m
Fin Spacing, $W_f$ = 0.0025 m
Fin density, $f_d$ = 10 / inch or 394 fins / m
Length of each tube, \( L_t \) = 0.61 m
Total length of coil, \( L \) = 73.2 m
Crossectional area of the air duct = 2 ft x 2 ft or 0.3721 m\(^2\)
Total air side evaporator area, \( A_{oe} \) = 40.65 m\(^2\)
Total inside evaporator tube area, \( A_{pie} \) = 2.57 m\(^2\)
Total outside evaporator tube area, \( A_{poe} \) = 2.75 m\(^2\)
Total evaporator fin area, \( A_{fe} \) = 37.89 m\(^2\)

**D.2 Thermophysical Properties of R-404A**

This section presents the curve fitted data for finding thermodynamic and transport properties of Refrigerant-404A. The curves are fitted to data for saturated liquid and vapor lines. These lines are as shown below in Figure D.1.

![Pressure–enthalpy diagram for a refrigerant](image)

*Figure D.1: Pressure–enthalpy diagram for a refrigerant*

ASHRAE [43] data are curve fitted in order to simplify and speed up the calculation process, as it is very cumbersome to solve the non-linear differential
equations to get the thermo-physical properties of a refrigerant. The error obtained between the curve fitted data and the values given in ASHRAE [43] is about ±1%.

The following equations are for R-404A

D.2.1 Temperature (T, °C) - Enthalpy (H, J/kg)

D.2.1.1 Liquid saturation line

\[ H = a_1 + a_2 (T) + a_3 (T)^2 + a_4 (T)^3 + a_5 (T)^4 + a_6 (T)^5 \]  \hspace{1cm} \text{(D1)}

\[ \begin{align*}
  a_1 &= 200867 \\
  a_2 &= 1461 \\
  a_3 &= -2.904 \\
  a_4 &= -0.104 \\
  a_5 &= 0.003367 \\
  a_6 &= 0.000006053
\end{align*} \]

D.2.1.2 Vapor saturation line

\[ H = a_1 + a_2 (T) + a_3 (T)^2 + a_4 (T)^3 + a_5 (T)^4 + a_6 (T)^5 \]  \hspace{1cm} \text{(D2)}

\[ \begin{align*}
  a_1 &= 364895 \\
  a_2 &= 507.9 \\
  a_3 &= -2.837 \\
  a_4 &= -0.104 \\
  a_5 &= 0.00143 \\
  a_6 &= 0.0000001563
\end{align*} \]

D.2.2 Temperature (T, °C) - Viscosity (Mu, Pa.s)

D.2.2.1 Liquid saturation line

\[ \text{Mu} = a_1 + a_2 (T) + a_3 (T)^2 + a_4 (T)^3 + a_5 (T)^4 + a_6 (T)^5 \]  \hspace{1cm} \text{(D3)}

\[ \begin{align*}
  a_1 &= 0.0002873 \\
  a_2 &= -0.000003573 \\
  a_3 &= 2.956E-8 \\
  a_4 &= -2.724E-10 \\
  a_5 &= 1.842E-12 \\
  a_6 &= -7.487E-15
\end{align*} \]

D.2.2.2 Vapor saturation line

\[ \text{Mu} = a_1 + a_2 (T) + a_3 (T)^2 + a_4 (T)^3 + a_5 (T)^4 + a_6 (T)^5 \]  \hspace{1cm} \text{(D4)}
D.2.3 Temperature (T, °C) - Thermal Conductivity (K, W/m-K)

D.2.3.1 Liquid saturation line

\[ K = a_1 + a_2 (T) + a_3 (T)^2 + a_4 (T)^3 + a_5 (T)^4 + a_6 (T)^5 \]  

D.2.3.2 Vapor saturation line

\[ K = a_1 + a_2 (T) + a_3 (T)^2 + a_4 (T)^3 + a_5 (T)^4 + a_6 (T)^5 \]

D.2.4 Temperature (T, °C) - Specific Heat (Cp, J/kg-K)

D.2.4.1 Liquid saturation line

\[ Cp = a_1 + a_2 (T) + a_3 (T)^2 + a_4 (T)^3 + a_5 (T)^4 + a_6 (T)^5 \]  

D.2.4.2 Vapor saturation line

\[ Cp = a_1 + a_2 (T) + a_3 (T)^2 + a_4 (T)^3 + a_5 (T)^4 + a_6 (T)^5 \]
D.2.5 Temperature \((T, ^\circ C)\) – Latent Heat of Vaporization \((i_{fg}, J/kg)\)

\[ i_{fg} = a_1 + a_2(T) + a_3(T)^2 + a_4(T)^3 + a_5(T)^4 + a_6(T)^5 \]  \hspace{1cm} (D9)

\[ a_1 = 164028 \]
\[ a_2 = -953.1 \]
\[ a_3 = 0.06661 \]
\[ a_7 = 0.0000001052 \]
\[ a_4 = 0.07997 \]
\[ a_5 = -0.001937 \]
\[ a_6 = -0.00003747 \]

D.2.6 Temperature \((T, ^\circ C)\) – Density of Refrigerant \((ROH, \text{kg/m}^3)\)

D.2.6.1 Liquid saturation line

\[ ROH = a_1 + a_2(T) + a_3(T)^2 + a_4(T)^3 + a_5(T)^4 + a_6(T)^5 \]  \hspace{1cm} (D10)

\[ a_1 = 1149 \]
\[ a_2 = -4.039 \]
\[ a_3 = -0.001111 \]
\[ a_7 = 6.619E-10 \]
\[ a_4 = 0.0002518 \]
\[ a_5 = -0.00006991 \]
\[ a_6 = -0.0000001268 \]

D.2.6.2 Vapor saturation line

\[ ROH = a_1 + a_2(T) + a_3(T)^2 + a_4(T)^3 + a_5(T)^4 + a_6(T)^5 \]  \hspace{1cm} (D11)

\[ a_1 = 31.55 \]
\[ a_2 = 1.057 \]
\[ a_3 = 0.01356 \]
\[ a_7 = 1.548E-09 \]
\[ a_4 = -0.00004312 \]
\[ a_5 = -3.002E-06 \]
\[ a_6 = 6.095E-08 \]

D.2.7 Temperature \((T, ^\circ C)\) – Pressure \((P, \text{MPa})\)

\[ P = a_1 + a_2(T) + a_3(T)^2 + a_4(T)^3 + a_5(T)^4 + a_6(T)^5 \]  \hspace{1cm} (D12)

\[ a_1 = 0.6311 \]
\[ a_2 = 0.02064 \]
\[ a_3 = 0.0001319 \]
\[ a_7 = -1.5858E-11 \]
\[ a_4 = -0.000001458 \]
\[ a_5 = 7.63E-08 \]
\[ a_6 = 8.333E-10 \]
D.3 Inputs for Prototype Heat Pump Cross Flow Dryer Simulation Model

Initial moisture content of material, $M_o = 70\%$

Initial air temperature before entering the dryer, $T_{ao} = 14^\circ C$

Total length of dryer bed, $y = 8$ m

Thickness of layer of material, $X = 0.025$ m

Mass flowrate of air, $G_a = 86$ kg/min-m$^2$ or 1000 cfm

Specific heat of air, $C_{pa} = 1.008$ kJ/kg-K

Specific heat of water vapour, $C_{pw} = 1.89$ kJ/kg-K

Inlet temperature of material, $T_{go} = 14^\circ C$

Latent heat of vaporization of free water, $L_w = 2501.64$ kJ/kg

Specific heat of material, $C_{pg} = 4.52$ kJ/kg-K

Specific heat of liquid, $C_{pl} = 4.186$ kJ/kg-K

Inlet humidity ratio of process air to the dryer, kg/kg of dry air

Inlet relative humidity of process air to the dryer, %

Amount of material dried, kg

Total time period for which the material was inside the drying chamber, min

D.4 Program code for prototype heat pump cross flow dryer

The present program was written for a prototype cross-flow heat pump dryer. Refer to the attached diskette for the program code. The program is given under the file name $prototypehp$. In this program both dryer and heat pump models are combined together to get a single heat pump dryer simulation model. This program
will not work if the temperature of refrigerant exceeds its critical limit and will show a message stating it. For details on the model, refer to Chapter 5.