BOILING HEAT TRANSFER IN HORIZONTAL MICRO-FIN TUBES

By

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Two existing evaporation two-phase heat transfer models are validated using 526 experimental data points for pure refrigerants and refrigerant mixtures. The Kido et al. (1995) model fails to predict pure refrigerant data sets except their R22 experimental data set. The Cavallini et al. (1999) model successfully predicts the available R22 data sets; however, the model over-predicts the R12 and the R134a data sets. In addition, the Cavallini et al. (1999) mixture model fails to predict the available 155 refrigerant mixture data points. The proposed modified model, based on the Cavallini et al. (1999) model, successfully predicts the experimental data for pure refrigerant and for refrigerant mixtures.
DEDICATION

I would like to dedicate this research to my parents, Kong Jin Tang and Choon Booi Gan, my sister, Ai Ling and my brother, Soon Woh, for their encouragements and endless guidance. This research is also dedicated to my lovely wife to be, Mee Huong Ding for her sincere and infinite support.
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# TABLE OF CONTENTS

<table>
<thead>
<tr>
<th>PAGE</th>
<th>CONTENT</th>
</tr>
</thead>
<tbody>
<tr>
<td>ii</td>
<td>DEDICATION</td>
</tr>
<tr>
<td>iii</td>
<td>ACKNOWLEDGMENTS</td>
</tr>
<tr>
<td>vi</td>
<td>LIST OF TABLES</td>
</tr>
<tr>
<td>vii</td>
<td>LIST OF FIGURES</td>
</tr>
<tr>
<td>x</td>
<td>NOMENCLATURE</td>
</tr>
<tr>
<td>1</td>
<td>CHAPTER I. INTRODUCTION</td>
</tr>
<tr>
<td>3</td>
<td>II. LITERATURE SURVEY</td>
</tr>
<tr>
<td>34</td>
<td>Correlations for Pure Refrigerants Flowing inside Smooth Tubes</td>
</tr>
<tr>
<td>10</td>
<td>Correlations for Pure Refrigerants Flowing inside Micro-Fin Tubes</td>
</tr>
<tr>
<td>15</td>
<td>Correlations for Refrigerant Mixtures Flowing inside Smooth Tubes</td>
</tr>
<tr>
<td>20</td>
<td>Correlations for Refrigerant Mixtures Flowing inside Micro-Fin Tubes</td>
</tr>
<tr>
<td>23</td>
<td>III. HEAT TRANSFER MODEL</td>
</tr>
<tr>
<td>27</td>
<td>Kido et al. (1995) Heat Transfer Model</td>
</tr>
<tr>
<td>34</td>
<td>Cavallini et al. (1999) Heat Transfer Model for Pure Refrigerants</td>
</tr>
<tr>
<td>41</td>
<td>Cavallini et al. (1999) Heat Transfer Model for Zeotropic Mixtures</td>
</tr>
<tr>
<td>49</td>
<td>IV. CORRELATION OPTIMIZATION</td>
</tr>
<tr>
<td>68</td>
<td>V. CRITIQUES ON THE CAVALLINI et al. (1999) HEAT TRANSFER MODEL AND FUTURE WORK RECOMMENDATIONS</td>
</tr>
<tr>
<td>72</td>
<td>VI. CONCLUSIONS</td>
</tr>
<tr>
<td>74</td>
<td>REFERENCES</td>
</tr>
</tbody>
</table>
APPENDIX

A MathCAD Files for the Kido et al. (1995) Model ................................................................. 78
Sample Data File .......................................................................................................................... 79
Sample Model File ....................................................................................................................... 82
Sample Property File .................................................................................................................... 83
Sample Calculation File .............................................................................................................. 85

B MathCAD Files for the Cavallini et al. (1999) Pure Refrigerant Model ............................... 88
Sample Model File ....................................................................................................................... 89
Sample Calculation File .............................................................................................................. 91

C MathCAD Files for the Cavallini et al. (1999) Refrigerant Mixture Model ......................... 95
Sample Model File ....................................................................................................................... 96
Sample Calculation File .............................................................................................................. 99

D MathCAD Worksheet for Pure Refrigerant Correlation Optimization ............................... 102

E MathCAD Worksheet for Refrigerant Mixture Correlation Optimization .......................... 104
LIST OF TABLES

<table>
<thead>
<tr>
<th>TABLE</th>
<th>Description</th>
<th>Page</th>
</tr>
</thead>
<tbody>
<tr>
<td>3.1</td>
<td>Flow Conditions for Pure Refrigerants Flowing inside Micro-fin Tubes</td>
<td>24</td>
</tr>
<tr>
<td>3.2</td>
<td>Tube Geometries for Pure Refrigerants Flowing inside Micro-fin Tubes</td>
<td>25</td>
</tr>
<tr>
<td>3.3</td>
<td>Flow Conditions for Refrigerant Mixtures Flowing inside Micro-fin Tubes</td>
<td>26</td>
</tr>
<tr>
<td>3.4</td>
<td>Tube Geometries for Refrigerant Mixtures Flowing inside Micro-fin Tubes</td>
<td>26</td>
</tr>
<tr>
<td>3.5</td>
<td>Mean Absolute Deviation Between the Experimental Data and the Predicted Data from the Kido et al. (1995) Model</td>
<td>33</td>
</tr>
<tr>
<td>3.6</td>
<td>Empirical Constants in the Cavallini et al. (1999) Model</td>
<td>34</td>
</tr>
<tr>
<td>3.7</td>
<td>Range of Experimental Conditions</td>
<td>34</td>
</tr>
<tr>
<td>3.8</td>
<td>Mean Absolute Deviation (MAD) Between the Experimental Data and the Prediction Data from the Cavallini et al. (1999) Pure Refrigerant Model</td>
<td>40</td>
</tr>
<tr>
<td>3.9</td>
<td>Mean Absolute Deviation (MAD) Between the Experimental Data and the Prediction Data from the Cavallini et al. (1999) Mixture Model</td>
<td>47</td>
</tr>
<tr>
<td>4.1</td>
<td>Independent Parameters in the Cavallini et al. (1999) Pure Refrigerant Model</td>
<td>50</td>
</tr>
<tr>
<td>4.2</td>
<td>Empirical Constants in the Cavallini et al. (1999) Pure Refrigerant Model and the Modified Pure Refrigerant Model</td>
<td>51</td>
</tr>
<tr>
<td>4.3</td>
<td>Mean Absolute Deviation (MAD) Achieved by the Cavallini et al. (1999) Pure Refrigerant Model and the Modified Pure Refrigerant Model</td>
<td>52</td>
</tr>
<tr>
<td>4.4</td>
<td>Additional Independent Parameters Required in the Modified Refrigerant Mixture Model</td>
<td>62</td>
</tr>
<tr>
<td>4.5</td>
<td>Empirical Constants in the Modified Refrigerant Mixture Model</td>
<td>63</td>
</tr>
<tr>
<td>4.6</td>
<td>Mean Absolute Deviation (MAD) Achieved by the Cavallini et al. (1999) Mixture Model and the Modified Mixture Model</td>
<td>63</td>
</tr>
<tr>
<td>FIGURE</td>
<td>Description</td>
<td>Page</td>
</tr>
<tr>
<td>--------</td>
<td>--------------------------------------------------------------------------------------------------------</td>
<td>------</td>
</tr>
<tr>
<td>2.1</td>
<td>Heat Transfer Model for Binary Mixture</td>
<td>19</td>
</tr>
<tr>
<td>3.1</td>
<td>Kido et al. (1995) Model on the Kido et al. (1995) Data Set (with dry-out data points)</td>
<td>29</td>
</tr>
<tr>
<td>3.2</td>
<td>Kido et al. (1995) Model on the Kido et al. (1995) Data Set (without dry-out data points)</td>
<td>29</td>
</tr>
<tr>
<td>3.4</td>
<td>Kido et al. (1995) Model on the Kuo and Wang (1996a) Data Set (constant $G$, increasing $q$)</td>
<td>31</td>
</tr>
<tr>
<td>3.5</td>
<td>Kido et al. (1995) Model on the Kuo and Wang (1996a) Data Set (constant $q$, increasing $G$)</td>
<td>31</td>
</tr>
<tr>
<td>3.10</td>
<td>Cavallini et al. (1999) Model on the Bogart and Thors (1999) R134a Data Set</td>
<td>37</td>
</tr>
<tr>
<td>3.11</td>
<td>Cavallini et al. (1999) Model on the Eckels et al. (1991) R134a Data Set</td>
<td>38</td>
</tr>
</tbody>
</table>
3.15 Cavallini et al. (1999) Mixture Model on the Murata and Hashizume (1993) R123/R134a (90/10 mole%) Data Set ............... 42

3.16 Cavallini et al. (1999) model on the Bogart and Thors (1999) R407c Data Set ............................................................... 43


3.18 Cavallini et al. (1999) Model on the Kuo and Wang (1996a) R407c Data Set (Constant $G = 200 \text{ kg/m}^2\cdot\text{s}$, $q = 6, 10, \text{ and } 14 \text{ kW/m}^2$) ....................................................... 45

3.19 Cavallini et al. (1999) Model on the Kuo and Wang (1996a) R407c Data Set (Constant $q = 10 \text{ kW/m}^2$, $G = 100, 200, \text{ and } 300 \text{ kg/m}^2\cdot\text{s}$) ...................................................... 45

3.20 Cavallini et al. (1999) Model on the Cui et al. (1986) R502 Data Set ................................................................................. 46

3.21 Cavallini et al. (1999) Model on the Bogart and Thors (1999) R507a Data Set ................................................................. 47

4.1 Comparison Between the Cavallini et al. (1999) and the Modified Pure Refrigerant Models on the Eckel et al. (1991) R12 Data Set .................................................................................. 53

4.2 Comparison Between the Cavallini et al. (1999) and the Modified Pure Refrigerant Models on the Murata and Hashizume (1993) R123 Data Set ................................................................. 53

4.3 Comparison Between the Cavallini et al. (1999) and the Modified Pure Refrigerant Models on the Eckels et al. (1991) R134a Data Set .................................................................................. 54

4.4 Comparison Between the Cavallini et al. (1999) and the Modified Pure Refrigerant Models on the Eckels et al. (1994) R134a Data Set .................................................................................. 55

4.5 Comparison Between the Cavallini et al. (1999) and the Modified Pure Refrigerant Models on the Eckels et al. (1998a) R134a Data Set .................................................................................. 55

4.6 Comparison Between the Cavallini et al. (1999) and the Modified Pure Refrigerant Models on the Eckels et al. (1998b) R134a Data Set .................................................................................. 56

4.7 Comparison Between the Cavallini et al. (1999) and the Modified Pure Refrigerant Models on the Bogart and Thors (1999) R134a Data Set .................................................................................. 57
4.8 Comparison Between the Cavallini et al. (1999) and the Modified Pure Refrigerant Models on the Bogart and Thors (1999) R22 Data Set ................................................................. 57
4.9 Comparison Between the Cavallini et al. (1999) and the Modified Pure Refrigerant Models on the Bogart and Thors (1994) R22 Data Set ........................................................ 58
4.10 Comparison Between the Cavallini et al. (1999) and the Modified Pure Refrigerant Models on the Hitachi Cable (1987) R22 Data Set .......................................................... 59
4.11 Comparison Between the Cavallini et al. (1999) and the Modified Pure Refrigerant Models on the Muzzio et al. (1998) R22 Data Set .................................................................. 60
4.12 Comparison Between the Cavallini et al. (1999) and the Modified Pure Refrigerant Models on the Schlager (1988) R22 Data Set .................................................................... 60
4.13 Comparison Between the Cavallini et al. (1999) and the Modified Pure Refrigerant Models on the Yasuda (1990) R22 Data Set ................................................................. 61
4.14 Comparison Between the Cavallini et al. (1999) and the Modified Mixture Models on the Bogart and Thors (1999) R407c Data Set ............................................................. 64
4.15 Comparison Between the Cavallini et al. (1999) and the Modified Mixture Models on the Ebisu and Torikoshi (1998) R407c Data Set ......................................................... 65
4.16 Comparison Between the Cavallini et al. (1999) and the Modified Mixture Models on the Cui et al. (1986) R502 Data Set ................................................................. 66
4.17 Comparison Between the Cavallini et al. (1999) and the Modified Mixture Models on the Bogart and Thors (1999) R507a Data Set ......................................................... 67
5.1 Detailed Cross Sectional of a Micro-fin Tube ................................................................. 69
**NOMENCLATURE**

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>$A$</td>
<td>Total heat transfer surface area ($m^2$)</td>
</tr>
<tr>
<td>$A_c$</td>
<td>Cross-sectional flow area ($m^2$)</td>
</tr>
<tr>
<td>$A_f$</td>
<td>Cross-sectional fin area including wall ($m^2$)</td>
</tr>
<tr>
<td>AreaRatio</td>
<td>Dimensionless parameter ($A_{mf}/A_{sm}$) describes the increase rate of the heat transfer surface area due to micro-fin (equation 3.3)</td>
</tr>
<tr>
<td>$Bo$</td>
<td>Boiling number (equation 2.6)</td>
</tr>
<tr>
<td>$B_{on}^W$</td>
<td>Bond number adopted from Webb (1988) (equation 2.36)</td>
</tr>
<tr>
<td>$B_{on}^*$</td>
<td>Modified bond number used in Kido et al. (1995) correlation. (equation 2.26)</td>
</tr>
<tr>
<td>$b$</td>
<td>Width of micro-fin valley bottom (m)</td>
</tr>
<tr>
<td>$bd$</td>
<td>Bubble diameter</td>
</tr>
<tr>
<td>$Co$</td>
<td>Convective number (equation 2.5)</td>
</tr>
<tr>
<td>$Coef$</td>
<td>Number of Coefficients (empirical constants) (equation 4.1)</td>
</tr>
<tr>
<td>$c_p$</td>
<td>Specific heat (J/kg-K)</td>
</tr>
<tr>
<td>$d_i$</td>
<td>Inner-tube diameter (m)</td>
</tr>
<tr>
<td>$d_h$</td>
<td>Hydraulic diameter (m)</td>
</tr>
<tr>
<td>$d_{mean}$</td>
<td>Mean inner-tube diameter (m)</td>
</tr>
<tr>
<td>$d_o$</td>
<td>Outer-tube diameter (m)</td>
</tr>
<tr>
<td>$E$</td>
<td>Enhancement factor (equation 2.2)</td>
</tr>
<tr>
<td>$e$</td>
<td>Micro-fin fin height (m)</td>
</tr>
<tr>
<td>$Fr$</td>
<td>Froude number (equation 2.7)</td>
</tr>
<tr>
<td>$G$</td>
<td>Total mass flux ($kg/m^2\cdot s$)</td>
</tr>
<tr>
<td>$g$</td>
<td>Gravitational acceleration ($m/s^2$)</td>
</tr>
<tr>
<td>Symbol</td>
<td>Description</td>
</tr>
<tr>
<td>--------</td>
<td>-------------</td>
</tr>
<tr>
<td>$h_{cb}$</td>
<td>Heat transfer coefficient for convective boiling region (W/m$^2$-K)</td>
</tr>
<tr>
<td>$h_l$</td>
<td>Heat transfer coefficient for liquid phase only (W/m$^2$-K)</td>
</tr>
<tr>
<td>$h_{nb}$</td>
<td>Heat transfer coefficient for nucleate boiling region (W/m$^2$-K)</td>
</tr>
<tr>
<td>$h_{pb}$</td>
<td>Heat transfer coefficient for pool boiling (W/m$^2$-K)</td>
</tr>
<tr>
<td>$h_{ip}$ or $h$</td>
<td>Two-phase heat transfer coefficient (W/m$^2$-K)</td>
</tr>
<tr>
<td>$i_{fg}$</td>
<td>Specific enthalpy of vaporization (J/kg)</td>
</tr>
<tr>
<td>$i_{fg, m}$</td>
<td>Specific enthalpy of vaporization for refrigerant mixture at isobaric condition (J/kg)</td>
</tr>
<tr>
<td>$k$</td>
<td>Thermal conductivity (W/m-K)</td>
</tr>
<tr>
<td>$L$</td>
<td>Heated test section length (m)</td>
</tr>
<tr>
<td>$M$</td>
<td>Molecular weight</td>
</tr>
<tr>
<td>$MAD$</td>
<td>Mean absolute deviation (equation 3.1)</td>
</tr>
<tr>
<td>$N$</td>
<td>Total number of data points</td>
</tr>
<tr>
<td>$n_f$</td>
<td>Number of micro-fins</td>
</tr>
<tr>
<td>$P$</td>
<td>Pressure (Pa)</td>
</tr>
<tr>
<td>$P_r$</td>
<td>Reduced pressure</td>
</tr>
<tr>
<td>$P_c$</td>
<td>Critical pressure</td>
</tr>
<tr>
<td>$Pr$</td>
<td>Prandtl number (dimensionless)</td>
</tr>
<tr>
<td>$p$</td>
<td>Micro-fin pitch (m)</td>
</tr>
<tr>
<td>$q$</td>
<td>Surface heat flux (W/m$^2$)</td>
</tr>
<tr>
<td>$Re$</td>
<td>Reynolds number (dimensionless)</td>
</tr>
<tr>
<td>$Rx$</td>
<td>Geometry parameter proposed by Hori and Shinohara (1991) (equation 2.35)</td>
</tr>
<tr>
<td>$S$</td>
<td>Suppression factor (equation (2.2))</td>
</tr>
<tr>
<td>$S_p$</td>
<td>Perimeter of one fin and channel taken perpendicular to the axis of the fin (m)</td>
</tr>
<tr>
<td>$SER$</td>
<td>Standard error of regression function introduced by Montgomery and Peck (1992) (equation 4.1)</td>
</tr>
<tr>
<td>$T$</td>
<td>Temperature (K)</td>
</tr>
</tbody>
</table>
$T_c$  Critical Temperature

$\Delta T$  Wall superheat (K)

$\Delta T_m$  Wall superheat for refrigerant mixture (K)

$\Delta T G$  Temperature glide for refrigerant mixture effect (K)

$t$  Width of micro-fin top (m)

$th$  Tube wall thickness (without micro-fin) (m)

$w$  Velocity (m/s)

$X$  Liquid-phase composition based on mole for more volatile component

$X_{H}$  Martinelli Parameter (equation 2.4)

$x$  Vapor quality

$\Delta$  Change in vapor quality

$Y$  Vapor-phase composition based on mole for more volatile component

$YY$  A factor proposed by Yoshida et al. (1983) for horizontal in-tube flow boiling (equation 2.23)

Greek Symbols

$\alpha$  Thermal diffusivity ($m^2/s$)

$\beta$  Micro-fin apex angle ($^\circ$, degree)

$\gamma$  Micro-fin helix angle ($^\circ$, degree)

$\lambda$  Mass transfer coefficient

$\mu$  Dynamic viscosity (N·s/m$^2$)

$\nu$  Kinematic viscosity (m$^2$/s)

$\phi$  Two-phase multiplier from Cavallini et al. (1999)

$\psi$  Dimensionless parameter proposed by Shah (1977)

$\rho$  Density (kg/m$^3$)

$\sigma$  Surface tension (N/m)
\[ \theta \quad \text{Contact angle (°, degree)} \]

Subscripts

\( b \quad \text{Local bubble point} \)
\( cb \quad \text{Convective boiling} \)
\( \text{experimental} \quad \text{Experimental results} \)
\( f \quad \text{Fluid} \)
\( i \quad \text{Liquid-vapor interface} \)
\( i \quad \text{Counter} \)
\( ib \quad \text{Incipient boiling condition} \)
\( \text{ideal} \quad \text{Ideal condition} \)
\( in \quad \text{Inlet} \)
\( l \quad \text{Liquid-phase only} \)
\( lf \quad \text{Liquid film} \)
\( m \quad \text{Refrigerant mixtures} \)
\( mf \quad \text{Micro-fin tube} \)
\( nb \quad \text{Nucleate boiling} \)
\( \text{out} \quad \text{Outlet} \)
\( pb \quad \text{Pool boiling} \)
\( \text{predicted} \quad \text{Prediction results} \)
\( sat \quad \text{Saturation} \)
\( sh \quad \text{Sensible heating} \)
\( sm \quad \text{Smooth tube} \)
\( T \quad \text{Total} \)
\( tp \quad \text{Two-phase} \)
\( v \quad \text{Vapor-phase only} \)
\( w \quad \text{Wall} \)
<table>
<thead>
<tr>
<th></th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>More volatile component in binary mixture</td>
</tr>
<tr>
<td>2</td>
<td>Less volatile component in binary mixture</td>
</tr>
</tbody>
</table>
CHAPTER I
INTRODUCTION

Micro-fin tubes have been used extensively in the commercial air-conditioning industry since the early eighties. The performance of micro-fin tubes has been determined to be more effective than that of conventional smooth tubes in cooling capability. Chamra et al. (1996) conducted experiments that showed that micro-fin tubes with helix angles, $\chi$ of 15° to 27° have nearly two times the heat transfer coefficient as compared to conventional smooth tubes. Their results also showed that the pressure drops for all the tested micro-fin tubes increased by less than 50%. Chamra et al. (1996) experimental results proved that the implementation of micro-fin tubes are more efficient than conventional smooth tubes. For the past decade, many researchers have been involved in determining the performance of pure refrigerants in smooth tubes and micro-fin tubes. The heat transfer coefficients for pure refrigerants in both smooth and micro-fin tubes have been evaluated quite successfully. However, the implementation of the Montreal Protocol minimized the usage and eliminated many pure refrigerants that have the potential to deplete the ozone layer. Those include chlorine-containing refrigerants such as CFC12 (Chlorofluorocarbon 12) and HCFC22 (Hydrochlorofluorocarbon 22).

New alternative refrigerants which have less harmful effect on the ozone layer have been introduced. These alternative refrigerants include pure HFC (Hydrofluorocarbon) refrigerants and mixture of HFC refrigerants. The efficiencies of these alternative refrigerants are not clearly understood and only limited research have been conducted on them. According to Cavallini et al. (1999), these new refrigerant mixtures generally exhibit heat transfer degradation during the evaporation process because of mass transfer diffusion, sensible heating effects, and non-equilibrium effects in two-phase flow (liquid and vapor phases). At a constant saturation pressure, a refrigerant mixture (except an azeotrope) evaporates or
condenses within a range of saturation temperatures. This temperature range is called the “temperature glide.” Temperature glide is defined as the temperature difference between the bubble-point temperature and the dew-point temperature. However, a pure refrigerant evaporates and condenses at a constant saturation temperature at an isobaric condition. Refrigerant mixtures have different temperature glides for different molar compositions. There are two types of refrigerant mixtures, azeotropic and zeotropic. Over the entire composition range for an azeotropic mixture, there is a particular composition for which the azeotropic mixture behaves exactly like a pure refrigerant. In other words, for this particular composition, an azeotropic mixture undergoes two-phase processes with a constant saturation temperature at an isobaric condition. At this particular composition, the azeotropic mixture is called “azeotrope.” In reality, a perfect azeotrope is highly uncommon and its specific composition is extremely hard to attain. A zeotropic mixture behaves relatively similar to an azeotropic mixture. However, for the entire molar composition range for a zeotropic mixture, there is no single composition that allows the zeotropic mixture to behave like the azeotrope.

The main purpose of the current research is to carefully review the previous research concerning the in-tube boiling for smooth and micro-fin tubes using pure refrigerants and refrigerant mixtures. Two existing evaporative heat transfer models are further analyzed and evaluated to ensure their validity for the existing experimental database. Improvements to the existing heat transfer model are proposed. The recommended heat transfer model will be a general model that is capable of predicting heat transfer coefficients for different refrigerants, including pure refrigerants and refrigerant mixtures, flowing inside different configurations of micro-fin tubes.
CHAPTER II
LITERATURE SURVEY

Since late eighties, many correlations have been introduced to predict the in-tube boiling heat transfer coefficients for horizontal tubes. Due to the complexity of the heat transfer process in in-tube flow boiling, many proposed mathematical models are empirical or semi-empirical. According to Darabi et al. (1995), the proposed correlations can be classified into four main categories:

- models based on dimensional analysis
- models based on the addition of the nucleate boiling contribution and the convective boiling contribution
- models that use the larger of the nucleate boiling component or the convective boiling component based on the dimensionless parameter, the boiling number ($Bo$)
- asymptotic models based on a power-type addition of the nucleate boiling component and the convective boiling component.

The following literature review is presented in four parts: the correlations for pure refrigerant inside smooth tubes, the correlations for pure refrigerant inside micro-fin tubes, the correlations for refrigerant mixtures inside smooth tubes, and the correlations for refrigerant mixtures inside micro-fin tubes.

Correlations for Pure Refrigerant Flowing inside Smooth Tubes

The first general correlation for in-tube saturated flow boiling was proposed by Chen (1966). This correlation was proposed for vertical tube application only. However, the Chen correlation served as the basis for future development of both vertical and horizontal tube applications. Chen proposed that
two-phase heat transfer for saturated flow boiling comprises two basic mechanisms, convective boiling and nucleate boiling. The convective boiling component is dependent on fluid flow, and the nucleate boiling component is associated with bubble growth and bubble nucleation. These two mechanisms were also determined to be additive in their contributions to the two-phase heat transfer coefficient. The Chen model is written as

$$h_p = h_{i,h} + h_{p,h}$$

$$h_p = h_i \times E + h_{p,h} \times S$$

The factors $E$ and $S$ are the convective boiling enhancement factor and the nucleate boiling suppression factor, respectively. The convective heat transfer coefficient for liquid phase only, $h_i$, was proposed to be evaluated from Dittus-Boelter (1930) equation.

$$h_i = 0.023 \left( \frac{k_i}{d_i} \right) \times \text{Re}_i^{0.8} \times \text{Pr}_i^{0.4}$$

The pool boiling heat transfer coefficient, $h_{p,h}$, was determined from the Forster and Zuber (1955) pool boiling correlation.

The enhancement factor $E$, is always greater than unity since the two-phase heat transfer coefficient is greater than the single-phase heat transfer coefficient. In two-phase flow, the heat transfer is enhanced due to the increase in vapor quality and the decrease in liquid film. The higher heat transfer coefficient in two-phase flow is also due to the higher fluid velocity in two-phase flow. Chen correlated the $E$- factor with the Martinelli parameter, $X_{ii}$.

$$X_{ii} = \left( \frac{1-x}{x} \right)^{0.9} \times \left( \frac{\rho_v}{\rho_l} \right)^{0.5} \times \left( \frac{\mu_v}{\mu_l} \right)^{0.1}$$

The suppression factor $S$, always takes a value less than unity. As the vapor quality increases, the nucleate boiling effect is greatly suppressed because of the thinner liquid film. The convective boiling contribution becomes dominant as the vapor quality increases. The $S$ factor is correlated with the two-phase Reynolds number.
The Chen correlation for vertical tube application was based on 665 data points for water and hydrocarbons. The mean deviation for these data is 12%. Recent researchers, for example, Kandlikar (1990) and Liu and Winterton (1991) found mean deviation of more than 40% for the Chen model.

Shah (1977) proposed a correlation for boiling in vertical, horizontal tubes and annuli. The correlation is based on three flow-boiling regimes:

a. the nucleate boiling dominated
b. the bubble suppression
   (both nucleate boiling and convective terms are equally important)
c. the convective dominated

Shah introduced a new dimensionless parameter, $\Upsilon$ which is the ratio of the two-phase heat transfer coefficient and the heat transfer coefficient for the liquid phase only. He also wrote the two-phase heat transfer coefficient as a function of the convective number, $Co$, the boiling number, $Bo$, and the Froude number, $Fr$.

Shah determined that the viscosity has no significant effect in determining the two-phase heat transfer coefficient, and he replaced the Martinelli parameter with a new parameter, the convective number, $Co$.

\[
Co = \left( \frac{1}{x} - 1 \right)^{0.8} \left( \frac{\rho_s}{\rho_l} \right)^{0.5}
\]

(2.5)

The boiling number, $Bo$, and the Froude number, $Fr$, are two commonly used dimensionless terms.

\[
Bo = \frac{q}{G \sqrt{\gamma}}
\]

(2.6)

\[
Fr = \frac{G^2}{\rho \sqrt{\gamma dx}}
\]

(2.7)

The single-phase heat transfer coefficient is determined from the Dittus-Boelter (1930) equation.

The Shah correlation was based on 500 data points from 29 data sets with 18 independent experimental studies with a mean deviation of 9.5%. Recent researchers, Kandlikar (1990) and Gungor and
Winterton (1986) found that the Shah correlation has a mean deviation around 25%, which is a relatively good prediction.

The Shah correlation is relatively effective in predicting the subcooled flow boiling in pipes and annuli for both horizontal and vertical configurations. This model is not further analyzed, as the main objective for the present study is to concentrate on the saturated boiling region.

Bjorge et al. (1982) proposed a correlation for in-tube saturated flow boiling using the heat flux superposition equation.

\[ q = q_s + q_n \left[ 1 - \left( \frac{T_w - T_{sat}}{T_w - T_{sat,ln}} \right)^3 \right] \]  

(2.8)

The heat flux superposition equation consists of the convective boiling heat flux and the nucleate boiling heat flux. In this correlation, the authors introduce a fluid-dependent empirical constant, \( B_M \). The two-phase heat transfer coefficient is determined by dividing the total heat flux by the wall superheat. This correlation was compared with 8 sets of data for water and had a mean deviation of 15%. The presence of the fluid dependent factor, \( B_M \), limits the usage of this correlation. Gungor and Winterton (1986) validated this correlation with their data bank of 3700 data points, and they found the correlation to have a standard deviation greater than 55%. Kandlikar (1990) found that this correlation has a mean deviation of more than 50%.

Gungor and Winterton (1986) further refined the heat transfer models proposed by Chen (1966) and Shah (1977). Gungor and Winterton used the same additive model as in equation 2.1 and their model was proposed for saturated flow boiling in vertical and horizontal tubes. With minor modification to the tube diameter, the correlation was also applicable for flow boiling in annuli. The enhancement factor \( E \) in the model was correlated using the Martinelli parameter, \( X_m \), and the boiling number, \( Bo \). The boiling number was introduced to account for the significant heat transfer enhancement in the two-phase flow, particularly in the nucleate boiling regime. This heat transfer enhancement further increases the total heat transfer rate. The suppression factor, \( S \), was correlated using the two-phase Reynolds number. The two-phase Reynolds number is written as the product of the enhancement factor, \( E \), and the liquid-phase
Reynolds number, $Re_t$. The suppression factor, $S$, was proposed to account for heat transfer suppression that is controlled by the effectiveness of the convection boiling heat transfer, $h_{cb}$. The Cooper (1984) correlation for pool boiling heat transfer coefficient was used in Gungor and Winterton (1986) correlation. The Froude number, $Fr$, was also incorporated in the correlation to account for tube orientation (horizontal or vertical). Kandlikar (1990) validated the Gungor and Winterton (1986) correlation with his 5000 data points from 24 data sets, and the model achieved a mean deviation around 25%.

Klimenko (1988) introduced a correlation for convective boiling heat transfer for both vertical and horizontal channels. His correlation is in dimensionless form, and his approach is quite similar to Shah’s (1977) approach. Klimenko further refined his correlation in 1990. Klimenko (1990) correlation is restricted to wetted tube wall, and the heat transfer is assumed to be independent of channel diameter. Klimenko proposed his correlation based on 3125 data points from 75 sources for 21 different fluids including water, organic fluids, refrigerants, and cryogens. His model achieved a mean deviation of 14.4% compared with the experimental data.

The Jung et al. (1989) correlation used the Chen (1966) model as the basis for their correlation. This correlation is also extended to predict the heat transfer coefficient for flow boiling inside smooth tubes with refrigerant mixtures. The mixture effects in the correlation will be discussed in the next section. For pure refrigerant flowing inside a smooth tube, Jung et al. (1989) observed the same behavior as discussed by Chen (1966). The observation during the boiling process suggested that there are two main regions that exist over a wide range of vapor quality. The heat transfer coefficient is a strong function of heat flux in the nucleate boiling region and the heat transfer coefficient is independent of the heat flux in the convective boiling region.

Jung et al. (1989) also determined that in the region where convective boiling is dominant and the nucleate boiling contribution is suppressed, the two-phase heat transfer coefficient $h_{tn}$, is proportional to mass flux to the eight-tenths power, $G^{0.8}$. The convective boiling component is evaluated from the single-phase heat transfer coefficient with an enhancement factor. The Dittus-Boelter (1930) equation was chosen for the single-phase heat transfer coefficient because of its simplicity and popularity as compared to
Petukhov equation (1970). The enhancement factor, $E$, was then determined as a function of the Martinelli parameter. The authors proved that the $E$ factor is independent of $Pr$ number, and this result was supported by the analytical study from Kunz and Yerazunis (1969). The $E$ factor is determined by a regression analysis based on experimental data for the convective boiling region only with a correlation coefficient of 0.999. The $E$ factor proposed by Jung et al. (1989) is written as equation 2.9.

$$E = 2.37 \times \left(0.29 + \frac{1}{\chi_n}\right)^{0.85}$$  \hspace{1cm} (2.9)

For the region where nucleate boiling is dominant, the nucleate boiling component is a function of quality, heat flux, and mass flow rate. This region was modeled by a pool boiling heat transfer coefficient with a suppression factor. The Stephan and Abdelsalam (1980) pool boiling correlation was selected because of its accuracy in predicting the pool boiling heat transfer coefficient for a refrigerant. The Stephan and Abdelsalam pool boiling correlation is as follows:

$$h_{pb} = 207 \frac{k_j}{bd} \left(\frac{q \times bd}{k_j \times d_{at}}\right)^{0.745} \left(\frac{\rho_v}{\rho_l}\right)^{0.581} \times Pr_0^{0.533}$$  \hspace{1cm} (2.10)

$$bd = 0.0146 \times \theta \left(\frac{2 \times \sigma}{g \times (\rho_l - \rho_v)}\right)^{0.5} \hspace{1cm} \text{with} \hspace{0.5cm} \theta = 35^\circ$$  \hspace{1cm} (2.11)

The Martinelli parameter and the boiling number were chosen to model the suppression factor, which accounts for the changes in quality, heat flux, and mass flow rate. The suppression factor is

$$S = 4048 \times X_a^{1.22} \times Bo^{1.13} \hspace{1cm} \text{for} \hspace{0.5cm} X_a \leq 1$$  \hspace{1cm} (2.12)

$$S = 2.0 - 0.1 \times X_a^{-0.28} \times Bo^{-0.33} \hspace{1cm} \text{for} \hspace{0.5cm} 1 < X_a \leq 5$$  \hspace{1cm} (2.13)

The Jung et al. (1989) correlation is applicable for pure refrigerants and for azeotropic mixtures at the azeotrope. The mass flux for this correlation is evaluated from the liquid-phase mass flux, $G(1-x)$ instead of the total mass flux, $G$. This correlation successfully predicts the two-phase heat transfer coefficient for the pure refrigerants flowing inside smooth tubes with the mean deviation around 7%.

Liu and Winterton (1991) proposed a correlation for in-tube flow boiling based on Kutateladze’s power-type addition model (1961). The correlation is based on 4200 saturated flow boiling data points.
from 30 different sources, and the working fluids included water, refrigerants and some hydrocarbons. The 
correlation is listed as equation 2.14.

\[
h_{pg} = \left( e_F \times E \times h_l \right)^2 + \left( e_S \times S \times h_{ps} \right)^{1/2}
\]  

(2.14)

Compared with the previously discussed addition models, this correlation further suppressed the nucleate 
boiling effect as the vapor quality increases. Both the \( E \) and \( S \) factors were determined by regression 
analysis. The liquid-phase and pool boiling heat transfer coefficients were determined from Dittus-Boelter 
(1930) equation and the Copper (1984) correlation, respectively. The \( e_F \) and \( e_S \) factors were written as 
a function of Froude number to account for the horizontal flow. The Liu and Winterton (1991) correlation 
predicted the two-phase heat transfer coefficient for the experimental data with a mean deviation around 
20%.

Murata and Hashizume (1993) recommended a correlation for in-tube flow boiling for pure and 
refrigerant mixtures. Their model was also extended to evaluate the performance of micro-fin tubes. The 
Murata and Hashizume model for pure refrigerant flowing inside smooth tubes is discussed in this section. 
The remaining correlations will be discussed in the next section.

The Murata and Hashizume model was developed based on the Chen (1966) model as in equations 
2.1 and 2.2. As recommended by Chen (1966), the two-phase heat transfer coefficient is the arithmetic sum 
of the convective boiling and the nucleate boiling components. The convective boiling component is 
evaluated from the liquid-phase heat transfer coefficient with an enhancement factor. Similarly, the 
nucleate boiling component is expressed as the product of a pool boiling correlation with a suppression 
factor. Murata and Hashizume (1993) adopted the Dittus-Boelter (1930) equation to evaluate the single- 
phase heat transfer coefficient, and the enhancement factor, \( E \), is correlated as a function of Martinelli 
parameter.

\[
E = 2.44 \left( \frac{1}{X_p} \right)^{0.863}
\]  

(2.15)

This enhancement factor closely agrees with the enhancement factor recommended by Jung et al. (1989) 
(equation 2.9).
The pool boiling heat transfer coefficient was obtained from the Nishikawa et al. (1982) correlation and is given as

$$ h_{pb} = 31.4 \times \left[ \frac{P_{c}^{0.2} \times F_p}{M^{0.1} \times \sqrt[3]{\gamma}} \right] \times q_{pb}^{0.8} $$

(2.16)

where

$$ F_p = \left( \frac{P}{P_c} \right)^{0.23} \left[ 1 - 0.99 \times \left( \frac{P}{P_c} \right)^{0.9} \right] $$

(2.17)

The suppression factor $S$, was obtained from the $S$ factor derived analytically by Bennett and Chen (1980) and is written as

$$ S = \frac{k_i}{h_{ch} \cdot \delta} \times \left[ 1 - \exp \left( - \frac{h_{ch} \cdot \delta}{k_i} \right) \right] $$

(2.18)

where

$$ \delta = C_{d} \times \left[ \frac{\sigma}{g} (\rho_f - \rho_g) \right]^{0.5} $$

(2.19)

The empirical constant $C_{d}$ was determined as 0.08 by Murata and Hashizume (1990).

Murata and Hashizume (1993) correlation predicted the two-phase heat transfer coefficient for the R123 experimental data flowing inside smooth tube with a relatively small mean deviation around 5% in the high mass flux region. The correlation failed to predict the two-phase heat transfer coefficient in the low mass flux region.

Correlations for Pure Refrigerants Flowing inside Micro-Fin Tubes

Kandlikar (1991) modeled the heat transfer coefficient for micro-fin tubes based on the greater component of nucleate boiling or convective boiling. His model was based on Kandlikar (1990) earlier work for horizontal and vertical smooth tube configurations. In his correlation, the nucleate boiling and the convective boiling components are in additive form. The boiling number, $Bo$, and the convective number, $Co$, are incorporated in the correlation for the nucleate boiling and the convective boiling components,
respectively. A fluid-dependent parameter, $F_{fr}$, was included in the nucleate boiling component. Two empirical constants, $E_{NB}$ and $E_{CB}$ were introduced to account for the tube geometry in the nucleate boiling and the convective boiling components, respectively. These two empirical constants are dependent on the tube geometry and independent of the refrigerant and the operating condition. The Kandlikar (1991) correlation used empirical constants to account for the heat transfer enhancement effect from micro-fin tubes. His correlation does not clearly describe the effect of tube geometries on the heat transfer coefficient in micro-fin tubes. In addition, the fluid-dependent parameter limits the general usage of his correlation, as each working fluid requires a different value for the fluid-dependent parameter. This correlation achieved a mean deviation around 10% with Kahnpara et al. (1987) data points.

As discussed in the previous section, Murata and Hashizume (1993) developed their correlation for micro-fin tubes in conjunction with their correlation proposed for pure refrigerant flowing inside smooth tube. Their correlation for micro-fin tubes was quite similar to the smooth tube correlation, but with slight modifications. The correlation remains in the form recommended by Chen (1966) as in equations 2.1 and 2.2. The single-phase heat transfer coefficient is evaluated in a form relatively similar to the Dittus-Boelter (1930) equation but with a different empirical constant

$$h_i = 0.036 \frac{k_i}{d_i} \times \text{Re}_i^{0.8} \times \text{Pr}_i^{0.4}$$

The enhancement factor $E$, was also modified to accommodate the enhancement due to the presence of micro-fins.

$$E = 2.2 \left( \frac{1}{X_n} \right)^{0.5}$$

The pool boiling heat transfer coefficient recommended by Nishikawa et al. (1982) was changed with an empirical constant of 48 instead of 31.4. This empirical constant was determined from the best fit to the data at a vapor quality of $x = 0.2$. 

\[ h_{pb} = 48 \times \left( \frac{P_r^{0.2} \times F_p}{M^{0.1} \times I_c^0.9} \right) \times q_{pb}^{0.8} \]  \hspace{1cm} (2.22)

The suppression factor \( S \), remained unchanged as it was shown in equations 2.18 and 2.19.

This correlation was tested on R123 experimental data, and the correlation successfully predicted the experimental two-phase heat transfer coefficient within \( \pm 20\% \). A higher deviation was observed at low mass flux and low heat flux regions. The deviation was likely caused by the capillary action in micro-fin tubes.

Kido et al. (1995) proposed a correlation for flow boiling in micro-fin tubes. Their correlation is developed based on their experimental data for pure refrigerant R22. The authors claim that the heat transfer coefficients in micro-fin surfaces must be evaluated from the real heat transfer surface area instead of the nominal surface area. The authors first determined the appropriate functional groups that affected the heat transfer performance and, then, evaluated the dependency of each group on the total heat transfer coefficient using their experimental data. Five parameters were identified as the independent variables in predicting the heat transfer coefficient. These parameters are listed as follows:

\[ YY = Bo \times 40^4 + 0.23 \times (Bo \times 40^4)^{0.69} \times X_p^{-2.0} \]  \hspace{1cm} (2.23)

\[ Re = \frac{G \times d_m}{\mu_i} \]  \hspace{1cm} (2.24)

\[ Pr_f = \frac{V}{\alpha} \]  \hspace{1cm} (2.25)

\[ Bon^* = \frac{p^2 \times g \times (\rho_L - \rho_v)}{\sigma} \times \left\{ \frac{p}{d_{mean}} \right\} \left( \frac{p}{e} \right) \left( \frac{p}{b} \right) \]  \hspace{1cm} (2.26)

\[ 1 + \tan(\gamma) \]  \hspace{1cm} (2.27)

The first parameter is factor \( YY \), which was proposed by Yoshida et al. (1983) for horizontal in-tube flow boiling. The factor \( YY \), is written as a function of the Martinelli parameter, \( X_p \), and the boiling number, \( Bo \). The second and the third parameters are the all-liquid-phase Reynolds number and the all-liquid-phase Prandtl number. The mean diameter, \( d_m \), is equivalent to the diameter of the smooth tube with the internal
cross-sectional area equal to that of the micro-fin tube. The modified bond number \( (Bon^*) \) is the factor that accounts for the micro-fin geometry. The last factor, \( l + \tan(\gamma) \), describes the dependency on the micro-fin helix angle. The final form of the correlation with the appropriate limitations is as follows:

\[
Nu_p = 2.5 \times \frac{Y^0.2 \times Re^{0.27} \times Pr^{0.4} \times (1 + \tan(\gamma))^{1.8}}{Bon^{0.2}}
\]

(2.28)

limitation: \( 1 < YY < 8 \times 10^2 \)
\( 2 \times 10^3 < Re < 1 \times 10^4 \)
\( 3 \times 10^3 < Bon^* < 4 \times 10^2 \)
\( 0 < \gamma < 20 \)

The Kido et al. (1995) correlation predicted 80% of their experimental two-phase heat transfer coefficient with an error around 20%.

Cavallini et al. (1999) proposed a heat transfer model for refrigerant evaporating inside micro-fin tubes based on Chen’s additive model and Cavallini et al. (1995) earlier work on refrigerant condensing inside micro-fin tubes. Their model is capable of predicting two-phase heat transfer coefficient for pure refrigerant and zeotropic refrigerant mixtures. The model for zeotropic refrigerant mixtures will be discussed in the following section.

The Cavallini et al. (1999) heat transfer model for a pure refrigerant is expressed as the sum of the nucleate boiling component and the convective boiling component as described in equations 2.1 and 2.2. Additional parameters have been added to equation 2.1 to account for the heat transfer enhancement due to the presence of micro-fins. The predicted two-phase heat transfer coefficient is defined with reference to the effective tube inner diameter, tube inner diameter minus two times the fin height, \( d_i - 2e \), and to the difference between wall temperature and the saturation temperature. All the thermodynamic and the transport properties of refrigerants were computed from REFPROP 5.12 developed by NIST (1996).

Cavallini et al. (1999) adopted the Cooper (1984) pool boiling correlation to evaluate the nucleate boiling component. The suppression factor, \( S \), is evaluated as a function of Martinelli parameter, \( X_m \). The factor \( F_1 \) was introduced by Steiner (1993) to correlate flow boiling experimental data in smooth tubes.
\[ h_{nb} = h_{pb} \times S \times F_1(d_x) \quad (2.29) \]

\[ h_{pb} = 55 \times P_r^{0.12} \times (-\log P_r)^{0.55} \times M^{0.5} \times q_{nb}^{0.67} \quad (2.30) \]

\[ S = A \times X_{ht}^B \quad \text{if } X_{ht} > 1, \text{ then } X_{ht} = 1 \quad (2.31) \]

\[ F_1(d_x) = \left( \frac{d_x}{d_i} \right)^{c} \quad d_x=0.01 \text{ m} \quad (2.32) \]

where \( A, B, \) and \( C, \) are empirical constants.

Cavallini et al. (1999) suggested that the condensation and evaporation processes in micro-fin tubes share the same mechanism and that the equation from forced convection condensation inside micro-fin tubes can be used for forced convective evaporation with minor modifications. Based on the Cavallini et al. (1995) correlation for condensation inside low-fin, micro-fin and cross-grooved tubes, the convective boiling component is written as

\[ h_{eb} = N_{tu} \times \Phi \times \frac{k}{d_i} \times R_x^{5.5} \times (Bon_{w} \times Fr_{r})^{T} \times F_1(d_x) F_3(G) \quad (2.33) \]

where \( \Phi = \left[ \frac{(1-x)+2.63 \times x}{\left( \frac{\rho_i}{\rho_r} \right)^{0.8}} \right] \quad (2.34) \)

\[ R_x = \frac{1}{\cos(\gamma)} \times \left[ \frac{2 \times \pi \times n_j \times \left( \frac{1-\sin \left( \frac{\beta}{j} \right)}{2} \right)}{\pi \times d_i \times \cos \left( \frac{\beta}{2} \right)} + 1 \right] \quad (2.35) \]

\[ Bon_{w} = \frac{g \times \rho_{l} \times \pi \times \pi \times d_i}{8 \times \sigma \times n_j} \quad (2.36) \]

\[ F_2(d_x) = \left( \frac{d_x}{d_i} \right)^{y} \quad d_x=0.01 \text{ m} \quad (2.37) \]

\[ F_3(G) = \left( \frac{G_i}{G} \right)^{z} \quad G_i=100 \text{ kg/m}^2\text{-s} \quad (2.38) \]
Nusselt number for all-liquid phase only, $\text{Nu}_{x}$, is evaluated from the Dittus-Boelter (1930) equation. $\Phi$ is the two-phase multiplier written as a function of the densities of the liquid and vapor phases and the quality. This parameter goes to one as $x$ goes to zero. $Rx$ is the factor suggested by Hori and Shinohara (1991), and the cosine of the helix angle $\gamma$ represents the effect of the heat transfer area increase. For helix angles greater than $30^\circ$, $\gamma$ is set to $30^\circ$ as there were insufficient data to verify the effect for $\gamma > 30^\circ$. The bond number $\text{Bon}_w$, adopted from Webb (1988), is used to account for surface tension effects. The constants $S^*, T, V,$ and $Z$ are determined empirically.

The Cavallini et al. (1999) correlation was validated on 643 experimental data points for pure refrigerants consist of R12, R22, R32, R123, R125, and R134a. Cavallini et al. (1999) selected the mean absolute deviation, $\text{MAD}$, as the performance criterion. $\text{MAD}$ is defined as the average of the normalized difference between the predicted two-phase heat transfer coefficient and the experimental two-phase heat transfer coefficient.

$$\text{MAD} = \frac{1}{N} \sum \frac{h_{\text{predicted}} - h_{\text{experimental}}}{h_{\text{experimental}}}$$

(2.39)

Their correlation predicted the experimental results with a mean absolute deviation around 14.1%.

Correlations for Refrigerant Mixtures Flowing inside Smooth Tubes

Jung et al. (1989) further modified their correlation for pure refrigerants flowing inside horizontal smooth tubes to include binary refrigerant mixture effects. The authors observed significant heat transfer degradation for the boiling process in a binary mixture in comparison to the boiling process in a pure refrigerant. The heat transfer degradation was mainly observed in the nucleate boiling region. As discussed by Scriven (1959) in his bubble-radius equation for binary mixtures, the bubble growth rate for a binary mixture is lower than that for a pure refrigerant. The heat transfer reduction is due to mass transfer resistance, and the loss of wall superheat from mass diffusivity. The heat transfer reduction is also caused
by the composition difference between the two phases ($Y$-$X$), where $X$ and $Y$ are the liquid and vapor phases composition based on mole for the more volatile component of the mixtures. Due to the complexity in Scriven’s equation with the presence of mass diffusivity for the liquid-phase parameter, Thome (1983) proposed a method to evaluate the binary mixture effect via the phase equilibrium data. In line with Thome’s (1983) reasoning, Unal (1986) developed the following correlation to account for the binary mixture in the form of a ratio of wall superheat for the mixture to wall superheat for the mixture at ideal condition:

$$\frac{\Delta T_w}{\Delta T_{\text{ideal}}} = C_m = [1 + (b_2 + b_3) \times (1 + b_4)] \times (1 + b_5)$$

(2.40)

where

$$\Delta T_{\text{ideal}} = X_1 \times \Delta T_1 + X_2 \times \Delta T_2$$

$$b_2 = (1 - X) \times \ln \left( \frac{1.01 - X}{1.01 - Y} \right) + X \times \ln \left( \frac{X}{Y} \right) + |Y - X|^{1.5}$$

$$b_3 = 0 \quad \text{for} \quad X \geq 0.01$$

$$b_3 = \left( \frac{Y}{X} \right)^{0.1} - 1 \quad \text{for} \quad X < 0.01$$

$$b_4 = 152 \times \left( \frac{P}{P_c,1} \right)^{3.9}$$

$$b_5 = 0.92 \times (Y - X)^{0.001} \times \left( \frac{P}{P_c,1} \right)^{0.66}$$

$$\frac{X}{Y} = 1 \quad \text{for} \quad X = Y = 0$$

The nucleate pool boiling heat transfer coefficient, $h_{pb,m}$, for a binary mixture is then determined as

$$\frac{h_{pb,m}}{h_{\text{ideal}}} = \frac{1}{C_m}$$

(2.41)

where

$$h_{\text{ideal}} = \frac{1}{h_1 + h_2 + \frac{X_1}{h_1} \times \frac{X_2}{h_2} \times h_1 \times h_2}$$
$h_1$ and $h_2$ are the pool boiling heat transfer coefficients for component 1 and 2 based on the Stephan and Abdelsalam correlation (1980).

In order to account for the nucleate boiling suppression at lower quality region for refrigerant mixtures, the suppression factor, $S$, derived in equation 2.12 or 2.13, is divided by $C_m$, and the nucleate boiling component for binary mixture then becomes

$$h_{bh} = \frac{S}{C_m} \cdot h_{bh,m}$$  \hfill (2.42)

For the convective boiling component, the binary mixture effect was found to be small. To account for this effect, which is the effect of the mass transfer resistance, phase equilibrium data ($Y-X$) were used.

$$h_c = E \cdot C_m \cdot h_t$$  \hfill (2.43)

where $E = \text{equation } 2.8$

$$C_m = 1 - 0.35(Y - X)^{1.56}$$

The Jung et al. (1989) correlation successfully predicted the experimental two-phase heat transfer coefficient for R22 and R114 mixtures and R12 and R152a mixtures at different compositions with a mean deviation around 9.6%. The authors claimed that the good prediction of this correlation is mainly due to the accuracy of the Unal correlation (1986) and the Stephan and Abdelsalam pool boiling correlation (1980).

Murata and Hushizume (1993) discussed the refrigerant mixtures effect for in-tube flow boiling for smooth and micro-fin tubes. The correlations for smooth tubes and micro-fin tubes are relatively similar to each other. With minor modifications for the smooth tube correlation, the correlation can be applied to micro-fin tubes. The minor modifications account for the heat transfer enhancement due to the presence of micro-fins. The same modification was observed in Murata and Hashizume (1993) refrigerant mixture correlation.

Murata and Hushizume (1993) identified degradations in the heat transfer coefficients in refrigerant mixtures as compared to pure refrigerants. The authors claimed that the degradation is due to
the mixture effects. Based on the heat transfer model developed by Sardesai et al. (1982), the reduction in heat transfer coefficient for refrigerant mixtures is mainly due to the sensible heating of the vapor phase plus the rise in the saturation temperature along the flow direction attributed to nucleate boiling.

The Murata and Hushizume (1993) correlation, with graphical explanation in figure 2.1, for refrigerant mixtures in smooth tubes was developed based on the following six assumptions,

1. The two-phase flow pattern is annular.
2. The temperature of the vapor phase, \( T_v \), is equal to the saturation temperature, \( T_{sat} \), at a given pressure, \( P_{sat} \), and equilibrium quality, \( x \).
3. The sensible heat transfer to the liquid phase is negligible compared to the latent heat of vaporization.
4. The heat transfer to the vapor phase is equal to the sensible heat used to change the temperature of the vapor phase as the saturation temperature rises in the flow direction.
5. The heat transfer coefficient of the liquid film, \( h_{lpr} \), is equal to the convective boiling heat transfer for the equivalent pure fluid, \( h_{cb\_ideal} \).
6. The mixture effects on nucleate boiling are expressed as

\[
h_{nb} = h_{pb} \times S_{ideal}
\]  
(2.44)

\[
h_{pb} = \frac{h_{pb\_ideal}}{[1 + A_{nb} \times (Y_1 - X_1)]}
\]  
(2.45)

\[
h_{pb\_ideal} = X_1 \times h_{pb\_1} + (1 - X_1) \times h_{pb\_2}
\]  
(2.46)

\( h_{pb\_1} \), \( h_{pb\_2} \), and \( h_{pb} \) are the pool boiling heat transfer coefficients for component 1, component 2, and the binary mixture. The first two components are evaluated from equation 2.15 with the same wall superheat for convective boiling. Factor \( A_{nb} \) is the factor that account for the degradation of heat transfer due to mixture effects and was determined empirically as 15.
Murata and Hushizume (1993) claimed that the nucleate boiling component must be evaluated using the wall superheat instead of the nucleate boiling heat flux, \( q_{nb} \) or the total heat flux, \( q \).

The heat flux into the vapor core at the liquid-vapor interface is written as

\[
q_v = h_v (T_i - T_e) \tag{2.47}
\]

The liquid-vapor interface temperature is slightly higher than the saturation temperature at the equilibrium condition. By neglecting the diffusive resistance of the two-phase convection, the total heat flux at the tube wall can be written as

\[
q = h_{if} (T_w - T_e) + h_{nb} (T_w - T_{sat}) \tag{2.48}
\]

With assumption 2, \( T_e = T_{sat} \), and by definition of the two-phase heat transfer coefficient in terms of the heat flux and the wall superheat, the two-phase heat transfer coefficient can be rearranged as

\[
h_{ip} = \frac{h_{if} + h_{nb}}{1 + \left( \frac{q_v}{q_v} \times \frac{h_{if}}{q_{eb}} \right)} \tag{2.49}
\]

From assumptions 2 and 4

\[
\frac{q_v}{q} = x \times C_{p,x} \left( \frac{\partial T_{sat}}{\partial i} \right) \tag{2.50}
\]

and with assumption 5

\[
h_{if} = h_{eb\_ideal} \tag{2.51}
\]
Hence, the two-phase heat transfer coefficient is simplified to be

\[ h_{tp} = \frac{h_{ch,\text{ideal}} + h_{nb}}{1 + \left( x \times c_{p,\text{avg}} \times \frac{\partial T_{\text{sat}}}{\partial \dot{i}} \right)_p \times h_{ch,\text{ideal}} / h_i} \]  

(2.52)

The two-phase heat transfer coefficient (equation 2.51) coincides with the Murata and Hashizume (1993) pure refrigerant correlation (equations 15 to 19) when \( X_f, Y_i \), and \( (\partial T_{\text{sat}}/\partial i)_p \) are equal to zero. Vapor phase heat transfer coefficient, \( h_i \), is determined from the analogy between heat transfer and shear stress at the liquid-vapor interface as discussed by Murata and Hushizume (1993).

Murata and Hashizume (1993) correlation was tested against experimental data for R123 and R134a mixtures. The correlation successful predicted the experimental heat transfer coefficients within \( \pm 20\% \). It achieved a mean deviation around 5\% for mass fluxes of 200 and 300 kg/m\(^2\)-s. This correlation tends to under-predict the heat transfer coefficient in the low mass flux region for smooth tubes.

**Correlations for Refrigerant Mixtures Flowing inside Micro-fin Tubes**

As a continuation of the Murata and Hashizume (1993) smooth tube correlation for refrigerant mixtures, Murata and Hashizume (1993) identified further degradation in heat transfer due to the presence of micro-fins. The proposed correlation for binary mixtures flowing inside micro-fin tubes is relatively similar to the correlation for binary mixtures flowing inside smooth tubes. With minor modifications as listed in equations 2.20 to 2.22, to account for micro-fin tube effects, the two-phase heat transfer coefficient for refrigerant mixtures flowing inside micro-fin tubes can be determined. Compared with the result achieved in smooth tubes, this correlation for micro-fin tubes under-predicts the experimental two-phase heat transfer coefficient at lower mass flux region. The mean deviation is around 8\%.

Cavallini et al. (1999) recommended a two-phase heat transfer model for zeotropic refrigerant mixtures flowing inside micro-fin tubes. Their work is based on their correlation for pure refrigerants
inside micro-fin tubes discussed in the previous section. Their model is limited to a maximum total isobaric temperature glide around 7-8°C.

Cavallini et al. (1999) started their model with reference to the cylindrical envelope surface area of the finned surface at the fin tip and to the difference between wall temperature $T_w$ and the local bubble-point temperature $T_b$. They observed a degradation in heat transfer in a zeotropic mixture with respect to the ideal mixing of the pure components. The degradation is caused by the mass diffusion of the more volatile component to the vapor liquid interface, and the mass diffusion further increases the bubble point temperature and reduces the temperature driving force. Additional heat is required to heat the liquid and the vapor to the boiling temperature, which constantly increases along the tube. The sensible heating of the vapor phase is associated to single phase (vapor phase) convective heat transfer coefficient. As recommended by Stephan (1992), the two-phase heat transfer coefficient for refrigerant mixtures is

$$h_{fp} = \left[ \frac{1}{h_{T_f}} + \frac{\delta Q_{Sh}}{\delta Q_T} \frac{1}{h_v} \right]^{-1} \tag{2.53}$$

$\delta Q_{Sh}/\delta Q_T$ is the ratio between the sensible heat flow rate heating the vapor and the total heat flow rate. For a maximum temperature glide around 7-8°C, this ratio is often assumed to be constant. The two-phase heat transfer coefficient then becomes

$$h_{fp} = \left[ \frac{1}{h_{T_f}} + \frac{x \kappa_{\rho \cdot \cdot \cdot} \Delta T G}{i_{SG} \frac{1}{h_v}} \right]^{-1} \tag{2.54}$$

The heat transfer coefficient for the vapor phase only is determined from the Dittus-Boelter (1930) equation. The heat transfer coefficient for liquid film, $h_{lj}$, is determined from the Cavallini et al. (1999) correlation for pure refrigerant with the properties of the refrigerant mixture (liquid and vapor at their equilibrium composition). To account for the mass diffusion effect, the nucleate boiling term is multiplied by a mixture correction factor suggested by Thome (1996) for vaporization in plain tubes.

$$h_{lj} = F_c \times h_{nh} + h_{cv} \tag{2.55}$$
\[ F_c = \left[ 1 + \frac{h_{IP, ideal} \times \Delta TG}{q_{ub}} \right] \times \left[ 1 - \exp \left( -\frac{q_{ub}}{\rho \times i_g \times \lambda} \right) \right]^{-1} \]  

where \( h_{IP, ideal} \) is the ideal heat transfer coefficient calculated with a pure refrigerant flow boiling correlation with mixture properties. \( q_{ub} \) is the local nucleate boiling heat flux; however, for approximation, total heat flux, \( q \), can be assumed. Thome (1996) pointed out that this approximation is justifiable for the small boiling ranges observed in refrigerant mixtures. The mass transfer coefficient, \( \lambda \), is set to be 0.0003 m/s by Thome (1996).

The Cavallini (1999) correlation was tested on 110 experimental data points for refrigerant mixtures and the mean absolute deviation between the predicted heat transfer coefficients and the experimental heat transfer coefficient is determined to be 21%. Large deviations are observed at low mass flow rates for R407c, and it is believed that the liquid and vapor phases are not in equilibrium. At high vapor qualities, typically around 0.7-0.8, which is the region where the heat transfer coefficient decreases with the quality, a large deviation is also observed.
CHAPTER III
HEAT TRANSFER MODEL

Two correlations discussed in the previous chapter for micro-fins tubes are further evaluated. The two correlations are the Kido et al. (1995) model and the Cavallini et al. (1999) model. The Murata and Hashizume (1993) and the Kandlikar (1991) correlations are not considered because these correlations use only empirical constants to account for heat transfer enhancement due to the presence of micro-fins. The Kido et al. (1995) and the Cavallini et al. (1999) models are reproduced, and they are validated using the available experimental data sets.

Experimental Database

A database of the available experimental data for pure refrigerants and refrigerant mixtures flowing inside micro-fins tubes was collected in order to validate the existing evaporation heat transfer models. Tables 3.1 and 3.2 show the collected pure refrigerants experimental data for flow inside micro-fins tubes. Tables 3.3 and 3.4 present the experimental data for refrigerant mixtures (zeotropic and azeotropic mixtures). Table 3.1 and table 3.3 list the flow conditions (saturation pressure $P_{sat}$, saturation temperature $T_{sat}$, heat flux $q$, mass flux $G$, and mean vapor quality $\chi$), and table 3.2 and 3.4 delineate the tube geometries (outer tube diameter $d_o$, minimum wall thickness $t_h$, fin height $e$, number of fin $n_f$, helix angle $\gamma$, apex angle $\beta$ and heated test section length $L$).
Table 3.1: Flow Conditions for Pure Refrigerants Flowing inside Micro-fin Tubes

<table>
<thead>
<tr>
<th>Reference</th>
<th>Runs</th>
<th>Fluid</th>
<th>$P_{\text{sat}}$ (kPa)</th>
<th>$T_{\text{sat}}$ (°C)</th>
<th>q (kW/m²)</th>
<th>G (kg/m²·s)</th>
<th>x (mean)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Bogart and Thors (1994)</td>
<td>87</td>
<td>R22</td>
<td>1.67</td>
<td>10.5 – 35.5</td>
<td>120 – 410</td>
<td>0.10 – 0.80</td>
<td></td>
</tr>
<tr>
<td>Bogart and Thors (1999)</td>
<td>50</td>
<td>R22, R134a</td>
<td>530.0</td>
<td>10.5 – 35.5</td>
<td>25 – 275</td>
<td>0.10 – 0.95</td>
<td></td>
</tr>
<tr>
<td>Eckels et al. (1991)</td>
<td>25</td>
<td>R134a</td>
<td>5, 10, 15</td>
<td>13.6 – 64.3</td>
<td>130 – 400</td>
<td>0.05 – 0.88</td>
<td></td>
</tr>
<tr>
<td>Eckels et al. (1994)</td>
<td>11</td>
<td>R134a</td>
<td>1</td>
<td>18.5 – 59.3</td>
<td>85 – 375</td>
<td>0.05 – 0.88</td>
<td></td>
</tr>
<tr>
<td>Eckels et al. (1998a)</td>
<td>8</td>
<td>R134a</td>
<td>1</td>
<td>18.2 – 54.5</td>
<td>125 – 375</td>
<td>0.05 – 0.88</td>
<td></td>
</tr>
<tr>
<td>Eckels et al. (1998b)</td>
<td>9</td>
<td>R134a</td>
<td>2</td>
<td>12.8 – 42.2</td>
<td>85 – 250</td>
<td>0.05 – 0.88</td>
<td></td>
</tr>
<tr>
<td>Hitachi Cable (1987)</td>
<td>22</td>
<td>R22</td>
<td>0.8</td>
<td>10.0</td>
<td>100 – 300</td>
<td>0.60</td>
<td></td>
</tr>
<tr>
<td>Kido et al. (1995)</td>
<td>90/</td>
<td>R22</td>
<td>490.0</td>
<td>9.3</td>
<td>86 – 345</td>
<td>0.10 – 0.90</td>
<td></td>
</tr>
<tr>
<td>Kuo and Wang (1996a)</td>
<td>24</td>
<td>R22</td>
<td>6</td>
<td>6.0 – 14.0</td>
<td>100 – 300</td>
<td>0.10 – 0.80</td>
<td></td>
</tr>
<tr>
<td>Kuo and Wang (1996b)</td>
<td>5</td>
<td>R22</td>
<td>10</td>
<td>10.0</td>
<td>200</td>
<td>0.10 – 0.80</td>
<td></td>
</tr>
<tr>
<td>Murata and Hashizume (1993)</td>
<td>31</td>
<td>R123</td>
<td>202.7</td>
<td>0 – 30.0</td>
<td>93 – 278</td>
<td>0.10 – 1.00</td>
<td></td>
</tr>
<tr>
<td>Muzzio et al. (1998)</td>
<td>26</td>
<td>R22</td>
<td>5</td>
<td>5.4 – 24.1</td>
<td>90 – 400</td>
<td>0.35 – 0.75</td>
<td></td>
</tr>
<tr>
<td>Schlager L. M. (1988)</td>
<td>25</td>
<td>R22</td>
<td>0 – 6</td>
<td>15.4 – 51.7</td>
<td>125 – 400</td>
<td>0.10 – 0.90</td>
<td></td>
</tr>
<tr>
<td>Shinohara and Tobe (1985)</td>
<td>9</td>
<td>R22</td>
<td>400.0</td>
<td>10.0</td>
<td>120 – 300</td>
<td>0.60</td>
<td></td>
</tr>
<tr>
<td>Yasuda et al. (1990)</td>
<td>16</td>
<td>R22</td>
<td>0.6</td>
<td>10.0</td>
<td>80 – 300</td>
<td>0.60</td>
<td></td>
</tr>
</tbody>
</table>
Table 3.2: Tube Geometries for Pure Refrigerants Flowing inside Micro-fin Tubes

<table>
<thead>
<tr>
<th>Reference</th>
<th>Tube Material</th>
<th>$d_o$ (mm)</th>
<th>$t_h$ (mm)</th>
<th>$e$ (mm)</th>
<th>$n_f$</th>
<th>$\gamma$ (°)</th>
<th>$\beta$ (°)</th>
<th>$L$ (m)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Bogart and Thors (1994)</td>
<td>Copper</td>
<td>9.53</td>
<td>0.33</td>
<td>0.2</td>
<td>60</td>
<td>18</td>
<td>23</td>
<td>50</td>
</tr>
<tr>
<td></td>
<td></td>
<td>15.88</td>
<td>0.51</td>
<td>0.3</td>
<td>75</td>
<td>23</td>
<td>50</td>
<td></td>
</tr>
<tr>
<td>Bogart and Thors (1999)</td>
<td>Copper</td>
<td>15.88</td>
<td>0.51</td>
<td>0.3</td>
<td>75</td>
<td>23</td>
<td>50</td>
<td></td>
</tr>
<tr>
<td>Eckels et al. (1991)</td>
<td>Copper</td>
<td>9.52</td>
<td>0.40</td>
<td>0.2</td>
<td>60</td>
<td>17</td>
<td>50</td>
<td></td>
</tr>
<tr>
<td>Eckels et al. (1994)</td>
<td>Copper</td>
<td>9.52</td>
<td>0.30</td>
<td>0.2</td>
<td>60</td>
<td>17</td>
<td>50</td>
<td></td>
</tr>
<tr>
<td>Eckels et al. (1998a)</td>
<td>Copper</td>
<td>9.52</td>
<td>0.30</td>
<td>0.2</td>
<td>60</td>
<td>18</td>
<td>50</td>
<td></td>
</tr>
<tr>
<td>Eckels et al. (1998b)</td>
<td>Copper</td>
<td>12.70</td>
<td>0.40</td>
<td>0.2</td>
<td>60</td>
<td>17</td>
<td>50</td>
<td></td>
</tr>
<tr>
<td>Hitachi Cable (1987)</td>
<td>Copper</td>
<td>9.50 - 9.52</td>
<td>0.28 - 0.29</td>
<td>0.2 - 0.21</td>
<td>60</td>
<td>17 - 18</td>
<td>40 - 53</td>
<td></td>
</tr>
<tr>
<td>Kido et al. (1995)</td>
<td>Copper</td>
<td>7.00</td>
<td>0.30 - 0.35</td>
<td>0.15 - 0.21</td>
<td>60 - 100</td>
<td>3 - 18</td>
<td>20 - 53</td>
<td></td>
</tr>
<tr>
<td>Kuo and Wang (1996a)</td>
<td>Copper</td>
<td>9.52</td>
<td>0.30</td>
<td>0.2</td>
<td>60</td>
<td>18</td>
<td>50</td>
<td></td>
</tr>
<tr>
<td>Kuo and Wang (1996b)</td>
<td>Copper</td>
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<td>0.30</td>
<td>0.2</td>
<td>60</td>
<td>18</td>
<td>50</td>
<td></td>
</tr>
<tr>
<td>Murata and Hashizume (1993)</td>
<td>Copper</td>
<td>12.70</td>
<td>1.00</td>
<td>0.3</td>
<td>60</td>
<td>30</td>
<td>50</td>
<td></td>
</tr>
<tr>
<td>Muzzio et al. (1998)</td>
<td>Copper</td>
<td>9.52</td>
<td>0.30 - 0.34</td>
<td>0.20 - 0.15</td>
<td>60</td>
<td>18 - 25</td>
<td>53 - 90</td>
<td></td>
</tr>
<tr>
<td>Schlager L. M. (1988)</td>
<td>Copper</td>
<td>9.52</td>
<td>0.40 - 0.50</td>
<td>0.2 - 0.38</td>
<td>60 - 21</td>
<td>18 - 30</td>
<td>50 - 10</td>
<td></td>
</tr>
<tr>
<td>Shinohara and Tobe (1985)</td>
<td>Copper</td>
<td>9.52</td>
<td>0.30</td>
<td>0.12 - 0.20</td>
<td>60</td>
<td>50 - 65</td>
<td>7 - 25</td>
<td></td>
</tr>
<tr>
<td>Yasuda et al. (1995)</td>
<td>Copper</td>
<td>9.52</td>
<td>0.30</td>
<td>0.2 - 0.25</td>
<td>60</td>
<td>18</td>
<td>30</td>
<td>40</td>
</tr>
<tr>
<td></td>
<td></td>
<td>7.94</td>
<td></td>
<td></td>
<td>50</td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

Notes: $d_o$: outer diameter, $t_h$: thickness of tube, $e$: fin spacing, $n_f$: number of fins, $\gamma$: fin angle, $\beta$: tube angle, $L$: length.
Table 3.3: Flow Conditions for Refrigerant Mixtures Flowing inside Micro-fin Tubes

<table>
<thead>
<tr>
<th>Reference</th>
<th>Runs</th>
<th>Fluid</th>
<th>$P_{sat}$ (kPa)</th>
<th>$q$ (kW/m²)</th>
<th>$G$ (kg/m²⋅s)</th>
<th>$\alpha$ (mean)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Bogart and Thors (1999)</td>
<td>12</td>
<td>R407c, R507a</td>
<td>530.00</td>
<td>10.5 – 35.5</td>
<td>25 – 275</td>
<td>0.10 – 0.90</td>
</tr>
<tr>
<td></td>
<td>21</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Cui et al. (1986)</td>
<td>45</td>
<td>R502</td>
<td>320.00</td>
<td>11.0 – 41.2</td>
<td>126 – 396</td>
<td>0.24 – 1.00</td>
</tr>
<tr>
<td>Ebisu et al. (1998)</td>
<td>8</td>
<td>R407c</td>
<td>545.57</td>
<td>7.5</td>
<td>150 – 400</td>
<td>0.20 – 0.80</td>
</tr>
<tr>
<td>Kuo and Wang (1996a)</td>
<td>32</td>
<td>R407c</td>
<td>600.00</td>
<td>6.0 – 14.0</td>
<td>100 – 300</td>
<td>0.10 – 0.80</td>
</tr>
<tr>
<td>Murata and Hashizume (1993)</td>
<td>37</td>
<td>R123+ R134a</td>
<td>202.70 243.20</td>
<td>0 – 30.0</td>
<td>93, 185 278</td>
<td>0.10 – 1.00</td>
</tr>
</tbody>
</table>

Table 3.4: Tube Geometries for Refrigerant Mixtures Flowing inside Micro-fin Tubes

<table>
<thead>
<tr>
<th>Reference</th>
<th>Tube Material</th>
<th>$d_o$ (mm)</th>
<th>th (mm)</th>
<th>e (mm)</th>
<th>$n_f$</th>
<th>$\gamma$ (°)</th>
<th>$\beta$ (°)</th>
<th>L (m)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Bogart and Thors (1999)</td>
<td>Copper</td>
<td>15.88</td>
<td>0.50</td>
<td>0.3</td>
<td>75</td>
<td>23</td>
<td>50</td>
<td>4.88</td>
</tr>
<tr>
<td>Cui et al. (1986)</td>
<td>Copper</td>
<td>16.00</td>
<td>1.50</td>
<td>0.2 – 0.9</td>
<td>20 – 60</td>
<td>8 – 30</td>
<td>75 – 90</td>
<td>0.40</td>
</tr>
<tr>
<td>Ebisu et al. (1998)</td>
<td>Copper</td>
<td>7.00</td>
<td>0.25</td>
<td>0.18</td>
<td>50</td>
<td>18</td>
<td>40</td>
<td>0.54</td>
</tr>
<tr>
<td>Kuo and Wang (1996a)</td>
<td>Copper</td>
<td>9.52</td>
<td>0.30</td>
<td>0.2</td>
<td>60</td>
<td>18</td>
<td>50</td>
<td>1.30</td>
</tr>
<tr>
<td>Murata and Hashizume (1993)</td>
<td>Copper</td>
<td>12.70</td>
<td>1.00</td>
<td>0.3</td>
<td>60</td>
<td>30</td>
<td>60</td>
<td>0.73</td>
</tr>
</tbody>
</table>

Kido et al. (1995) used condensing R114 to heat the test section, and Murata and Hashizume (1993) used a sheathed heater around the micro-fin tube to provide constant heat flux. Other experimental data were generated with the test sections heated by water in the counter-flow condition. Most of the data sets were obtained using modified Wilson Plot techniques, except the data sets from Cui et al. (1986), Ebisu and Torikoshi (1998), Kido et al. (1995), Murata and Hashizume (1993), and Muzzio et al. (1998) which were obtained from energy balances. Kido et al. (1995), Kuo and Wang (1996), and Murata and Hashizume (1993) presented their heat transfer coefficient data at constant mass flux with varying vapor quality. Other experimental data were presented at constant vapor quality with varying mass flux. The reported uncertainties for in-tube two-phase heat transfer coefficients varied from 7.0 % to 17.5 %. The
experimental data provide average two-phase heat transfer coefficients, $h_{ip}$, over the entire test section length, $L$.

The experimental data from Schlager (1988) were collected from the tables presented in his thesis. The remaining experimental data sets were collected from the graphs documented in the published technical papers. The extraction of the data points from the graphs was accomplished with the aid of computer software, Digitize XY Data (DigXY) created by Mark James (1998). The required graph containing the experimental data was first scanned in bitmap format (filename.bmp) before DigXY analyzed it. DigXY allows the user to specify and to rotate the mapping region of the graph. The boundaries of the graph (minimum and maximum points at x and y axes) are defined before the data points are collected. The data points were sent to a text file (filename.txt), which comprised two columns, where the first column is the x-axis data and the second column is the corresponding y-axis data.

The thermodynamic and transport properties for pure refrigerants and refrigerant mixtures are obtained from REFPROP 6.01 computer software. REFPROP 6.01 is capable of generating thermodynamic and transport properties for pure refrigerants and refrigerant mixtures at specified saturation conditions (saturation temperature or saturation pressure).

The mean absolute deviation ($MAD$) is set as the criterion to determine the effectiveness of the heat transfer model. $MAD$ is defined as the average of the normalized difference between the predicted two-phase heat transfer coefficient and the experimental two-phase heat transfer coefficient. The heat transfer model is considered acceptable if the achieved $MAD$ is less than 30%.

$$MAD = \frac{1}{N} \sum \frac{h_{\text{experimental}} - h_{\text{predicted}}}{h_{\text{experimental}}} \quad (3.1)$$

Kido et al. (1995) Heat Transfer Model

Kido et al. (1995) developed a semi-empirical model to calculate the two-phase heat transfer coefficient. The model is expressed in dimensionless form as
\[ \text{Nu}_p = 2.5 \times \frac{Y_{\text{yy}}^{0.2} \times \text{Re}^{0.27} \times \text{Pr}^{0.4} \times (1 + \tan(\gamma))^{1.8}}{\text{Bon}^{0.2}} \]  

(3.2)

Kido et al. (1995) introduced a new dimensionless parameter, \textit{AreaRatio}, which accounts for the enhancement in heat transfer surface area due to the presence of micro-fins. The dimensionless parameter is calculated as

\[ \text{AreaRatio} = \frac{A_{\text{eff}}}{A_{\text{sm}}} = \frac{\sqrt{(p - t - b)^2 + 4x^2} + t + b}{p \times \cos(\gamma)} \]  

(3.3)

Equation 3.2 is multiplied by the \textit{AreaRatio} to obtain the Nusselt number that accounts for the heat transfer enhancement due to the presence of micro-fins.

The Kido et al. (1995) model is verified using four pure refrigerant experimental data sets, which include refrigerants R22, R123, and R134a. These data sets are from Kido et al. (1995), Kuo and Wang (1996a), Murata and Hashizume (1993), and Bogart and Thors (1999).

MathCAD 2000 computer software is used to perform all the calculations. The experimental data and the Kido et al. (1995) model are entered in MathCAD environment in two separate files, namely a data file and a model file. An additional MathCAD property file is programmed with an interpolation function (cubic spline function) such that it will automatically obtain the required refrigerant properties from the property tables (text files) generated from REFPROP 6.01. The accuracy of the interpolated property values is four decimal points. A sample MathCAD data file, a model file, and a property file are presented in Appendix A. Another calculation file is created in MathCAD to call all the required information and generate the necessary results. A sample MathCAD calculation file for the Kido et al. (1995) model is presented in Appendix A.

The Kido et al. (1995) model is first validated on their R22 experimental data set. The validation results are used to ensure the Kido et al. (1995) model is regenerated accurately.

Figure 3.1 shows the prediction results using the Kido et al. (1995) model on their R22 experimental data set. The circular symbols indicate the experimental data points. The diagonal solid line shows the perfect match of the predicted two-phase heat transfer coefficient and the experimental two-
phase heat transfer coefficient. The dashed lines present the ±30% mean absolute deviation lines. There are some minor over-prediction results in figure 3.1. The model achieves a mean absolute deviation around 30%. The Kido et al. (1995) experimental data include a few data points at high vapor quality region, which are identified to be in the dry-out region. Once these data points are excluded from the data set, the model provides a relatively low mean absolute deviation of 11.8%. The prediction results agree with the results presented by Kido et al. (1995) and the prediction results are shown in figure 3.2.

Figure 3.1: Kido et al. (1995) Model on the Kido et al. (1995) Data Set (with dry-out data points)

Figure 3.2: Kido et al. (1995) Model on the Kido et al. (1995) Data Set (without dry-out data points)
The Kido et al. (1995) model is also validated using the R22 experimental data set from Kuo and Wang (1996a). Figure 3.3 illustrates the prediction results with a 45.9% mean absolute deviation. Generally, the model under-predicts the experimental data.

![Graph showing comparison between predicted and experimental heat transfer coefficients]

Figure 3.3: Kido et al. (1995) Model on the Kuo and Wang (1996a) Data Set

Further analysis is done on this data set. In figures 3.4 and 3.5, the lines represent the prediction results and symbols show the experimental data. Figure 3.4 illustrates the two-phase heat transfer coefficient versus the vapor quality at constant mass flux $G$ (200 kg/m$^2\cdot$s) but increasing heat flux $q$ (6, 10, and 14 kW/m$^2$). The arrows in figure 3.4 symbolize the increasing heat flux. Figure 3.5 presents the two-phase heat transfer coefficient versus vapor quality at constant heat flux $q$ (10 kW/m$^2$) but increasing mass flux $G$ (100, 200, and 300 kg/m$^2\cdot$s). The arrows in figure 3.5 symbolize the increasing mass flux. The model predicts that the two-phase heat transfer coefficient increases with increasing heat flux and increasing mass flux. However, the model under-predicts the experimental results by 50%. The experimental results tend to have a convex behavior as the quality increases, but the model exhibits a concave behavior as the quality increases.
Figure 3.4: Kido et al. (1995) Model on the Kuo and Wang (1996a) Data Set (constant $G$, increasing $q$)

Figure 3.5: Kido et al. (1995) Model on the Kuo and Wang (1996a) Data Set (constant $q$, increasing $G$)
Figure 3.6 presents a comparison between the Murata and Hashizume (1993) R123 data set and the Kido et al. (1995) model. The model achieves a mean absolute deviation of 26.1%. The model provides relatively close prediction to the data points in the low quality region, $0.1 < x < 0.5$. As the vapor quality increases, larger deviation is observed.

Figure 3.6: Kido et al. (1995) Model on the Murata and Hashizume (1993) Data Set

The R134a experimental data from Bogart and Thors (1999) were also tested using the Kido et al. (1995) model. Figure 3.7 shows the two-phase heat transfer coefficient, $h_{tp}$, versus mass flux, $G$. The solid line represents the model, and the cross symbols represent the data set. The mean absolute deviation achieved is around 38.5%. The model fails to predict the behavior of the heat transfer coefficient with increasing mass flux. It over-predicts the experimental data at low mass flux ($G<100 \text{ kg/m}^2\cdot\text{s}$), but it under-predicts the data in the medium and high mass flux regions.
Figure 3.7: Kido et al. (1995) Model on the Bogart and Thors (1999) Data Set

Table 3.5 shows the mean absolute deviation using the Kido et al. (1995) model on four different data sets previously discussed. Among those data sets, the model predicts only the Kido et al. (1995) and Murata and Hashizume (1995) data sets within a 30% mean absolute deviation. In addition, the Kido et al. (1995) model has several restrictions on the \( Y \) factor, \( Re, Bon^* \), and \( \gamma \) The maximum allowable \( Re \) and \( \gamma \) are 10,000 and 20\(^\circ\), respectively. These restrictions limit the usage of this model. Generally, this model is relatively poor and does not provide good prediction results for most of the refrigerants. The model should provide good predictions for R22 data sets since the empirical constants were generated from R22 data set; however, the model fails to predict the R22 data set of Kuo and Wang (1996a).

Table 3.5: Mean Absolute Deviation Between the Experimental Data and the Predicted Data from the Kido et al. (1995) Model.

<table>
<thead>
<tr>
<th>Reference</th>
<th>Refrigerant</th>
<th>MAD</th>
</tr>
</thead>
<tbody>
<tr>
<td>Kido et al. (1995)</td>
<td>R22</td>
<td>11.8%</td>
</tr>
<tr>
<td>Kuo and Wang (1996a)</td>
<td>R22</td>
<td>45.9%</td>
</tr>
<tr>
<td>Murata and Hashizume (1993)</td>
<td>R123</td>
<td>26.1%</td>
</tr>
<tr>
<td>Bogart and Thors (1999)</td>
<td>R134a</td>
<td>38.5%</td>
</tr>
</tbody>
</table>
Cavallini et al. (1999) Heat Transfer Model for Pure Refrigerants

The Cavallini et al. (1999) model is a more comprehensive evaporative two-phase heat transfer model that accounts for heat transfer enhancement in micro-fin tubes for pure refrigerant and refrigerant mixtures (zeotropic mixtures). The two-phase heat transfer coefficient for pure refrigerant is written as

\[ h_{op} = h_{opb} \times \left( A \times \frac{X}{T} \right) \times F_1^C + \frac{h_1 \times \phi \times R \times X \times S^* \times (Bon^W \times F_{T_1}) \times F_2^V \times F_3^Z}{\} \]  

(3.4)

The empirical constants \( A, B, C, S^*, T, V, \) and \( Z \) were generated from 361 data points where 94% of the data points are from R22 experimental data. The remaining 6% are from the R12 experimental data of Kimura and Ito (1985). The empirical constants for the Cavallini model are listed in table 3.6.

<table>
<thead>
<tr>
<th>Condition ( G )</th>
<th>A ( \text{kg/m}^2 \cdot \text{s} )</th>
<th>B</th>
<th>C</th>
<th>S*</th>
<th>T</th>
<th>V</th>
<th>Z</th>
</tr>
</thead>
<tbody>
<tr>
<td>( G &lt; 500 )</td>
<td>1.36</td>
<td>0.36</td>
<td>0.38</td>
<td>2.14</td>
<td>-0.15</td>
<td>0.59</td>
<td>0.36</td>
</tr>
<tr>
<td>( G &gt; 500 )</td>
<td>1.36</td>
<td>0.36</td>
<td>0.38</td>
<td>2.14</td>
<td>-0.21</td>
<td>0.59</td>
<td>0.36</td>
</tr>
</tbody>
</table>

The Cavallini et al. (1999) model was validated on 16 pure refrigerant data sets from 9 different researchers with a total of 371 data points. The refrigerants tested include R12, R22, R123, and R134a. A majority of these experiments used R22 as the working fluid. Table 3.7 shows the range of the experimental conditions for all the data sets used to validate the Cavallini et al. (1999) model.

<table>
<thead>
<tr>
<th>Experimental Condition</th>
<th>Range</th>
<th>Experimental Condition</th>
<th>Range</th>
</tr>
</thead>
<tbody>
<tr>
<td>Heat flux (kW/m²)</td>
<td>0 – 65</td>
<td>Fin height (mm)</td>
<td>0.12 – 0.30</td>
</tr>
<tr>
<td>Mass flux (kg/m² · s)</td>
<td>25 – 410</td>
<td>Number of fin</td>
<td>21 – 75</td>
</tr>
<tr>
<td>Mean vapor quality</td>
<td>0.05 – 0.90</td>
<td>Helix angle (°)</td>
<td>7 – 30</td>
</tr>
<tr>
<td>Outer tube diameter (mm)</td>
<td>8.00 – 16.00</td>
<td>Apex angle (°)</td>
<td>10 – 90</td>
</tr>
<tr>
<td>Max. tube thickness (mm)</td>
<td>0.3 – 1.0</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>
The validation process was similar to that in the previous section; MathCAD 2000 computer software was used to perform the necessary calculations to implement the Cavallini et al. (1999) model. The sample property file and data file are similar to the file format presented in Appendix A. Sample model and calculation files are shown in Appendix B.

Figure 3.8 presents the Cavallini et al. (1999) model using the Eckels et al. (1991) R12 experimental data. As shown in figure 3.8, the model generally over-predicts R12 experimental data and achieves a mean absolute deviation of 31.3%. The model exhibits a consistent trend of the two-phase heat transfer coefficient, $h_{tp}$, variation as the mass flux, $G$, increases. The result in figure 3.8 contradicts the result reported by Cavallini et al. (1999). Cavallini et al. (1999) claimed that the model achieved a smaller mean absolute deviation of 14.8%.

![Figure 3.8: Cavallini et al. (1999) Model on the Eckels et al. (1991) R12 Data Set](image)

The Cavallini et al. (1999) model is also tested on the R123 experimental data of Murata and Hashizume (1993). Figure 3.9 shows the prediction results. The observed mean absolute deviation is about 13.1%. This result agrees with those presented by Cavallini et al. (1999). The model predicts well the experimental data except for a few data points that have a vapor quality close to 1. Murata and Hashizume...
(1993) generated the two-phase heat transfer coefficient data from mean values at two axial locations, inlet and outlet of the test section. At the conditions where \( x \sim 1 \), the authors selected the two-phase heat transfer coefficient obtained from the inlet of the test section instead of the mean value of the two axial locations. These data points are neglected because of the high uncertainty associated with them.

More recent and updated R134a data from Bogart and Thors (1999) are used to validate the Cavallini et al. (1999) model. Figure 3.10 represents the prediction results of the two-phase heat transfer coefficient, \( h_{tp} \), versus mass flux, \( G \). Figure 3.10 clearly demonstrates that the model provides good prediction at mass fluxes of 150 to 300 kg/m²-s. The model greatly over-predicts the experimental data in the low mass flux region. The mean absolute deviation achieved is 38.6%.
Other R134a experimental data sets are available from Eckels et al. (1991, 1994, 1998a, 1998b). For these particular data sets, the Cavallini et al. (1999) model is capable of predicting the trend of the two-phase heat transfer coefficient, $h_{tp}$, with increasing mass flux, $G$. As shown in figures 3.11 and 3.12, the model over-predicts all these data points. The mean absolute deviation observed is 33.9%. These results show a larger deviation than the result claimed by Cavallini et al. (1999).
The Cavallini et al. (1999) model was also validated by a total of nine experimental data sets for R22 from seven researchers.
Figure 3.13 presents the predicted two-phase heat transfer coefficient, $h_{tp}$, versus mass flux, $G$, for the Bogart and Thors (1999) R22 data set. The model over-predicts the data for mass fluxes from 30 to 90 kg/m²-s. In the medium and high mass flux regions, the model provides relatively good predictions. The mean absolute deviation is around 26.2%.

![Figure 3.13: Cavallini et al. (1999) Model on the Bogart and Thors (1999) R22 Data Set](image)

The Cavallini et al. (1999) model is also tested on experimental data obtained from Hitachi Cable review (1987). Figure 3.14 shows that the model is capable of predicting the experimental data within an acceptable range; however, high deviation is observed at the medium quality region, $0.4 < x < 0.8$. The mean absolute deviation is 23.3%.
Table 3.8 summarizes all the prediction results and the mean absolute deviations. The R22 prediction results for Bogart and Thors (1994), Muzzio et al. (1998), Schlager (1988), Shinohara and Tobe (1985) and Yasuda et al. (1990) data sets closely agree with the results claimed by Cavallini et al. (1999).

Table 3.8: Mean Absolute Deviation (MAD) Between the Experimental Data and the Prediction Data from the Cavallini et al. (1999) Pure Refrigerant Model

<table>
<thead>
<tr>
<th>Reference</th>
<th>Refrigerant</th>
<th>MAD</th>
</tr>
</thead>
<tbody>
<tr>
<td>Bogart and Thors (1994)</td>
<td>R22</td>
<td>11.4%</td>
</tr>
<tr>
<td>Bogart and Thors (1999)</td>
<td>R134a</td>
<td>38.6%</td>
</tr>
<tr>
<td></td>
<td>R22</td>
<td>26.2%</td>
</tr>
<tr>
<td>Eckels et al. (1991)</td>
<td>R12</td>
<td>31.3%</td>
</tr>
<tr>
<td></td>
<td>R134a</td>
<td>45.5%</td>
</tr>
<tr>
<td>Eckels et al. (1994)</td>
<td>R134a</td>
<td>29.2%</td>
</tr>
<tr>
<td>Eckels et al. (1998a)</td>
<td>R134a</td>
<td>23.8%</td>
</tr>
<tr>
<td>Eckels et al. (1998b)</td>
<td>R134a</td>
<td>16.1%</td>
</tr>
<tr>
<td>Hitachi Cable (1987)</td>
<td>R22</td>
<td>23.3%</td>
</tr>
<tr>
<td>Kuo and Wang (1996a)</td>
<td>R22</td>
<td>11.6%</td>
</tr>
<tr>
<td>Kuo and Wang (1996b)</td>
<td>R22</td>
<td>18.1%</td>
</tr>
<tr>
<td>Murata and Hashizume (1993)</td>
<td>R123</td>
<td>13.1%</td>
</tr>
<tr>
<td>Muzzio et al. (1998)</td>
<td>R22</td>
<td>3.0%</td>
</tr>
<tr>
<td>Schlager L. M. (1988)</td>
<td>R22</td>
<td>5.8%</td>
</tr>
<tr>
<td>Shinohara and Tobe (1985)</td>
<td>R22</td>
<td>7.7%</td>
</tr>
<tr>
<td>Yasuda et al. (1990)</td>
<td>R22</td>
<td>11.6%</td>
</tr>
</tbody>
</table>
Overall, the Cavallini et al. (1999) heat transfer model for pure refrigerants is considered relatively successful in predicting R22 experimental data sets. However, as tabulated in table 3.7, the model fails to predict R134a and R12 experimental data. The Cavallini et al. (1999) model is a better predictive model than the Kido et al. (1995) model. The Cavallini et al. (1999) model is also applicable to a wider range of flow conditions and flow parameters than the Kido et al. (1995) model.

Cavallini et al. (1999) Heat Transfer Model for Zeotropic Mixtures

The Cavallini et al. (1999) pure refrigerant model is extended to predict heat transfer performance for zeotropic refrigerant mixtures. The heat transfer model for zeotropic mixtures applies only to mixtures with a maximum temperature glide around 7-8°C. The final form of the model is

\[
h_{p\_w} = \left[ \frac{1}{(F_c \cdot h_{nb} + h_{\nu})^2} \left( \frac{x \cdot c_{p\_w} \cdot \Delta TG}{h_{w}} \right) \right]^{-1}
\]

This model is validated for zeotropic mixtures R407c and R123/R134a data sets. The validity of this model is also tested on azeotropic mixtures R502 and R507a. A total of 155 data points for refrigerant mixtures are available and 89 of them are zeotropic mixture data points.

Detailed calculations are performed in the MathCAD 2000 environment, and the sample calculations and sample model files are presented in Appendix C. In the calculation process, the mixture model uses the same property file and the data file as generated for the pure refrigerant model calculation process.

Figure 3.15 shows the prediction results using the Murata and Hashizume (1993) R123/R134a (90/10 mole%) experimental data. The model provides a relatively reliable prediction with a 23.5% mean absolute deviation. The results agree with those of Cavallini et al. (1999); the mixture model yields a higher deviation in the high vapor quality region, 0.60 < x < 0.95. The model under-estimates the experimental data in this region.
Three R407c experimental data sets, 52 data points, are available from three different sources to validate the Cavallini et al. (1999) mixture model. These data sets are from Bogart and Thors (1999), Ebisu and Torikoshi (1998), and Kuo and Wang (1996).

The Bogart and Thors (1999) experimental data provide the two-phase heat transfer coefficient, $h_{tp}$, variation with increasing mass flux. Figure 3.16 illustrates the experimental data and model predictions. The model fails to predict the experimental data at the low mass flux region, where $G$ is between 30 and 100 kg/m$^2$-s. Cavallini et al. (1999) also observed a high deviation in their model while predicting R407c data sets in the low mass flux region, namely $G \sim 100$ kg/m$^2$-s. The Cavallini et al. (1999) model achieved a large mean absolute deviation at 63.1%.
The Ebisu and Torikoshi (1998) R407c data set is presented in figure 3.17, and the model over-predicts the data set with a mean absolute deviation of 68.8%. However, the model predicts the correct trend of the two-phase heat transfer coefficient, $h_{tp}$, variation with increasing mean vapor quality, $x$. The experimental condition was maintained at a constant mass flux $G = 300 \text{ kg/m}^2\text{-s}$ and a constant heat flux $q = 7.5 \text{ kW/m}^2$. The result presented in figure 3.18 contradicts the claim by Cavallini et al. (1999), who observed a maximum deviation of 40%.
Figures 3.18 and 3.19 represent the R407c data sets from Kuo and Wang (1996a) compared with the Cavallini et al. (1999) mixture model. Figure 3.18 illustrates the variation of two-phase heat transfer coefficient, $h_{tp}$, with respect to vapor quality, $x$, at constant mass flux, $G$, with increasing heat flux, $q$. Figure 3.19 shows the same data but is at constant heat flux, $q$, with increasing mass flux, $G$. The model predicts the experimental data with low deviation in the high mass flux and the high heat flux regions. At low and medium mass fluxes and heat fluxes regions, the deviations are significant. The observed mean absolute deviation is 57.5%.
Figure 3.18: Cavallini et al. (1999) Model on the Kuo and Wang (1996a) R407c Data Set (Constant $G = 200 \text{ kg/m}^2\cdot\text{s}$, $q = 6, 10, \text{ and } 14 \text{ kW/m}^2$)

Figure 3.19: Cavallini et al. (1999) Model on the Kuo and Wang (1996a) R407c Data Set (Constant $q = 10 \text{ kW/m}^2$, $G = 100, 200, \text{ and } 300 \text{ kg/m}^2\cdot\text{s}$)
The Cavallini et al. (1999) mixture model was developed for zeotropic refrigerant mixtures. The model is supposed to be applicable to azeotropic refrigerant mixtures since both zeotropic and azeotropic mixtures behave relatively alike except that the azeotropic mixtures behave essentially as a pure fluid at a particular mixture composition, the azeotrope.

A total of 66 data points for azeotropic mixtures, R502 and R507a, are used to validate the Cavallini et al. (1999) mixture model. Bogart and Thors (1999) conducted an experiment with R507a while Cui et al. (1986) used R502 as the working fluid.

Figure 3.20 shows the prediction results using Cui et al. (1986) R502 data set. The model generally under-predicts the R502 data set. Larger deviation are observed in the high mass flux region ($250 < G < 350$ kg/m$^2$-s) and small helix angle ($\gamma = 8^\circ$) tubes. The model achieves a mean absolute deviation at 46.6%.

![Figure 3.20: Cavallini et al. (1999) Model on the Cui et al. (1986) R502 Data Set](image)

The Bogart and Thors (1999) R507a data set is presented in figure 3.21 with the Cavallini et al. (1999) model prediction. The model over-predicts all the data with higher deviations observed at the low mass flux region, $30 < G < 100$ kg/m$^2$-s.
Figure 3.21: Cavallini et al. (1999) Model on the Bogart and Thors (1999) R507a Data Set

Table 3.8 summarizes the prediction results using the Cavallini et al. (1999) mixture model on all the available refrigerant mixture data sets. Even though the model was originally developed for zeotropic mixtures, the results demonstrate that the model generates slightly better predictions for azeotropic mixtures than for zeotropic mixtures. The mean absolute deviations shown in table 3.8 are still relatively large and these values are larger than the acceptable mean absolute deviation, which is 30%.

Table 3.9: Mean Absolute Deviation (MAD) Between the Experimental Data and the Prediction Data from the Cavallini et al. (1999) Mixture Model

<table>
<thead>
<tr>
<th>Reference</th>
<th>Refrigerant</th>
<th>MAD</th>
</tr>
</thead>
<tbody>
<tr>
<td>Bogart and Thors (1999)</td>
<td>R407c</td>
<td>63.1%</td>
</tr>
<tr>
<td></td>
<td>R507a</td>
<td>50.1%</td>
</tr>
<tr>
<td>Cui et al. (1986)</td>
<td>R502</td>
<td>46.6%</td>
</tr>
<tr>
<td>Ebisu and Torikoshi (1998)</td>
<td>R407c</td>
<td>68.8%</td>
</tr>
<tr>
<td>Kuo and Wang (1996a)</td>
<td>R407c</td>
<td>57.5%</td>
</tr>
<tr>
<td>Murata and Hashizume (1993)</td>
<td>R123+</td>
<td>23.5%</td>
</tr>
<tr>
<td></td>
<td>R134a</td>
<td></td>
</tr>
</tbody>
</table>
Overall, the Kido et al. (1995) pure refrigerant model fails to generate acceptable prediction results to most of the pure refrigerant data sets except their own R22 data sets. The Cavallini et al. (1999) pure refrigerant model provides excellent prediction results for pure refrigerant data sets, except data sets using R12 and R134a. The Cavallini et al. (1999) mixture model provides relatively high mean absolute deviations on the available zeotropic mixture (R407c) data sets, except the mixture of R123/R134a. The Cavallini et al. (1999) mixture model was originally developed for zeotropic mixtures, however, it applicability was tested on azeotropic mixtures. The mean absolute deviations observed in azeotropic mixtures are slightly lower than that of the zeotropic mixtures. This mixture model is likely applicable to azeotropic mixtures since both azeotropic mixtures (except the azeotrope) and zeotropic mixtures have similar behaviors in two-phase flow processes.

The Cavallini et al. (1999) pure refrigerant and refrigerant mixture models are proven to be relatively reliable; however, minor modifications are required in order to achieve better prediction results. The next two chapters introduce the correlation optimization process and make appropriate recommendations to further refine the Cavallini et al. (1999) pure refrigerant and refrigerant mixture models.
CHAPTER IV
CORRELATION OPTIMIZATION

Due to the complexity of the two-phase boiling process in refrigerants, empirical constants are required in the mathematical models in order to provide relatively reliable and consistent prediction results. Cavallini et al. (1999) used a non-linear best fitting procedure to generate seven empirical constants in their pure refrigerant heat transfer model. Cavallini et al. (1999) successfully generated the empirical constants, and their pure refrigerant model is capable of predicting the majority of R22 experimental data sets with acceptable mean absolute deviation. However, their model failed to generate acceptable predictions for the R12 and the R134a experimental data sets.

This chapter introduces a simple and efficient method to generate empirical constants with the aid of MathCAD 2000 software. The empirical constants can be generated, without lengthy programming algorithms, with one simple mathematical function in the MathCAD environment, the Minimize function.

When using the Minimize function in generating the required empirical constants, the standard error of regression function (SER) introduced by Montgomery and Peck (1992) is selected as the reference minimum function.

\[
SER = \left( \frac{\sum_{i=1}^{N} \left[ h_{ip} - \text{function}(h_{ip}) \right]^2}{N - \text{Coeff}} \right)^{0.5}
\]  

(4.1)

\text{Coeff} in equation 4.1 represents the number of empirical constants used in the function for the two-phase heat transfer coefficient \( h_{ip} \).
With the aid of the MathCAD Minimize function, new empirical constants are generated using equation 4.1. The predictive results using the new empirical constants are then compared with the predictive results presented by Cavallini et al. (1999).

Each independent parameter in the Cavallini et al. (1999) pure refrigerant model is first generated from the 371 data points presented in Table 3.1. These independent parameters are listed in Table 4.1. Table 4.1 also shows the corresponding nomenclature used in the MathCAD 2000 environment. The number of components in each independent parameter must be equal to the total number of data point, 371. Equation 4.2 is the SER function for the two-phase heat transfer coefficient for pure refrigerants applied to equation 4.1. This SER function is used as a reference function to achieve a minimum value with the new empirical constants.

$$\text{SER}(P) = \left( \sum_{i=1}^{N} \left[ h_p - \left( h_{ph} \times P1 \times (X_{t})^2 \times F1^p + h_l \times \Phi \times Rx^p \times (Bon^w + Fr) \times F2^{p6} \times F3^{p7} \right) \right] \right)^{0.5} / (N - 7)$$  \hspace{1cm} (4.2)

Variables $P1$, $P2$, $P3$, $P4$, $P5$, $P6$, and $P7$ are the new empirical constants. Since there are seven coefficients required to be determined in equation 4.2, 7 is subtracted from total number of data points $N$.

<table>
<thead>
<tr>
<th>Independent Parameter</th>
<th>Nomenclature /Parameter in MathCAD 2000</th>
</tr>
</thead>
<tbody>
<tr>
<td>$h_{ip}$</td>
<td>$Htp$</td>
</tr>
<tr>
<td>$h_{ph}$</td>
<td>$Hph$</td>
</tr>
<tr>
<td>$h_l$</td>
<td>$Hl$</td>
</tr>
<tr>
<td>$X_{tt}$</td>
<td>$Xtt$</td>
</tr>
<tr>
<td>$Rx$</td>
<td>$Rx$</td>
</tr>
<tr>
<td>$Bon^w + Fr$</td>
<td>$BonWFr$</td>
</tr>
<tr>
<td>$\Phi$</td>
<td>$Phi$</td>
</tr>
<tr>
<td>$F1$</td>
<td>$F1$</td>
</tr>
<tr>
<td>$F2$</td>
<td>$F2$</td>
</tr>
<tr>
<td>$F3$</td>
<td>$F3$</td>
</tr>
</tbody>
</table>

Table 4.1: Independent Parameters in the Cavallini et al. (1999) Pure Refrigerant Model

With the initial guessed values for all the unknown empirical constants ($P1$, $P2$, $P3$, $P4$, $P5$, $P6$, and $P7$), the Minimize function in MathCAD 2000 determines these empirical constants such that the
"SER\(P\)" is minimized. A detailed MathCAD calculation worksheet for the entire process is presented in Appendix D.

With the new empirical constants, the Cavallini et al. (1999) pure refrigerants model is modified and has the form

\[
h_n = h_{n0} \times P1 \times \left(X_n \right)^{p2} \times F_j^{p1} + h_j \times \Phi \times Rx^{p4} \times \left(Boh^{p6} F_r \right)^{p5} \times F_2^{p6} \times F_3^{p7}
\] (4.3)

The new empirical constants for the modified model are listed in table 4.2. The new empirical constants have accuracy of 4 significant digits. For comparison, the Cavallini et al. (1999) empirical constants are listed in the same table.

Table 4.2: Empirical Constants in the Cavallini et al. (1999) Pure Refrigerant Model and the Modified Pure Refrigerant Model

<table>
<thead>
<tr>
<th>Empirical Constant</th>
<th>Cavallini et al. (1999)</th>
<th>Modified Model</th>
</tr>
</thead>
<tbody>
<tr>
<td>(A / P1)</td>
<td>1.36</td>
<td>1.5160</td>
</tr>
<tr>
<td>(B / P2)</td>
<td>0.36</td>
<td>1.1610</td>
</tr>
<tr>
<td>(C / P3)</td>
<td>0.38</td>
<td>-1.7640</td>
</tr>
<tr>
<td>(S^* / P4)</td>
<td>2.14</td>
<td>2.6220</td>
</tr>
<tr>
<td>(T / P5)</td>
<td>-0.15 (G &lt; 500 \text{ kg/m}^2\text{-s})</td>
<td>-0.2158</td>
</tr>
<tr>
<td>(V / P6)</td>
<td>0.59</td>
<td>0.5927</td>
</tr>
<tr>
<td>(Z / P7)</td>
<td>0.36</td>
<td>0.0582</td>
</tr>
</tbody>
</table>

The accuracy of the prediction results for the modified model and the Cavallini et al. (1999) pure refrigerant model are presented in table 4.3.
Table 4.3 clearly shows that the prediction results improved significantly with the modified model. Strong improvements are observed in the R12 and the R134a data sets. These improvements are achieved without affecting the predictive accuracy for the R22 data sets.

Figure 4.1 illustrates the improvement achieved using the modified model on the Eckel et al. (1991) R12 data sets. The modified model gives a mean absolute deviation of 14.9% compared to the prediction results of 31.3% from the Cavallini et al. (1999) model. A significant improvement is observed. The modified model also captures the appropriate heat transfer coefficient, \( h_{tp} \), variation with increasing mass flux, \( G \).

Figure 4.2 presents part of the data from the Murata and Hashizume (1993) R123 data set and the prediction results from both the Cavallini et al. (1999) and the modified models. The modified model again provides better predictions, especially in the high quality region, where \( 0.6 < x < 0.9 \). For the entire R123 data set of Murata and Hashizume (1993), the modified model gives a mean absolute deviation of 11.2% compared to 13.1% achieved by the Cavallini et al. (1999) model.
Heat Transfer Coefficient, $h$

Vapor Quality, $x$

Cavallini et al. (1999) Model
Experimental Data
Modified Model

Figure 4.1: Comparison Between the Cavallini et al. (1999) and the Modified Pure Refrigerant Models on the Eckel et al. (1991) R12 Data Set

Figure 4.2: Comparison Between the Cavallini et al. (1999) and the Modified Pure Refrigerant Models on the Murata and Hashizume (1993) R123 Data Set
The modified model also generates excellent prediction results to the Eckels et al. (1994, 1998a, 1998b) data sets. Figures 4.3 to 4.6 illustrates the prediction results of the two models using the Eckel et al. R134a data sets. These four figures demonstrate that the modified model generates closer prediction for the R134a experimental data sets than the original Cavallini et al. (1999) model.

Figure 4.3: Comparison Between the Cavallini et al. (1999) and the Modified Pure Refrigerant Models on the Eckels et al. (1991) R134a Data Set
Figure 4.4: Comparison Between the Cavallini et al. (1999) and the Modified Pure Refrigerant Models on the Eckels et al. (1994) R134a Data Set

Figure 4.5: Comparison Between the Cavallini et al. (1999) and the Modified Pure Refrigerant Models on the Eckels et al. (1998a) R134a Data Set
Figure 4.6: Comparison Between the Cavallini et al. (1999) and the Modified Pure Refrigerant Models on the Eckels et al. (1998b) R134a Data Set

Figure 4.7 illustrates the prediction results of the two models for the Bogart and Thors (1999) R134a data set. In this case, the modified model only slightly improves the prediction result; however, it still over-predicts the experimental results for the low mass flux region, $30 < G < 100 \text{ kg/m}^2\text{-s}$. With the additional parameter, the boiling number $Bo$, included in the nucleate boiling term, to be discussed in the last chapter, the above over-prediction is likely to be reduced. Similar trends are also observed in the Bogart and Thors (1999) R22 data set shown in figure 4.8. For these two data sets, the modified model generates excellent predictions for mass fluxes, $G > 120 \text{ kg/m}^2\text{-s}$. 
Figure 4.7: Comparison Between the Cavallini et al. (1999) and the Modified Pure Refrigerant Models on the Bogart and Thors (1999) R134a Data Set

Figure 4.8: Comparison Between the Cavallini et al. (1999) and the Modified Pure Refrigerant Models on the Bogart and Thors (1999) R22 Data Set
Figure 4.9 shows the results of the two predictive models for the Bogart and Thors (1994) R22 data set. The Cavallini et al. (1999) model predicts the data set with a mean absolute deviation of 11.4%, and the modified model produces a lower mean absolute deviation of 6.5%. The modified model produces a closer match to the experimental data set.

![Graph showing comparison between models](image)

Figure 4.9: Comparison Between the Cavallini et al. (1999) and the Modified Pure Refrigerant Models on the Bogart and Thors (1994) R22 Data Set

Figure 4.10 presents a set of R22 data from the Hitachi Cable Review (1987) compared with the results of the two predictive models. The modified model provides better prediction result at medium quality region, $0.4 < x < 0.8$. The mean absolute deviation of the modified model is 15.5% compared to 23.3% for the Cavallini et al. (1999) model.
Figure 4.10: Comparison Between the Cavallini et al. (1999) and the Modified Pure Refrigerant Models on the Hitachi Cable (1987) R22 Data Set

Compared to the Cavallini et al. (1999) model, the modified model provides slightly higher mean absolute deviation for certain R22 data sets, for instance, Muzzio et al. (1998), Schlager (1988), and Yasuda et al. (1990). For data from Muzzio et al. (1998), the modified model provides about 2% higher mean absolute deviation than the Cavallini et al. (1999) model. As depicted in figure 4.11, the increase of the mean absolute deviation in this particular case is not significant. Figure 4.12 illustrates the higher mean absolute deviation in the modified model for the Schlager (1988) data set. The increase of the mean absolute deviation is not significant. Mean absolute deviations of the modified model observed from Yasuda et al. (1990) R22 data set are even higher. The prediction results are shown in figure 4.13. The mean absolute deviation from the modified model is 17.7% while the Cavallini et al. (1999) model achieves a mean absolute deviation at 11.6%, which is a 6% lower.
Figure 4.11: Comparison Between the Cavallini et al. (1999) and the Modified Pure Refrigerant Models on the Muzzio et al. (1998) R22 Data Set

Figure 4.12: Comparison Between the Cavallini et al. (1999) and the Modified Pure Refrigerant Models on the Schlager (1988) R22 Data Set
Overall, the modified model successfully generates excellent prediction results to most of the experimental data sets, which include the pure refrigerants R22, R12, R134a, and R123. The mean absolute deviation is around $\pm 16\%$. The modified model provides only slightly higher mean absolute deviation for the Eckel et al. (1991) R134a data set and the Bogart et al. (1999) R22 and R134a data sets.

With the success in generating better prediction results using the modified pure refrigerant model, better predictions are possible for the refrigerant mixture model using similar concepts in generating the empirical constants. For preliminary calculation, the Cavallini et al. (1999) mixture model from equation 3.9 is selected; however, the empirical constants in the nucleate boiling and convective boiling terms must be generated from refrigerant mixture data sets. The refrigerant mixture data are obtained from the available 155 data points shown in table 3.3. The two-phase heat transfer coefficient, $h_{tp,ideal}$, used in the mixture correction factor $F_c$ is obtained from the modified pure refrigerant model with the newly determined empirical constants listed in table 4.2.
Similar to what is done in the modified pure refrigerant model, all the independent parameters listed in table 4.1 must be generated from the 155 refrigerant mixture data points. Three additional independent parameters are required to account for the mixture effects. These additional independent parameters are listed in table 4.4 with their corresponding nomenclatures in the MathCAD 2000 environment.

Table 4.4: Additional Independent Parameters Required in the Modified Refrigerant Mixture Model

<table>
<thead>
<tr>
<th>Independent Parameter</th>
<th>Nomenclature / Parameters in MathCAD 2000</th>
</tr>
</thead>
<tbody>
<tr>
<td>( F_c )</td>
<td>( F_c )</td>
</tr>
<tr>
<td>( \frac{x \times c_m \times \Delta TG}{i_{fg}} )</td>
<td>( DQv )</td>
</tr>
<tr>
<td>( h_v )</td>
<td>( h_v )</td>
</tr>
</tbody>
</table>

For this case, the standard error of regression \( SER \) function becomes

\[
SER(M) = \left[ \frac{1}{N - 7} \left( \sum_{j=1}^{N} \left( \frac{1}{F_c \times h_{mb} + h_{vb}} \times \frac{x \times c_m \times \Delta TG}{i_{fg}} \times \frac{1}{h_v} \right) \right) \right]^{0.5} \tag{4.4}
\]

where

\[
h_{mb} = h_{mb} \times (M1 \times X_a^{M2}) \times F_1^{M3} \tag{4.5}
\]

and

\[
h_{vb} = h_1 \times \phi \times (B0n^W \times Fr^v)^M5 \times F_2^{M6} \times F_3^{M7} \tag{4.6}
\]

Variables \( M1, M2, M3, M4, M5, M6, \) and \( M7 \) are the new empirical constants to be determined from the refrigerant mixture data sets. Since there are seven coefficients in the equation 4.4 - 4.6, 7 is subtracted from total number of data points \( N \) in equation 4.4. A detailed MathCAD calculation worksheet for the entire correlation optimization process is presented in Appendix E.

With the new empirical constants from the refrigerant mixture data sets, the modified mixture model takes the form of equation 2.52 with the modifications in equations 4.5 and 4.6. The new empirical
constants for the modified mixture model are listed in table 4.5. The new empirical constants also have accuracy of 4 significant digits.

Table 4.5: Empirical Constants in the Modified Refrigerant Mixture Model

<table>
<thead>
<tr>
<th>Empirical Constant</th>
<th>Modified Mixture Model</th>
</tr>
</thead>
<tbody>
<tr>
<td>M1</td>
<td>0.7098</td>
</tr>
<tr>
<td>M2</td>
<td>1.2040</td>
</tr>
<tr>
<td>M3</td>
<td>3.3010</td>
</tr>
<tr>
<td>M4</td>
<td>0.8317</td>
</tr>
<tr>
<td>M5</td>
<td>0.1578</td>
</tr>
<tr>
<td>M6</td>
<td>-1.0780</td>
</tr>
<tr>
<td>M7</td>
<td>0.4217</td>
</tr>
</tbody>
</table>

The prediction results for the modified mixture model and the Cavallini et al. (1999) mixture model are presented in table 4.6.

Table 4.6: Mean Absolute Deviation (MAD) Achieved by the Cavallini et al. (1999) Mixture Model and the Modified Mixture Model

<table>
<thead>
<tr>
<th>Reference</th>
<th>Refrigerant</th>
<th>MAD Cavallini et al. (1999) Mixture Model</th>
<th>MAD Modified Mixture Model</th>
</tr>
</thead>
<tbody>
<tr>
<td>Bogart and Thors (1999)</td>
<td>R407c</td>
<td>63.1%</td>
<td>32.1%</td>
</tr>
<tr>
<td></td>
<td>R507a</td>
<td>50.1%</td>
<td>16.5%</td>
</tr>
<tr>
<td>Cui et al. (1986)</td>
<td>R502</td>
<td>46.6%</td>
<td>11.1%</td>
</tr>
<tr>
<td>Ebisu and Torikoshi (1998)</td>
<td>R407c</td>
<td>68.8%</td>
<td>3.8%</td>
</tr>
<tr>
<td>Kuo and Wang (1996a)</td>
<td>R407c</td>
<td>57.5%</td>
<td>26.4%</td>
</tr>
<tr>
<td>Murata and Hashizume (1993)</td>
<td>R123/</td>
<td>23.5%</td>
<td>21.5%</td>
</tr>
<tr>
<td></td>
<td>R134a</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

The modified mixture model does not provide significant improvement to the prediction results on the Murata and Hashizume (1993) R123/R134a mixtures data set. The mean absolute deviation achieved is around 22%, which is comparable with the results obtained from the Cavallini et al. (1999) mixture model.

On the other hand, the modified mixture model shows significant improvement on the prediction results for all the R407c data sets, especially the data set from Ebisu and Torikoshi (1998). Figure 4.14 illustrates the prediction results between the Cavallini et al. (1999) mixture model and the modified mixture model using the Bogart and Thors (1999) R407c data set. Figure 4.14 shows that the modified mixture...
model provides predictions relatively close to the experimental data. The modified mixture model, however, slightly over-predicts some data in the low mass flux region, $30 < G < 100 \text{ kg/m}^2\text{-s}$. The mean absolute deviation improves from 63.1% to 32.1%.

![Graph showing comparison between models](image)

Figure 4.14: Comparison Between the Cavallini et al. (1999) and the Modified Mixture Models on the Bogart and Thors (1999) R407c Data Set

Figure 4.15 exemplifies the best achievement of the modified mixture model on the R407c data set from Ebisu and Torikoshi (1998). The Cavallini et al. (1999) mixture model obtained a mean absolute deviation close to 70% on this data set, but the modified mixture model achieves the mean absolute deviation of about 4%.
The modified mixture model also provides better prediction results to the Kuo and Wang (1996a) R407c data sets. Again, the prediction results of the modified mixture model show a significant improvement. Overall, the modified mixture model is capable of predicting most of the available R407c data sets within an acceptable mean absolute deviation.

Figure 4.16 demonstrates part of the R502 data sets from Cui et al. (1986) compared with both the mixture models. The Cavallini et al. (1999) mixture model fails to predict the trend of the two-phase heat transfer coefficient, $h_{tp}$, with respect to the increasing mass flux, $G$. However, the modified mixture model successfully predicts the trend of the two-phase heat transfer coefficient, $h_{tp}$, with increasing mass flux. The mean absolute deviation from the modified mixture model is around 11%. The Cavallini et al. (1999) mixture model possessed a 46.6% mean absolute deviation for this data set.
Figure 4.16: Comparison Between the Cavallini et al. (1999) and the Modified Mixture Models on the Cui et al. (1986) R502 Data Set

The figure shows that the modified mixture model provides better prediction results than the original Cavallini et al. (1999) mixture model, especially for the experimental data in the low mass flux region, $30 < G < 100$ kg/m$^2$-s. The modified model slightly over-predicts the data set in the high mass flux region, $G > 250$ kg/m$^2$-s. The mean absolute deviation from the modified mixture model is 16.5%.
As a whole, the modified mixture model has better prediction capability than the Cavallini et al. (1999) mixture model. The modified model provides average mean absolute deviations around 20% on all the available refrigerant mixture data sets. The modified model generates slightly higher mean absolute deviation, around 30%, on the Bogart and Thors (1999) R407c data set.
CHAPTER V
CRITIQUES ON THE CAVALLINI et al. (1999) HEAT TRANSFER MODEL and FUTURE WORKS RECOMMENDATIONS

The original Cavallini et al. (1999) pure refrigerant model is validated to predict all the available R22 experimental data sets within an acceptable mean absolute deviation (~30%). However, this model provides high mean absolute deviation for most the R134a and the R12 experimental data sets. High deviations are observed in the Bogart et al. (1999) R134a data set and the Eckels et al. (1991) R134a and R12 data sets. The Cavallini et al. (1999) pure refrigerant model has been modified to provide acceptable prediction results to most of the pure refrigerant data sets.

The boiling number, $Bo$, is an important dimensionless parameter that characterizes the behavior of the boiling process, especially in nucleate boiling region. The mathematical form of $Bo$ is listed in equation 2.6. Jung et al. (1989) and Gungor and Winterton (1987) used the boiling number $Bo$ to account for nucleate boiling effect in their heat transfer model. Murata and Hashizume (1993) proved that the two-phase heat transfer coefficient is strongly dependent on the Martinelli parameter, $X_{tt}$, and the boiling number, $Bo$, in smooth tubes. For the case of micro-fin tube, Murata and Hashizume (1993) showed that the effect of the boiling number, $Bo$, in the nucleate boiling region becomes unclear, but its effect is still significant in the experimental data. The Cavallini et al. (1999) pure refrigerant model can be improved by including the boiling number, $Bo$, in the nucleate boiling heat transfer coefficient, $h_{nb}$, term as shown in equation 2.29.

The hydraulic diameter, $d_h$, is a more appropriate parameter to evaluate the micro-fin tube heat transfer surface. As discussed by Bejan (1995), the hydraulic diameter, $d_h$, is a characteristic length that
accounts for “how close” the wall and its resistive effect are relative to the fluid stream. Kedzierski and Goncalves (1999) proposed a hydraulic diameter for micro-fin tube

\[ d_h = \frac{4 \times A_e \times \cos(\gamma)}{n_f \times S_p} \]  

(5.1)

\( A_e \) represents the cross-sectional flow area, which is written as equation 5.2, and \( S_p \) is the perimeter of one fin and channel taken perpendicular to the axis of the fin. Figure 5.1 illustrates the various fin parameters used in the calculation of hydraulic diameter \( d_h \).

\[ A_e = \frac{\pi \times d_o^2}{4} - n_f \times A_f \]  

(5.2)

![Figure 5.1: Detailed Cross Sectional of a Micro-fin Tube](image)

As discussed earlier, all the empirical constants used in Cavallini et al. (1999) pure refrigerant model were evaluated from 361 data points where 338 of them were R22 data sets and only 23 were R12 data sets. The Cavallini et al. (1999) pure refrigerant model is behaved to provide less accurate prediction to experimental data sets for other than R22. If experimental data sets from different type of refrigerant are used to generate the empirical constants, the new modified model should have better prediction capability for different refrigerants. The modified pure refrigerant model, discussed in chapter 4, proves the validity of this statement.
Therefore, with all the recommendations stated above, the new evaporation heat transfer model for pure refrigerant should have the following form:

\[ h_{\text{gb}} = h_{\text{gb}} \times \left(P1 \times X_a \times P^2\right) \times Bo^P \times Fr^P \times \Phi \times Rx^P \times \left(Bonw^w \times Frw^w \times Fz^P \times Fj^P \times Fr^P \times Fr^P \times Fr^P \right) \]  \hspace{1cm} (5.3)

\( P1, P2, P3, P4, P5, P6, P7, \) and \( P8 \) are the new empirical constants which were evaluated from data sets consisting of different refrigerants. The correlation optimization discussed in chapter 4 can be used as calculation guidelines. The hydraulic diameter, \( d_h \), should replace the inner-tube diameter, \( d_i \), used in equations 2.32, 2.35, 2.36, and 2.37.

The Cavallini et al. (1999) refrigerant mixture model generally provides high deviation for most refrigerant mixture data. However, the model predicts the \( \text{R123/R134a (90/10 mole\%)} \) data set with an acceptable mean absolute deviation of 23.5\%. Among all the tested refrigerants shown in table 3.8, the mixture of R123 and R134a has the highest temperature glide, 15°C, at isobaric condition. According to Cavallini et al. (1999), their mixture model is only valid for a maximum total isobaric temperature glide around 7-8°C. Cavallini et al. (1999) assumed that the ratio between the sensible heat flow rate heating the vapor and the total heat flow rate (\( \bar{\alpha}Q_{\text{gb}} / \bar{\alpha}T \)) is approximately constant and takes the form:

\[ \frac{\partial Q_{\text{gb}}}{\partial T} = \frac{x \times c_{p,v} \times \Delta T G}{i_{gb}} \]  \hspace{1cm} (5.4)

Murata and Hashizume (1993) used equation 5.4 to define the ratio between the sensible heat flow rate heating the vapor and the total heat flow rate (\( \bar{\alpha}Q_{\text{gb}} / \bar{\alpha}T \)); however, with the assumptions they proposed, equation 5.4 does not have any temperature glide limitation. The assumptions proposed by Murata and Hashizume (1993) are considered reasonable, and equation 5.4 should be implemented without any limitation on the temperature glide.

The Cavallini et al. (1999) mixture model used all the empirical constants generated from pure refrigerant data sets. A new set of empirical constants should be generated based on the refrigerant mixture data. In other words, the recommended new refrigerant mixture model should take the form of equation 5.5 as recommended by Cavallini et al. (1999) with the appropriate modification in nucleate boiling and convective boiling components listed in equations 5.6 and 5.7.
\[ h_u = \left[ \frac{1}{F_c \times h_{ub} + h_{iv}} + \frac{x \times C_{p,v} \times \Delta T G}{i_{lg}} \times \frac{1}{h_v} \right]^{-1} \]  

(5.5)

where, \( h_{ub} = h_{pb} \times (M1 \times X_u^{M2} \times Bo^{M3} \times F_i^{M4}) \)  

(5.6)

\[ h_{iv} = h_i \times \Phi \times R_{x^{M3}} \times (Bo^{M5} \times Fr_2^{M6} \times Fr_3^{M7} \times Fr_4^{M8}) \]  

(5.7)

\( M1, M2, M3, M4, M5, M6, M7, \) and \( M8 \) are the new empirical constants to be evaluated from refrigerant mixture data.

The mixture correction factor, \( F_c \), remains unchanged and takes the form of equation 2.56. The \( h_{ip,ideal} \) listed in equation 2.56 should be calculated with the pure refrigerant correlation (equation 5.3) with refrigerant mixture properties.

The recommended pure refrigerant model listed in equation 5.3, and the refrigerant mixture model listed in equations 5.5 to 5.7, should generate better prediction results. The validity of these new models is to be determined in the near future.
CHAPTER VI
CONCLUSIONS

The Kido et al. (1995) model and the Cavallini et al. (1999) model are validated to predict the two-phase heat transfer coefficient. The Kido et al. (1995) model fails to provide accurate prediction for most of the experimental data sets, except for their R22 data set. The Cavallini et al. (1999) pure refrigerant model successfully generated good prediction results for all the available R22 data sets. However, higher mean absolute deviations are observed when the model is used for the R12 and R134a data sets. The Cavallini et al. (1999) mixture model provides relatively high mean absolute deviation for most of the refrigerant mixture data sets. Even though, the mixture model is proposed for zeotropic mixtures only, the prediction results suggest that the mixture model can provide relatively good prediction for azeotropic mixtures.

The Cavallini et al. (1999) pure refrigerant model was modified with a set of new empirical constants generated with the aid of MathCAD Minimize function. The modified pure refrigerant model provides relatively better prediction results than the original Cavallini et al. (1999) pure refrigerant model. This modified pure refrigerant model improves the prediction results on the R12 and R134a data sets without excessively reducing the model capability of predicting R22 data sets. The Cavallini et al. (1999) mixture model was also modified with the empirical constants generated from the refrigerant mixture data sets. The modified mixture model again successfully produces smaller mean absolute deviation on the available refrigerant mixture experimental data.

The modified pure refrigerant model and the modified refrigerant mixture model are two preliminary models implemented such that they prove the effectiveness of the Minimize function in the MathCAD 2000 software. These two modified models successfully achieve the preliminary objective of generating more reliable prediction results with lower mean absolute deviation.
With all the recommendations proposed in chapter 5, the new pure refrigerant model, equation 5.3, and the refrigerant mixture model, equations 5.5 – 5.7, are expected to generate more accurate predictions than the original Cavallini et al. (1999) heat transfer model.
REFERENCES


James, Mark. *Digitize XY Application 1.0.1*; A simple program designed for digitizing data from bitmaps of graph data; Newport News, VA, 1998.


REFPROP 5.12: A program to provide tables and plots for the thermodynamic and transport properties of pure refrigerants and refrigerant mixtures; National Institute of Standards and Technology: Boulder, CO, 1996

REFPROP, 6.01: A program to provide tables and plots for the thermodynamic and transport properties of pure refrigerants and refrigerant mixtures; National Institute of Standards and Technology: Boulder, CO, 1998


APPENDIX A

MathCAD Files for the Kido et al. (1995) Model
Sample Data File

Experimental Data File
for R123 in Micro-Fin Tubes
from Murata and Hashizume (1993) JHT v115n3 680

Define flow conditions.

\[ T_{\text{sat}} = 320 \ \text{K} \quad P_{\text{sat}} = 202650 \ \text{Pa} \quad q := \text{W} \cdot \text{m}^{-2} \]

\[ G := \text{kg} \cdot \text{m}^{-2} \cdot \text{s}^{-1} \quad g := 9.807 \ \text{m} \cdot \text{s}^{-2} \]

Import property table generated by REFPROP.

- C:\\..\R123.txt
- C:\\..\R123 Info.txt

Calculate thermodynamics and transport properties by MathCAD cubic-spline interpolation:

Reference: C:\\MFT Project\Correlation\Properties.med(R)

Define tube configuration:

Tube properties,
\[ d_o := 12.7 \times 10^{-3} \ \text{m} \quad d_i := 10.7 \times 10^{-3} \ \text{m} \]

Micro-fin properties,
\[ \gamma := 30^\circ \text{deg} \quad \beta := 50^\circ \text{deg} \quad n_f := 60 \]

\[ e := 0.3 \times 10^{-3} \ \text{m} \quad p := \frac{\pi \cdot d_i}{n_f} \quad p = 0.00056 \ \text{m} \]

\[ L := 0.734 \ \text{m} \]

Mean diameter \((d_m)\) for the Kido correlation,
\[ p := \frac{\pi \cdot d_i}{n_f} \quad t := 0 \quad b := \frac{\pi \cdot d_i}{n_f} - 2 \cdot e \cdot \tan \left( \frac{\beta}{2} \right) \]

\[ \text{AreaRatio} := \frac{(p - t - b)^2 + 4e^2}{p \cdot \cos(\gamma)} + t + b \]

\[ d_{\text{mean}} := \sqrt{\left( 1 - \frac{n_f \cdot b \cdot e}{2} \right)^2 + (d_i)^2} \]

\[ \text{AreaRatio} = 1.943 \]
Experimental Data

\[ G_1 := 93 \quad G_2 := 185 \quad G_3 := 278 \, \text{kg} \cdot \text{m}^{-2} \cdot \text{s}^{-1} \]
\[ q_1 := 10000 \quad q_2 := 30000 \, \text{W} \cdot \text{m}^{-2} \]

\[
\text{Data}_{\text{exp}_G1 \_q1} = \begin{pmatrix}
0 & 1 \\
0 & \\
1 & \\
2 & 
\end{pmatrix}
\]

\[
\text{hit}_{\text{p}_G1 \_q1} = \text{Data}_{\text{exp}_G1 \_q1}^{\langle 1 \rangle}
\]

\[
\chi_{\text{exp}_G1 \_q1} = \frac{\text{Data}_{\text{exp}_G1 \_q1}}{100}
\]

\[
\text{Data}_{\text{exp}_G2 \_q1} = \begin{pmatrix}
0 & 1 \\
0 & \\
1 & \\
2 & 
\end{pmatrix}
\]

\[
\text{hit}_{\text{p}_G2 \_q1} = \text{Data}_{\text{exp}_G2 \_q1}^{\langle 1 \rangle}
\]

\[
\chi_{\text{exp}_G2 \_q1} = \frac{\text{Data}_{\text{exp}_G2 \_q1}}{100}
\]

\[
\text{Data}_{\text{exp}_G3 \_q1} = \begin{pmatrix}
0 & 1 \\
0 & \\
1 & \\
2 & 
\end{pmatrix}
\]

\[
\text{hit}_{\text{p}_G3 \_q1} = \text{Data}_{\text{exp}_G3 \_q1}^{\langle 1 \rangle}
\]

\[
\chi_{\text{exp}_G3 \_q1} = \frac{\text{Data}_{\text{exp}_G3 \_q1}}{100}
\]
\[\text{Data}_{\text{exp G1 q2}}=\begin{array}{|c|c|}
\hline
0 & 1 \\
\hline
0 &  \\
1 &  \\
2 &  \\
\hline
\end{array}\]

\[h_{\text{tp exp G1 q2}}=\text{Data}_{\text{exp G1 q2}}\frac{\langle \psi \rangle}{\langle \phi \rangle}\]

\[x_{\text{exp G1 q2}}=\frac{\text{Data}_{\text{exp G1 q2}}}{100}\]

\[\text{Data}_{\text{exp G2 q2}}=\begin{array}{|c|c|}
\hline
0 & 1 \\
\hline
0 &  \\
1 &  \\
2 &  \\
\hline
\end{array}\]

\[h_{\text{tp exp G2 q2}}=\text{Data}_{\text{exp G2 q2}}\frac{\langle \psi \rangle}{\langle \phi \rangle}\]

\[x_{\text{exp G2 q2}}=\frac{\text{Data}_{\text{exp G2 q2}}}{100}\]

\[\text{Data}_{\text{exp G3 q2}}=\begin{array}{|c|c|}
\hline
0 & 1 \\
\hline
0 &  \\
1 &  \\
2 &  \\
\hline
\end{array}\]

\[h_{\text{tp exp G3 q2}}=\text{Data}_{\text{exp G3 q2}}\frac{\langle \psi \rangle}{\langle \phi \rangle}\]

\[x_{\text{exp G3 q2}}=\frac{\text{Data}_{\text{exp G3 q2}}}{100}\]
Sample Model File

Source of Correlations: Kido et al. (1995)

Define Martinelli parameter,

\[
X_H(x) := \left( \frac{1 - x}{x} \right) \times \left( \frac{\rho}{\rho_1} \right)^{0.5} \times \left( \frac{\mu_1}{\mu} \right)^{0.1}
\]

Define Boiling number,

\[
Bo(x, q, G) := \frac{q}{Gd_{\text{mean}}}
\]

Define YY factor by Yoshida (1983),

\[
YY(x, q, G) := Bo(x, q, G) 	imes 10^4 + 0.23 \left( Bo(x, q, G) 	imes 10^4 \right)^{0.69} \times X_H(x)^{-2.0}
\]

Define Reynolds number,

\[
Re(x, q, G) := \frac{Gd_{\text{mean}}}{\mu_1}
\]

Define Prandtl number

\[
Pr = \text{Pr}
\]

Define modified bond number, \( Bon^* \)

\[
Bon_{\text{star}} := \frac{p^2 \cdot g^2 \cdot (\rho_1 - \rho)}{\sigma} \left( \frac{p}{d_{\text{mean}}} \right) \left( \frac{p}{e} \right) \left( \frac{b}{c} \right)
\]

Define two-phase Nusselt number and two-phase heat transfer coefficient,

\[
Nu_{tp}(x, q, G) := \frac{2.5 \cdot YY(x, q, G)^{0.2} \cdot Re(x, q, G)^{0.27} \cdot Pr_1^{0.4} \cdot (1 + \tan(\gamma))^{1.8}}{Bon_{\text{star}}^{0.2}}
\]

\[
h_{tp}(x, q, G) := \frac{k_p \cdot Nu_{tp}(x, q, G)}{d_{\text{mean}}} \cdot \text{AreaRatio}
\]
Sample Property File

Cubic Spline interpolation for all required properties

\[ P := \text{TABLE}^{(0)} \]
\[ T_{1} := \text{TABLE}^{(1)} \]
\[ T_{v} := \text{TABLE}^{(13)} \]
\[ \rho_{1} := \text{TABLE}^{(2)} \]
\[ \rho_{v} := \text{TABLE}^{(3)} \]
\[ i_{1} := \text{TABLE}^{(4)} \]
\[ i_{v} := \text{TABLE}^{(5)} \]
\[ c_{p_{1}} := \text{TABLE}^{(6)} \]
\[ c_{p_{v}} := \text{TABLE}^{(7)} \]
\[ \mu_{1} := \text{TABLE}^{(8)} \]
\[ \mu_{v} := \text{TABLE}^{(9)} \]
\[ k_{1} := \text{TABLE}^{(10)} \]
\[ k_{v} := \text{TABLE}^{(11)} \]
\[ \sigma_{1} := \text{TABLE}^{(12)} \]

\[ \rho_{1_{s}} := \text{cspline}(P, \rho_{1}) \]
\[ \rho_{v_{s}} := \text{cspline}(P, \rho_{v}) \]
\[ i_{1_{s}} := \text{cspline}(P, i_{1}) \]
\[ i_{v_{s}} := \text{cspline}(P, i_{v}) \]
\[ c_{p_{1_{s}}} := \text{cspline}(P, c_{p_{1}}) \]
\[ c_{p_{v_{s}}} := \text{cspline}(P, c_{p_{v}}) \]
\[ \mu_{1_{s}} := \text{cspline}(P, \mu_{1}) \]
\[ \mu_{v_{s}} := \text{cspline}(P, \mu_{v}) \]
\[ k_{1_{s}} := \text{cspline}(P, k_{1}) \]
\[ k_{v_{s}} := \text{cspline}(P, k_{v}) \]
\[ \sigma_{1_{s}} := \text{cspline}(P, \sigma_{1}) \]
\[ T_{1_{s}} := \text{cspline}(P, T_{1}) \]
\[ T_{v_{s}} := \text{cspline}(P, T_{v}) \]

\[ \rho_{1}(PP) := \text{interp}(\rho_{1_{s}}, P, \rho_{1_{s}}, PP) \]
\[ \rho_{v}(PP) := \text{interp}(\rho_{v_{s}}, P, \rho_{v_{s}}, PP) \]
\[ i_{1}(PP) := \text{interp}(i_{1_{s}}, P, i_{1_{s}}, PP) \]
\[ i_{v}(PP) := \text{interp}(i_{v_{s}}, P, i_{v_{s}}, PP) \]
\[ c_{p_{1}}(PP) := \text{interp}(c_{p_{1_{s}}}, P, c_{p_{1_{s}}}, PP) \]
\[ c_{p_{v}}(PP) := \text{interp}(c_{p_{v_{s}}}, P, c_{p_{v_{s}}}, PP) \]
\[ \mu_{1}(PP) := \text{interp}(\mu_{1_{s}}, P, \mu_{1_{s}}, PP) \]
\[ \mu_{v}(PP) := \text{interp}(\mu_{v_{s}}, P, \mu_{v_{s}}, PP) \]
\[ k_{1}(PP) := \text{interp}(k_{1_{s}}, P, k_{1_{s}}, PP) \]
\[ k_{v}(PP) := \text{interp}(k_{v_{s}}, P, k_{v_{s}}, PP) \]
\[ \sigma_{1}(PP) := \text{interp}(\sigma_{1_{s}}, P, \sigma_{1_{s}}, PP) \]
\[ \sigma_{v}(PP) := \text{interp}(\sigma_{v_{s}}, P, \sigma_{v_{s}}, PP) \]
\[ T_{1}(PP) := \text{interp}(T_{1_{s}}, P, T_{1_{s}}, PP) \]
\[ T_{v}(PP) := \text{interp}(T_{v_{s}}, P, T_{v_{s}}, PP) \]
Define fluid properties: (Source: REProp 6.01)

\[ \rho_1 := \rho \left( \frac{P_{\text{sat}}}{10^{-6}} \right) \]
\[ \rho_v := \rho_v \left( \frac{P_{\text{sat}}}{10^{-6}} \right) \]
\[ c_{p_1} := c_{p_1} \left( \frac{P_{\text{sat}}}{10^{-6}} \right) \times 10^3 \]
\[ c_{p_v} := c_{p_v} \left( \frac{P_{\text{sat}}}{10^{-6}} \right) \times 10^3 \]
\[ \mu_1 := \mu \left( \frac{P_{\text{sat}}}{10^{-6}} \right) \times 10^{-6} \]
\[ \mu_v := \mu_v \left( \frac{P_{\text{sat}}}{10^{-6}} \right) \times 10^{-6} \]
\[ k_1 := k_1 \left( \frac{P_{\text{sat}}}{10^{-6}} \right) \]
\[ k_v := k_v \left( \frac{P_{\text{sat}}}{10^{-6}} \right) \]
\[ i_{fg} := \left( i_i \left( \frac{P_{\text{sat}}}{10^{-6}} \right) - i_f \left( \frac{P_{\text{sat}}}{10^{-6}} \right) \right) \times 10^3 \]
\[ \sigma := \sigma \left( \frac{P_{\text{sat}}}{10^{-6}} \right) \]
\[ \Delta T_G := \left| T_v \left( \frac{P_{\text{sat}}}{10^{-6}} \right) - T_i \left( \frac{P_{\text{sat}}}{10^{-6}} \right) \right| \]
\[ \text{Pr}_1 := \frac{\mu \cdot c_{p_1}}{k_1} \]
\[ P_c := \left| \text{INFO}^{(1)} \right| \times 10^6 \]
\[ M := \left| \text{INFO}^{(0)} \right| \]

\[ \rho_1 = \text{kg} \cdot m^{-3} \]
\[ \rho_v = \text{kg} \cdot m^{-3} \]
\[ c_{p_1} = \text{J} \cdot \text{kg}^{-1} \cdot \text{K}^{-1} \]
\[ c_{p_v} = \text{J} \cdot \text{kg}^{-1} \cdot \text{K}^{-1} \]
\[ \mu_1 = \left( 10^{-6} \right) \text{Pa} \cdot \text{s} \]
\[ \mu_v = \left( 10^{-6} \right) \text{Pa} \cdot \text{s} \]
\[ k_1 = \text{W} \cdot \text{m}^{-1} \cdot \text{K}^{-1} \]
\[ k_v = \text{W} \cdot \text{m}^{-1} \cdot \text{K}^{-1} \]
\[ i_{fg} = \left( 10^3 \right) \text{J} \cdot \text{kg}^{-1} \]
\[ \sigma = \text{N} \cdot \text{m}^{-1} \]
\[ \Delta T_G = \text{K} \]
\[ \text{Pr}_1 = \cdot \]
\[ P_c = \cdot \]
\[ M = \cdot \]
Sample Calculation File

Source of Correlations:

Source of Experimental Data:
1. Murata et al. (1993) JHT v115, n3, 680

Import experimental data and experimental conditions from Murata and Hashizume (1993)
Import tube configuration:

Reference: C:\MFT Project\Correlation\Data Murata R123 MFT JHT_115_3_680_1993.mcd(R)

Import Kido et al. (1995) Correlation:
Reference: C:\MFT Project\Correlation\Kido Pure Model - h vs x.mcd(R)

Define range of quality,

\[ x := 0.1, 0.11..0.8 \]

\[ G_1 = \quad G_2 = \quad G_3 = \quad \text{kg m}^{-2} \text{s}^{-1} \]

\[ q_1 = \quad q_2 = \quad \text{W m}^{-2} \]
WRITEPRN("temp1.dat") := \text{htp}(x_{\text{exp}_{G1}}q1,G1)
WRITEPRN("temp2.dat") := \text{htp}_{G1}q1

APPENDPRN("temp1.dat") := \text{htp}(x_{\text{exp}_{G2}}q1,G2)
APPENDPRN("temp2.dat") := \text{htp}_{G2}q1

APPENDPRN("temp1.dat") := \text{htp}(x_{\text{exp}_{G3}}q1,G3)
APPENDPRN("temp2.dat") := \text{htp}_{G3}q1

APPENDPRN("temp1.dat") := \text{htp}(x_{\text{exp}_{G1}}q2,G1)
APPENDPRN("temp2.dat") := \text{htp}_{G1}q2

APPENDPRN("temp1.dat") := \text{htp}(x_{\text{exp}_{G2}}q2,G2)
APPENDPRN("temp2.dat") := \text{htp}_{G2}q2

APPENDPRN("temp1.dat") := \text{htp}(x_{\text{exp}_{G3}}q2,G3)
APPENDPRN("temp2.dat") := \text{htp}_{G3}q2
Plot for $h_{\text{predicted}}$ versus $h_{\text{experiment}}$

$h_{\text{predicted}} = \text{READPRN("temp1.dat")}$

$h_{\text{experiment}} = \text{READPRN("temp2.dat")}$

$N := \text{length}(h_{\text{predicted}})$

$N = 1$

$MAD = \frac{1}{N} \sum \frac{|h_{\text{experiment}} - h_{\text{predicted}}|}{h_{\text{experiment}}}$

$MAD = 1\%$
APPENDIX B

MathCAD Files for the Cavallini et al. (1999)  
Pure Refrigerant Model
Sample Model File

Source of Correlation: Cavallini et al. (1999)

Define inner tube diameter based on Cavallini correlation,

\[ d_i := d_i - 2e \quad d_i = \text{m} \]

Define empirical constants,

\[ A := 1.36 \quad B := 0.36 \quad C := 0.38 \quad S_{\text{star}} := 2.14 \]

\[ T(G) := \begin{cases} 
-0.15 & \text{if } G < 500 \\
-0.21 & \text{if } G \geq 500 
\end{cases} \]

\[ V := 0.59 \quad Z := 0.36 \]

Define Martinelli parameter,

\[ X_T(x) := \left( \frac{1 - x}{x} \right) ^{0.9} \left( \frac{\rho}{\rho_1} \right) ^{0.5} \left( \frac{\mu_1}{\mu_v} \right) ^{0.1} \]

Define suppression factor (Cavallini, 1999),

\[ S(x, q, G) := \begin{cases} 
1 & \text{if } X_T(x) > 1 \\
X_T(x) & \text{otherwise} 
\end{cases} \]

Define pool boiling correlation (Cooper, 1984),

\[ h_{p_b}(x, q, G) := 55 \left( \frac{P_{\text{sat}}}{P_c} \right) ^{0.12} \left( -\log \left( \frac{P_{\text{sat}}}{P_c} \right) ^{0.55} \cdot M \cdot 0.5 \cdot q \right) \]

Define flow boiling effect in smooth tube for nucleate boiling component (Steiner, 1993),

\[ F_1 := \left( \frac{0.01}{d_i} \right) \]

Define two-phase multiplier, \( \Phi \)

\[ \Phi(x) := \left[ (1 - x) + 2.63x \left( \frac{\rho_1}{\rho_v} \right) ^{0.5} \right] ^{0.8} \]
Define liquid phase heat transfer coefficient (Dittus-Boelter, 1930),

\[
h_f(x, q, G) := 0.023 \frac{k_l}{d_i} \left( \frac{G d_i}{\mu_1} \right)^{0.8} \left( \frac{c_p \rho H_1}{k_l} \right)^{0.3}
\]

Define enhancement factor due to micro-fin (Hori and Shinohara (1991),

\[
R_x := \left[ \frac{2 \pi n \phi \left( 1 - \sin \left( \frac{\beta}{2} \right) \right)}{\pi d_i \cos \left( \frac{\beta}{2} \right)} + 1 \right] \left[ \frac{1}{\cos(\gamma)} \right]
\]

Define Bond number from Webb (1988),

\[
BonW := \frac{g \rho \phi \epsilon \pi n d_i}{8 \pi \phi n_f}
\]

Define Froude number,

\[
Fr_v(G) := \left( \frac{G}{\rho v} \right)^2 \times \frac{1}{g d_i}
\]

Define flow parameters,

\[
F_2 := \left( \frac{0.01}{d_i} \right)
\]

\[
F_3(G) := \left( \frac{100}{G} \right)
\]

Define two-phase heat transfer coefficient (Cavallini, 1999),

\[
h_p(x, q, G) := h_p b(x, q, G) A S(x, q, G) B F_1 C \\
+ h_f(x, q, G) \Phi (x) R_x S_{star} B \left( \frac{G}{BonW} \right) D F_2 V F_3(G) Z
\]
Sample Calculation File

Source of Correlations:
1. Cavallini et al. (1999)

Source of Experimental Data:
1. Murata et al. (1993) JHT v115, n3, 680

Import experimental data and experimental conditions from Murata and Hashizume (1993):

Import Cavallini model (1999):

Define range of quality,

\[ x := 0.05, 0.06 \ldots 1 \]

\[ G1 = \quad G2 = \quad G3 = \quad \text{kg}\cdot\text{m}^{-2}\cdot\text{s}^{-1} \]

\[ q1 = \quad q2 = \quad \text{W}\cdot\text{m}^{-2} \]
\[ N := \text{length}(htp\_exp\_G1\_q) \]

\[
\text{MAD}_{q1\_G1} = \frac{1}{N} \sum \frac{|htp\_exp\_G1\_q1 - htp(\bar{X}_{exp\_G1\_q1}\_G1)|}{htp\_exp\_G1\_q1} \quad \text{MAD}_{q1\_G1} = 1\% 
\]

\[ N := \text{length}(htp\_exp\_G2\_q) \]

\[
\text{MAD}_{q1\_G2} = \frac{1}{N} \sum \frac{|htp\_exp\_G2\_q1 - htp(\bar{X}_{exp\_G2\_q1}\_G2)|}{htp\_exp\_G2\_q1} \quad \text{MAD}_{q1\_G2} = 1\% 
\]

\[ N := \text{length}(htp\_exp\_G3\_q) \]

\[
\text{MAD}_{q1\_G3} = \frac{1}{N} \sum \frac{|htp\_exp\_G3\_q1 - htp(\bar{X}_{exp\_G3\_q1}\_G3)|}{htp\_exp\_G3\_q1} \quad \text{MAD}_{q1\_G3} = 1\% 
\]
\[
N := \text{length}(htp_{exp \_G1 \_q})
\]

\[
\text{MAD}_{q2 \_G1} := \frac{1}{N} \sum \frac{|htp_{exp \_G1 \_q2} - htp(x_{exp \_G1 \_q3 q2, G1})|}{htp_{exp \_G1 \_q2}}
\]

\[
\text{MAD}_{q2 \_G2} := \frac{1}{N} \sum \frac{|htp_{exp \_G2 \_q2} - htp(x_{exp \_G2 \_q3 q2, G2})|}{htp_{exp \_G2 \_q2}}
\]

\[
\text{MAD}_{q2 \_G3} := \frac{1}{N} \sum \frac{|htp_{exp \_G3 \_q2} - htp(x_{exp \_G3 \_q3 q2, G3})|}{htp_{exp \_G3 \_q2}}
\]

\[
\text{WRITEPRN}("temp1.dat") := htp(x_{exp \_G1 \_q1 \_q1, G1})
\]

\[
\text{WRITEPRN}("temp2.dat") := htp_{exp \_G1 \_q1}
\]

\[
\text{APPENDPRN}("temp1.dat") := htp(x_{exp \_G2 \_q1 \_q1, G2})
\]

\[
\text{APPENDPRN}("temp2.dat") := htp_{exp \_G2 \_q1}
\]

\[
\text{APPENDPRN}("temp1.dat") := htp(x_{exp \_G3 \_q1 \_q1, G3})
\]

\[
\text{APPENDPRN}("temp2.dat") := htp_{exp \_G3 \_q1}
\]

\[
\text{APPENDPRN}("temp1.dat") := htp(x_{exp \_G1 \_q3 q2, G1})
\]

\[
\text{APPENDPRN}("temp2.dat") := htp_{exp \_G1 \_q2}
\]

\[
\text{APPENDPRN}("temp1.dat") := htp(x_{exp \_G2 \_q3 q2, G2})
\]

\[
\text{APPENDPRN}("temp2.dat") := htp_{exp \_G2 \_q2}
\]

\[
\text{APPENDPRN}("temp1.dat") := htp(x_{exp \_G3 \_q3 q2, G3})
\]

\[
\text{APPENDPRN}("temp2.dat") := htp_{exp \_G3 \_q2}
\]
Plot for $h_{\text{predicted}}$ versus $h_{\text{experiment}}$

$h_{\text{predicted}} = \text{READPRN("temp1.dat")}$

$h_{\text{experiment}} = \text{READPRN("temp2.dat")}$

\[
N := \text{length}(h_{\text{predicted}}) \quad \quad \quad N = 1
\]

\[
\text{MAD} := \frac{1}{N} \sum \frac{|h_{\text{experiment}} - h_{\text{predicted}}|}{h_{\text{experiment}}} \quad \quad \quad \text{MAD} = 1\%
\]
APPENDIX C

MathCAD Files for the Cavallini et al. (1999) Refrigerant Mixtures Model
Sample Model File

Source of Correlations: Cavallini et al. (1999)

Define inner tube diameter based on Cavallini correlation,
\[ d_i := d_i - 2\epsilon \quad d_i = 1 \text{ m} \]

Define empirical constants,
\[ A := 1.36 \quad B := 0.36 \quad C := 0.38 \quad S_{\text{star}} := 2.14 \]
\[ T(G) := \begin{cases} -0.15 & \text{if } G < 500 \\ -0.21 & \text{if } G \geq 500 \end{cases} \quad V := 0.59 \quad Z := 0.36 \]

Define Martinelli parameter,
\[ X_{\text{M}}(x) := \left[ 1 - \frac{1}{x} \right] 0.9 \left( \frac{\rho_v}{\rho} \right) 0.5 \left( \frac{\mu_1}{\mu_v} \right) 0.1 \]

Define suppression factor (Cavallini, 1999),
\[ S(x, q, G) := \begin{cases} 1 & \text{if } X_{\text{M}}(x) > 1 \\ X_{\text{M}}(x) & \text{otherwise} \end{cases} \]

Define pool boiling correlation (Cooper, 1984),
\[ h_{pb}(x, q, G) := 55 \left( \frac{P_{\text{sat}}}{P_c} \right) 0.12 \times \left( -\log \left( \frac{P_{\text{sat}}}{P_c} \right) \right) 0.55 \times M 0.5 \times 0.67 \]

Define flow boiling effect in smooth tube for nucleate boiling component (Steiner, 1993),
\[ F_1 := \left( \frac{0.01}{d_i} \right) \]

Define two-phase multiplier, \( \Phi \)
\[ \Phi(x) := \left[ 1 - x + 2.63x \left( \frac{\rho_1}{\rho_v} \right) 0.5 \right] 0.8 \]

Define liquid phase heat transfer coefficient (Dittus-Boelter, 1930),
\[ h_l(x, q, G) := 0.023 \sqrt{\frac{Gd_i}{\mu_1}} 0.8 \left( \frac{c_{pl} \mu_1}{k_l} \right) 0.3 \]
Define enhancement factor due to micro-fin (Hori and Shinohara (1991)),

\[
Rx := \left[ \frac{2\pi n_f \left( 1 - \sin \left( \frac{\beta}{2} \right) \right)}{\pi d_i \cos \left( \frac{\beta}{2} \right)} + 1 \right] \frac{1}{\cos(\gamma)}
\]

Define Bond number from Webb (1988),

\[
BonW := \frac{g \varphi \rho \pi n_f \xi_i}{8 \sigma \alpha}
\]

Define Froude number,

\[
Fr_v(G) := \left( \frac{G}{\rho \lambda} \right)^2 \frac{1}{g d_i}
\]

Define flow parameters,

\[
F_2 := \left( \frac{0.01}{d_i} \right)
\]

\[
F_3(G) := \left( \frac{100}{G} \right)
\]

Define two-phase heat transfer coefficient (Cavallini, 1999),

\[
h_{lp}(x, q, G) := h_{pb}(x, q, G) + X \cdot S(x, q, G) \cdot F_1 ... + h_f(x, q, G) \cdot \Phi(x) \cdot Rx \cdot S_{star} \cdot (BonW \cdot Fr_v(G)) \cdot T(\xi) \cdot F_2 \cdot V \cdot F_3(G) \cdot Z
\]
Define mass transfer coefficient and mixture correction factor, (Thome, 1996),

\[ \lambda := 0.0003 \frac{m}{s} \]

\[ F_c(x, q, G) := \left[ 1 + \left( \frac{h_{tp}(x, q, G) \triangle T_G}{q} \left( 1 - \exp \left( - \frac{q}{\rho f_{fg} \lambda} \right) \right) \right) \right]^{-1} \]

Define heat transfer coefficient for liquid film, (Cavallini, 1999),

\[ h_l(x, q, G) := h_{pl}(x, q, G) \Delta \delta(x, q, G) B_{F_1} C_{F_2} T(G) \]

\[ + h_l(x, q, G) \Delta \delta (x) - R_x S_{\text{star}} (\text{BonW} \cdot \text{Fr}_v(G)) T(G) \cdot F_2 V \cdot F_3(G) \]

Define heat transfer coefficient for vapor phase, (Dittus-Boelter, 1930),

\[ h_v(x, q, G) := 0.023 \frac{k_v}{d_1} \left( \frac{Gd_1}{\mu_v} \right)^{0.8} \left( \frac{c_p v^2 \mu_v}{k_v} \right)^{0.3} \]

Define two-phase heat transfer coefficient for mixture (Cavallini, 1999),

\[ h_{tp-mf}(x, q, G) := \left[ \frac{1}{F_c(x, q, G) \cdot h_{pl}(x, q, G) \cdot \Delta \delta(x, q, G) B_{F_1} C_{F_2} \cdot T(G) \cdot F_2 V \cdot F_3(G) \cdot h_v(x, q, G)} \right]^{-1} \]
Sample Calculation File

Source of Correlations:
1. Cavallini et al. (1999)

Source of Experimental Data:
1. Ebisu et al. (1998) AT v104, n2, 1044

Import experimental data and experimental conditions from Ebisu and Torikoshi (1998):

Reference: C:\MFT Project\Correlation\Data Ebisu R407c MFT AT_104_2_1044_1998.mcd(R)

Import Cavallini model (1999):

Reference: C:\MFT Project\Correlation\Cavallini Mixtures Model - h vs x.mcd(R)

Define range of quality,
\[ x := 0.2, 0.21, \ldots, 0.9 \]

\[ G_1 = \text{kg} \cdot \text{m}^{-2} \cdot \text{s}^{-1} \]

\[ q_1 = \text{W} \cdot \text{m}^{-2} \]

\[ H = \text{Heat Transfer Coefficient} \]

\[ \text{Vapor Quality} \]

\[ N := \text{length}(h_{\text{tp}} \cdot m \cdot \text{exp}_G1 \cdot q1) \]

\[ N = 1 \]

\[ \text{MAD}_{q1,G1} := \frac{1}{N} \sum \frac{|h_{\text{tp}} \cdot m \cdot \text{exp}_G1 \cdot q1 - h_{\text{tp}} \cdot m \cdot \text{exp}_G1 \cdot q1|}{h_{\text{tp}} \cdot m \cdot \text{exp}_G1 \cdot q1} \]

\[ \text{MAD}_{q1,G1} = 1 \text{%} \]

WRITEPRN("temp1.dat") := \( h_{\text{tp}} \cdot m \cdot \text{exp}_G1 \cdot q1, G1 \)

WRITEPRN("temp2.dat") := \( h_{\text{tp}} \cdot m \cdot \text{exp}_G1 \cdot q1 \)
Define range of mass flux,

\[
G := 100, 101..450 \quad x_1 = 1 \quad q_1 = 1 \quad \text{Wm}^{-2}
\]

\[
\text{Mass Flux}
\]

\[
N := \text{length}(htp\_m\_exp\_x1\_q) \quad N = 1
\]

\[
\text{MAD}_{q_1\_x1} := \frac{1}{N} \sum \frac{|htp\_m\_exp\_x1\_q1 - htp\_m(x1,q1,Ge_{exp\_x1\_q})|}{htp\_m\_exp\_x1\_q1}
\]

\[
\text{MAD}_{q_1\_x1} = 1\%
\]

\[
\text{APPENDPRN}("temp1.dat") := htp\_m(x1,q1,Ge_{exp\_x1\_q})
\]

\[
\text{APPENDPRN}("temp2.dat") := htp\_m\_exp\_x1\_q
\]
Plot for $h_{\text{predicted}}$ versus $h_{\text{experiment}}$

$h_{\text{predicted}} = \text{READPRN("temp1.dat")}$
$h_{\text{experiment}} = \text{READPRN("temp2.dat")}$

\[ N := \text{length}(h_{\text{predicted}}) \]

\[
\text{MAD} = \frac{1}{N} \sum \frac{|h_{\text{experiment}} - h_{\text{predicted}}|}{h_{\text{experiment}}}
\]

\[ N = 1 \]

\[ \text{MAD} = 1\% \]
APPENDIX D

MathCAD Worksheet for Pure Refrigerant Correlation Optimization
Define independent parameters using Cavallini et al. (1999) heat transfer model

\[
\begin{align*}
\text{htp} & := \text{READPRN}("htp.dat") \\
\text{hp} & := \text{READPRN}("hp.dat") \\
\text{hl} & := \text{READPRN}("hl.dat") \\
\text{Xtt} & := \text{READPRN}("Xtt.dat") \\
\text{Rx} & := \text{READPRN}("Rx.dat") \\
\Phi & := \text{READPRN}("\Phi.dat") \\
\text{F} & := \text{READPRN}("F.dat") \\
\text{F2} & := \text{READPRN}("F2.dat") \\
\text{F3} & := \text{READPRN}("F3.dat") \\
\text{BonWFr} & := \text{READPRN}("\text{BonWFr.dat}")
\end{align*}
\]

\[
\begin{align*}
\text{N} & := \text{length(htp)} \\
\text{N} & := \text{length(hp)} \\
\text{N} & := \text{length(hl)} \\
\text{N} & := \text{length(Xtt)} \\
\text{N} & := \text{length(BonWFr)} \\
\text{N} & := \text{length(\Phi)} \\
\text{N} & := \text{length(F)} \\
\text{N} & := \text{length(F2)} \\
\text{N} & := \text{length(F3)}
\end{align*}
\]

All \(N\) values must be equal

\[
\text{Unknown} = \mathbf{P} \\
\text{SER(} \mathbf{P} \text{)} := \sqrt{\sum_{i=1}^{N} \left[ \text{ht}_{i} \cdot \text{P}_{1} \cdot (X_{i})^{2} + \text{hp}_{i} \cdot \text{P}_{2} \cdot (F_{i})^{2} + \text{hl}_{i} \cdot \Phi_{i} \cdot \text{P}_{3} \cdot (\text{Rx}_{i})^{2} + \text{BonWFr}_{i} \cdot \text{P}_{4} \cdot (\text{F2}_{i})^{2} + \text{F3}_{i} \cdot \text{P}_{5} \right]^{2} / (N - 7)}
\]

<table>
<thead>
<tr>
<th>(0)</th>
<th>(0)</th>
<th>(0)</th>
<th>(0)</th>
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</thead>
</table>
\[
\begin{align*}
\mathbf{P} & := \\
\mathbf{Min} & := \text{Minimize(SER, } \mathbf{P} \text{)} \\
\text{SER(Min)} & = \text{Min}
\end{align*}
\]
APPENDIX E

MathCAD Worksheet for Refrigerant Mixtures
Correlation Optimization
Define independent variables from data files.

\[
\begin{align*}
\text{htp} & := \text{READPRN}("htp.dat") \\
\text{hnb} & := \text{READPRN}("hnb.dat") \\
\text{hcb} & := \text{READPRN}("hcb.dat") \\
\text{DQv} & := \text{READPRN}("DQv.dat") \\
\text{hv} & := \text{READPRN}("hv.dat") \\
\text{Fc} & := \text{READPRN}("Fc.dat") \\
\text{hp} & := \text{READPRN}("hp.dat") \\
\text{Xtt} & := \text{READPRN}("Xtt.dat") \\
\text{Rx} & := \text{READPRN}("Rx.dat") \\
\text{hl} & := \text{READPRN}("hl.dat") \\
\text{Phi} & := \text{READPRN}("Phi.dat") \\
\text{F1} & := \text{READPRN}("F1.dat") \\
\text{F2} & := \text{READPRN}("F2.dat") \\
\text{F3} & := \text{READPRN}("F3.dat") \\
\text{BonWFr} & := \text{READPRN}("BonWFr.dat")
\end{align*}
\]

For all \( N \) values must be equal

Unknown = M

Based on Root-Sum-Square eqn.

\[
\text{SER}(M) := \sqrt{\sum_{i=1}^{N} \left[ \frac{1}{\text{htp}_i - \frac{\text{Fc}_i \cdot \text{hp}_i}{\text{hl}_i} \cdot \frac{\text{Xtt}_i}{\text{hl}_i} \cdot M_1 \cdot M_2 \cdot M_3 \cdots M_7} {1 + \frac{\text{hl}_i \cdot \text{Phi}_i}{\text{hl}_i} \cdot \frac{\text{Rx}_i}{\text{hl}_i} \cdot M_4 \cdot M_5 \cdot M_6 \cdot M_7} \right]} 
\]
\[
M := \begin{pmatrix}
0 \\
0 \\
0 \\
0 \\
0
\end{pmatrix}
\]

Min1 := \text{Minimize(SER, } M) \\
Min1 = 1 \\
\text{SER(Min1) = 1}