Condensation of HFC-134a in an 18° Helix Angle Micro-Finned Tube


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CONDENSATION OF HFC-134a IN AN 18° HELIX ANGLE MICRO-FINNED TUBE

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University of Illinois at Urbana-Champaign, 1995
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ABSTRACT

An experimental study of local condensation heat transfer and pressure drop characteristics of HFC-134a in an 18° helix angle, 0.375" o.d. micro-finned tube was conducted.

The main goal of the study was to compare the heat transfer and pressure drop characteristics during condensation in the micro-finned tube to those in a smooth tube.

In general, the heat transfer enhancement factors decreased as the refrigerant mass flux increased. The average enhancement factors ranged from 2.0 at a mass flux of 110 klbm/ft²-hr, to 1.4 at a mass flux of 330 klbm/ft²-hr. The enhancement factors also depended on the flow regime. In the wavy flow regime, high enhancement factors were usually observed, while in the annular flow regime, low enhancement factors were observed except at very high vapor qualities.

The pressure drop penalty factors showed just a slight dependence on mass flux and vapor quality. In general, the penalty factors increased slightly as the mass flux increased and the vapor quality decreased. The average penalty factors ranged from 1.19 to 1.26.

A simple condenser model that combined the results of this study with pressure drop and heat transfer correlations from earlier studies was developed. The model was used to calculate the total length and refrigerant side pressure drop in a condenser for different operating conditions.
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**SYMBOLS**

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<tr>
<td>A</td>
<td>area</td>
</tr>
<tr>
<td>$A_{\text{cross-sect}}$</td>
<td>cross-sectional area</td>
</tr>
<tr>
<td>$A_{\text{microfin}}$</td>
<td>micro-finned tube refrigerant side surface area</td>
</tr>
<tr>
<td>$A_{\text{rat}}$</td>
<td>area ratio</td>
</tr>
<tr>
<td>$A_s$</td>
<td>refrigerant side surface area</td>
</tr>
<tr>
<td>$A_{\text{smooth}}$</td>
<td>smooth tube refrigerant side surface area</td>
</tr>
<tr>
<td>$c_p$</td>
<td>specific heat at constant pressure</td>
</tr>
<tr>
<td>$c_{p,l}$</td>
<td>specific heat at constant pressure for liquid refrigerant</td>
</tr>
<tr>
<td>$c_{p,v}$</td>
<td>specific heat at constant pressure for vapor refrigerant</td>
</tr>
<tr>
<td>$c_{p,w}$</td>
<td>specific heat at constant pressure for water</td>
</tr>
<tr>
<td>D</td>
<td>diameter</td>
</tr>
<tr>
<td>$D_{\text{eq,flow}}$</td>
<td>equivalent flow diameter</td>
</tr>
<tr>
<td>$D_{\text{T}_{w-r}}$</td>
<td>average tube wall-refrigerant temperature difference</td>
</tr>
<tr>
<td>EF</td>
<td>enhancement factor</td>
</tr>
<tr>
<td>Fr</td>
<td>Froude number</td>
</tr>
<tr>
<td>Fr_l</td>
<td>liquid-only Froude number</td>
</tr>
<tr>
<td>Fr_{so}</td>
<td>Soliman's modified Froude number</td>
</tr>
<tr>
<td>g</td>
<td>acceleration due to gravity</td>
</tr>
<tr>
<td>G</td>
<td>mass flux</td>
</tr>
<tr>
<td>Ga</td>
<td>Galileo number</td>
</tr>
<tr>
<td>h</td>
<td>heat transfer coefficient</td>
</tr>
<tr>
<td>i_{h,i}</td>
<td>enthalpy at heater inlet</td>
</tr>
<tr>
<td>i_l</td>
<td>saturated liquid enthalpy</td>
</tr>
<tr>
<td>i_{lv}</td>
<td>enthalpy of vaporization</td>
</tr>
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</table>

**DEFINITION**

\[
\frac{V^2}{gL}
\]

\[
\frac{(G/p_1)^2}{gD}
\]

Eq. (2.24), (2.25)
ir,i refrigerant enthalpy at test-condenser inlet
ir,o refrigerant enthalpy at test-condenser outlet
iTs,i enthalpy at test-condenser inlet
i.d. inside diameter
Ja liquid Jakob number

cp,l(T_{sat} - T_s) \over i_{iv}

k thermal conductivity
k_l liquid thermal conductivity
k_v vapor thermal conductivity
L length
m mass flow rate
m_r refrigerant mass flow rate
m_w water mass flow rate
Nu Nusselt number

Nu_{forced} forced convective Nusselt number during wavy flow

hD/k_l

o.d. outside diameter
P,p pressure
P_cr critical pressure
P_{red} reduced pressure

P/P_{cr}

PF penalty factor
Pr Prandtl number

\mu c_p/k

Pr_l liquid Prandtl number

\mu c_p,l/k_l

\dot{Q} heat transfer rate
\dot{Q}_h heat input in the heater
\dot{Q}_{l,h} heat loss in the heater
<table>
<thead>
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<td>$\dot{Q}_{l,ts}$</td>
<td>heat loss in the test-condenser</td>
</tr>
<tr>
<td>$\dot{Q}_{\text{microfin}}$</td>
<td>heat transfer rate in the micro-finned tube</td>
</tr>
<tr>
<td>$\dot{Q}_r$</td>
<td>refrigerant side heat transfer rate in the test-condenser</td>
</tr>
<tr>
<td>$\dot{Q}_{\text{smooth}}$</td>
<td>heat transfer rate in the smooth tube</td>
</tr>
<tr>
<td>$\dot{Q}_w$</td>
<td>water side heat transfer rate in the test-condenser</td>
</tr>
<tr>
<td>$R_a$</td>
<td>air-side thermal resistance per unit length</td>
</tr>
<tr>
<td>$R_{\text{ref}}$</td>
<td>thermal resistance per unit length of refrigerant</td>
</tr>
<tr>
<td>$Re$</td>
<td>Reynolds number</td>
</tr>
<tr>
<td>$Re_{L}$</td>
<td>superficial liquid Reynolds number</td>
</tr>
<tr>
<td>$Re_{lo}$</td>
<td>liquid only Reynolds number</td>
</tr>
<tr>
<td>$Re_{vo}$</td>
<td>vapor only Reynolds number</td>
</tr>
<tr>
<td>$T$</td>
<td>temperature</td>
</tr>
<tr>
<td>$T_a$</td>
<td>air temperature</td>
</tr>
<tr>
<td>$T_{h,i}$</td>
<td>temperature at heater inlet</td>
</tr>
<tr>
<td>$T_{\text{sat}}$</td>
<td>saturation temperature</td>
</tr>
<tr>
<td>$T_s$</td>
<td>surface temperature of tube wall in the test-condenser</td>
</tr>
<tr>
<td>$T_{ts,i}$</td>
<td>refrigerant temperature at test-condenser inlet</td>
</tr>
<tr>
<td>$T_{ts,o}$</td>
<td>refrigerant temperature at test-condenser outlet</td>
</tr>
<tr>
<td>$T_{w,i}$</td>
<td>water temperature at test-condenser inlet</td>
</tr>
<tr>
<td>$T_{w,o}$</td>
<td>water temperature at test-condenser outlet</td>
</tr>
<tr>
<td>$U_{\text{A}l,ts}$</td>
<td>overall conductance for test-condenser heat loss</td>
</tr>
<tr>
<td>$V$</td>
<td>velocity</td>
</tr>
<tr>
<td>$x$</td>
<td>vapor quality</td>
</tr>
<tr>
<td>$x_{ts,i}$</td>
<td>refrigerant quality at test-condenser inlet</td>
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</table>
$X_{TT}, X_u$  

Turbulent turbulent Lockhart Martinelli parameter

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

$\alpha$  

Void fraction

$\alpha_i$  

Inlet void fraction

$\alpha_o$  

Outlet void fraction

$\delta$  

Uncertainty

$\Delta P$  

Pressure drop

$\Delta P_{ACC}$  

Acceleration pressure drop

$\Delta P_{FRIC}$  

Frictional pressure drop

$\phi_i^2, \phi_L^2$  

Two-phase pressure drop multiplier

$\mu$  

Dynamic viscosity

$\mu_l$  

Dynamic viscosity of liquid refrigerant

$\mu_v$  

Dynamic viscosity of vapor refrigerant

$\rho$  

Density

$\rho_l$  

Density of liquid refrigerant

$\rho_v$  

Density of vapor refrigerant

$\upsilon$  

Specific volume

$\upsilon_l$  

Liquid specific volume

$\frac{A_{cross-sect, occupied by vapor}}{A_{cross-sect}}$  

Eq. (2.35)
SUBSCRIPTS AND OTHER SYMBOLS

SUBSCRIPTS

cr critical
e based on equivalent diameter
h heater
i inside or inlet
L,l liquid
o outlet
red reduced
sat saturation
ts test-condenser
v vapor
w water

OTHER SYMBOLS

A rate of A
\bar{A} average of A
CHAPTER 1
INTRODUCTION

Since 1989, investigators in projects 01 and 37 of the Air Conditioning and Refrigeration Center (ACRC) at the University of Illinois in Urbana-Champaign have studied the heat transfer and pressure drop characteristics of various refrigerants and mixtures of refrigerants and oils during condensation. Those studies were performed in smooth tubes with internal diameters ranging from 0.124" (3.14 mm) to 0.277" (7.04 mm), and produced the following results:

- A general understanding of the mechanisms that determine heat transfer and pressure drop during condensation in smooth tubes.
- Correlations that predict the heat transfer coefficient and the pressure drop during condensation in smooth tubes.
- A basic understanding of the effects of oil on heat transfer and pressure drop during condensation in smooth tubes.

Regulations that require better energy efficiency combined with an increased interest in more compact heat exchangers have led to the use of internally augmented tubes in evaporators and condensers. This research centers on one type of internally augmented tubes: micro-finned tubes.

The refrigerant chosen for these studies is the hydrofluorocarbon HFC-134a. This refrigerant contains no chlorine, so it is not harmful to the ozone layer. The properties of HFC-134a make it a good replacement for CFC-12.

The heat transfer and the pressure drop characteristics during condensation of HFC-134a inside a micro-finned tube, with a helix angle of 18 degrees, are presented in this report. Those characteristics are essential to the design of condensers using this type of tube. A detailed comparison between these results and results from previous work using
smooth tubes is also presented, which analyzes the conditions under which each type of tube should be used.

A lot of work has been performed on micro-finned tubes in the past. However, very little of that work has studied the local heat transfer and pressure drop characteristics of refrigerants inside micro-finned tubes. Studying the local characteristics is necessary in order to 1) fully understand the mechanisms that determine them and 2) optimize the design of heat exchangers.

Chapter 2 contains a literature review on internal condensation of refrigerants in micro-finned tubes and some information on internal condensation of refrigerants in smooth tubes, which is used later for comparison purposes.

Chapter 3 describes the experimental facility used to conduct the experiments.

Chapter 4 describes the procedures used to obtain, reduce and analyze the experimental data.

Chapter 5 presents the results of the experiments that were performed. A series of figures show the relation between different parameters and the heat transfer and pressure drop characteristics. Another group of figures compares the heat transfer and pressure drop characteristics to those previously obtained for a smooth tube.

Chapter 6 contains the conclusions obtained from the results of these experiments, and recommends areas for future research.
CHAPTER 2
LITERATURE REVIEW

This chapter contains a literature review of publications on internal condensation of refrigerants in micro-finned tubes. It also presents a summary of earlier work conducted in this project on condensation heat transfer by Dobson et al. [1994] and on two-phase frictional pressure drop by Souza et al. [1992, 1993] and Dobson et al. [1993]. This summary is necessary for comparison purposes in later sections of this paper between smooth tubes and micro-finned tubes.

2.1 Internal condensation in micro-finned tubes

During the last 10 to 15 years, considerable research has been conducted on evaporation and condensation inside micro-finned tubes. Even though not as much work has been done on condensation as in evaporation, there are enough publications on the subject to perform a complete literature review and to compare the experimental results. Micro-finned tubes are tubes with a large number of very small fins. A typical number of fins in a 0.375" (9.52 mm) o.d. micro-finned tube is 60. Since the difference between internally finned tubes and micro-finned tubes is basically the number and size of the fins, this review contains information about condensation in internally finned tubes too.

Luu and Bergles [1979] found that for condensation of R-113 in three different finned tubes (with 6, 16 and 32 fins respectively), the one with the highest number of fins exhibited the largest increase in heat transfer coefficient with respect to a smooth tube, at low flow rates. At high flow rates the heat transfer coefficients of the three tubes were similar. For the tube with the highest increase in heat transfer coefficient, the increase was about 120% with respect to the smooth tube, on a nominal area basis. The nominal area of a finned tube is equal to the area of a smooth tube with a diameter equal to the maximum inside diameter of the finned tube. For the three tubes, the increases in heat transfer
coefficients were larger than the increases in surface area. In most cases, the increases in pressure drop were modest. The quality change in the test condenser was around 100%. Wall temperature measurements were performed in order to calculate the average heat transfer coefficients.

Luu and Bergles [1980] achieved some success in predicting their data (Luu and Bergles, [1979]) with modified smooth tube correlations for heat transfer coefficients and pressure drops.

Said and Azer [1983] tested four different internally finned tubes and a smooth tube and compared the results. One of the finned tubes had 10 straight fins and the other three finned tubes had 16 fins, but with different helix angles, fin heights, etc. The refrigerant used was R-113. The highest heat transfer coefficient enhancement obtained was 51% on a nominal area basis with respect to the smooth tube. The heat transfer coefficient decreased as the vapor quality decreased. Heat transfer and pressure drop correlations were proposed, which are good predictors of the results of this study and many of the results of other studies. Wall temperature measurements were performed in order to calculate the heat transfer coefficients.

Venkatesh and Azer [1985] studied the condensation characteristics of R-11 inside four internally finned tubes and one smooth tube. The highest heat transfer coefficient enhancement found was 55% on a nominal area basis.

Khanpara [1986] conducted a detailed study of heat transfer and pressure drop characteristics of R-113 and R-22 in micro-finned tubes with different geometries. Local heat transfer coefficients were determined. The study found an increase in the heat transfer coefficient and the pressure drop as the mass flux and quality were increased. Heat transfer increases of up to 283% and pressure drop increases of up to 100% were found with respect to a smooth tube. The tubes with the largest fin height, 0.007" (0.18 mm), seemed to provide the best performance. Also, a helix angle in the 10 to 20 degree range and a number of fins around 60 appeared to produce the best results.
Schlager et al. [1987] published a survey of refrigerant heat transfer and pressure drop emphasizing oil effects and internal augmentation. It included evaporation and condensation. More specifically, it included publications on different augmentation techniques for internal condensation, such as rough surface, internal fins, mixer inserts and twisted tape and three publications on spiral micro-fins.

Schlager [1988] studied the effects of oil on heat transfer and pressure drop during condensation of R-22 inside a micro-finned tube which had 60 fins, a helix angle of 18 degrees, an o.d. of 0.375'' (9.52 mm) and a fin height of about 0.008'' (0.2 mm). For pure refrigerant, enhancement factors (see also Schlager et al.[1988a, 1988b, 1989a, 1989b]) between 1.9 (at the highest mass flux) and 2.4 (at the lowest mass flux) with respect to a smooth tube were found. The ratio of pressure drop in the micro-finned tube to the pressure drop in the smooth tube varied between 1.0 and 1.8, for pure refrigerants. The enhancement factors were larger than the pressure drop ratios for every mass flux. Average (not local) heat transfer coefficients were calculated, using a modified Wilson plot technique.

Kaushik and Azer [1988] proposed two heat transfer correlations for condensation inside internally finned tubes. They were tested against steam, R-113 and R-11 condensation data. Results of various investigators were compared with the values predicted by the correlations, and as a result, 65% of the data points (761 total data points) fell within ±30% of the predicted values. The final forms of the correlations are:

\[
\frac{\text{Nu}}{\text{Pr}^{1/3}} = 2.078 \text{Re}^{0.507} \left( \frac{\Delta x D_i}{L} \right)^{0.198} \left( \frac{p_i}{p_{cr}} \right)^{-0.140} F_1^{0.874} F_2^{-0.814} \quad \text{for } F_1 < 1.4 \quad (2.1)
\]

\[
\frac{\text{Nu}}{\text{Pr}^{1/3}} = 0.391 \text{Re}^{0.507} \left( \frac{\Delta x D_i}{L} \right)^{0.198} \left( \frac{p_i}{p_{cr}} \right)^{-0.140} F_1^{4.742} \quad \text{for } F_1 > 1.4 \quad (2.2)
\]

where
\[ Nu = \frac{h_i D_i}{k_1} \]  
(2.3)

\[ Re_e = \frac{G_e D_i}{\mu_1} \]  
(2.4)

\[ G_e = G[(1 - \bar{x}) + \bar{x}(\rho_1 / \rho_v)^{0.5}] \]  
(2.5)

\[ F_1 = A_{fa} / A_{fc} \]  
(2.6)

\[ F_2 = A_n / A_a \]  
(2.7)

where \( A_{fa} \) is the actual flow area, \( A_{fc} \) is the open core flow area, which is defined as the cross sectional area of a smooth tube with a diameter equal to the minimum inside diameter of the finned tube, \( A_n \) is the nominal heat transfer area and \( A_a \) is the actual heat transfer area (see Kaushik and Azer [1988]).

\( F_1 \) and \( F_2 \) are geometric parameters which include the helix angle effect. For smooth tubes, \( F_1 \) and \( F_2 \) are both equal to 1.

Schlager et al. [1989c] tested R-22 in three 0.375" (9.52 mm) o.d., 0.35" (8.9 mm) maximum i.d. micro-finned tubes with different internal geometries. For condensation, the study found enhancement factors ranging from 1.6 to 1.8 at 150 klbf/ft²-hr (200 kg/m²-s) and decreasing to the 1.4 - 1.6 range at 365 klbf/ft²-hr (500 kg/m²-s). The penalty factors ranged from about 1.0 to about 1.2.

Schlager et al. [1990a] published a literature review of performance predictions of refrigerant-oil mixtures in smooth and internally finned tubes. For internal condensation of pure refrigerants in finned tubes it presents four heat transfer correlations, the most recent of which was published by Kaushik and Azer [1988]. It also presents and mentions some
correlations used to predict the pressure drop of pure refrigerant in finned tubes during evaporation and condensation.

Schlager et al. [1990b] proposed equations to predict the performance of refrigerant-oil mixtures in smooth and internally finned tubes. The heat transfer correlation for condensation of pure refrigerants in finned tubes is based on experimental results from the same study. The o.d. of the micro-finned tube tested was 0.375" (9.52 mm), the condensation temperature 105 °F (41 °C), the inlet quality 85% and the outlet quality 15%. The proposed correlation calculates the enhancement factor as a function of mass flux. The enhancement factor \( EF_{a/s} \) is defined in this paper as the ratio of heat transfer in the micro-finned tube to the heat transfer in the smooth tube. It is also defined as the ratio of the heat transfer coefficient in the micro-finned tube to the heat transfer coefficient in the smooth tube, basing the heat transfer coefficient in the micro-finned tube on an inside heat transfer area equal to the area of a smooth tube with an inside diameter equal to the maximum diameter of the micro-finned tube. See also Schlager et al.[1988a, 1988b, 1989a, 1989b]. The final form of the correlation is:

\[
\frac{EF_1}{EF_2} = \left( \frac{G_1}{G_2} \right)^{0.21}
\]

(2.8)

For pressure drop in micro-finned tubes, no correlation was formulated, but a pressure drop penalty factor of 1.7 was determined for the micro-finned tube with respect to the smooth tube. The pressure drop penalty factor is defined as the ratio of pressure drop in the micro-finned tube to the pressure drop in the smooth tube. Agreement of all the equations with the experimental data was within ±20%.

Kaushik and Azer [1990] formulated a general pressure drop correlation for internal condensation in finned tubes. It predicted most of the authors' data points and data points of other investigators (68% of the data points were predicted to within ±40%). It was tested against steam and R-113 condensation data. The correlation is the following:
\[ \Delta p_f = \Delta p_s \left( \frac{A_{fa}}{A_{fc}} \right)^{3.72} \quad \text{for} \quad \left( \frac{A_{fa}}{A_{fc}} \right) > 1.4 \]  

(2.9)

\[ \Delta p_f = \Delta p_s \left( \frac{A_{fa}}{A_{fc}} \right)^{10.2} \left( \frac{A_{fa}}{A_{fn}} \right)^{-1.7} \quad \text{for} \quad \left( \frac{A_{fa}}{A_{fc}} \right) < 1.4 \]  

(2.10)

where \( \Delta p_f \) is the pressure drop in the finned tube, \( \Delta p_s \) is the pressure drop for a smooth tube with an inside diameter equal to the nominal inside diameter of the finned tube, \( A_{fa} \) is the actual flow area, \( A_{fc} \) is the open core area and \( A_{fn} \) is the nominal flow area (see Kaushik and Azer [1990]).

Schlager et al. [1990] studied the condensation of R-22 in three different microfinned tubes with a 0.5" (12.7 mm) o.d., 0.46" (11.7 mm) maximum i.d., 60 or 70 fins with heights ranging from 0.0059" (0.15 mm) to 0.0118" (0.3 mm) and helix angles between 15 and 25 degrees. A smooth tube was also tested for comparison purposes. The test conditions were the following: condensing temperature from 102 °F to 108 °F (39 °C - 42 °C), mass flux from 73 klb/m-ft²-hr (100 kg/m²-s) to 293 klb/m-ft²-hr (400 kg/m²-s), inlet quality 85% and outlet quality 10%. The average heat transfer coefficients in the micro-finned tubes, based on a nominal equivalent smooth tube area, were 1.5 to 2 times larger than those in the smooth tube. More specifically, at the low mass fluxes the enhancement factor had a value between 1.8 and 2.0 and it dropped to around 1.5 to 1.6 for the high mass fluxes. The trends and values for the heat transfer coefficients and enhancement factors were similar for the three micro-finned tubes. However, the tube with an 18 degree helix angle consistently produced the highest heat transfer coefficients and enhancement factors, the tube with a 15 degree helix angle consistently produced the second highest values and the tube with a 25 degree helix angle produced the lowest values of the three micro-finned tubes. An interesting geometric characteristic of the micro-finned tubes is that the tube with the best performance also had the largest fin height, and the tube
with the worst performance had the smallest fin height. Wilson plots were used to
determine the refrigerant side heat transfer coefficients. The pressure drop also increased,
but the increase was smaller than the increase in heat transfer coefficient. The penalty
factors (ratio of pressure drops in the micro-finned tubes to the smooth tubes) were
approximately 1.3 ± 0.1 for all three tubes, with an apparent tendency to fall with
increasing mass flux.

Eckels and Pate [1991] presented the results of experimental work concerning the
condensation characteristics of HFC-134a and CFC-12 in a smooth tube and in a micro-
finned tube. The micro-finned tube had a helix angle of 17 degrees, 60 fins with a height
of 0.008" (0.2 mm), an o.d. of 0.375" (9.52 mm) and a maximum i.d. of 0.314" (8.72
mm). The area ratio of the micro-finned tube, defined as the ratio of the inside heat transfer
area of the micro-finned tube to the inside heat transfer area of a smooth tube having a
diameter equal to the maximum inside diameter of the micro-finned tube, was 1.5. The
smooth tube had an o.d. of 0.375" (9.52 mm) and an i.d. of 0.314" (8.0 mm). The
condensing temperature varied between 86 °F and 122 °F (30 °C - 50 °C). The inlet
qualities ranged from 80% to 88% while the outlet qualities ranged from 5% to 13%. The
mass flux ranged from 96 klb m²/ft²-hr to 294 klb m²/ft²-hr (130 kg/m²-s to 400 kg/m²-s).
For condensation of HFC-134a, the heat transfer coefficients in the micro-finned tube were
between 70% (at the highest mass flux) and 110% (at the lowest mass flux) higher than
the heat transfer coefficients in the smooth tube. The enhancement factors ranged between
1.75 and 2.5 at the highest and lowest mass fluxes, respectively. In the micro-finned tube,
the heat transfer coefficients increased by as much as 30% when the condensing
temperature was reduced from 122 °F (50 °C) to 86 °F (30 °C). The pressure drops were
also higher in the micro-finned tube than in the smooth tube during condensation of HFC-
134a. For example, at a mid-flow range for 104 °F (40 °C) condensing temperature, the
penalty factor was about 1.5. For most of the experimental conditions, the penalty factors
were lower than the enhancement factors, but large uncertainties in the pressure drops at
low mass fluxes do not allow any general conclusions. For CFC-12, the trends were similar to those of HFC-134a. For example, the enhancement factors ranged from 1.7 (all condensing temperatures, highest mass flux) to 2.3 (86 °F condensing temperature, lowest mass flux). One of the few significant differences between the performance of HFC-134a and CFC-12 were observed at the lower mass fluxes, where the penalty factor was significantly higher for HFC-134a than for CFC-12. Again, the penalty factors during condensation of CFC-12 were lower than the enhancement factors.

Koops [1992] and Koops and Azer [1993] also studied the condensation characteristics of HFC-134a and CFC-12 in smooth and internally finned tubes. Two sets of tubes were tested. Each set consisted of one smooth tube and one internally finned tube, both with the same o.d.. The outside diameters of the tubes were 0.75" (19.05 mm) and 0.625" (15.88 mm) and the finned tubes had 38 and 30 fins respectively. The condensation temperatures were 95 °F and 104 °F (35 °C and 40 °C), the inlet superheat less than 7.2 °F (4 °C) and the exit subcooling less than 10.8 °F (6 °C). The mass flux ranged from 18 klb_m/ft^2-hr to 165 klb_m/ft^2-hr (25 kg/m^2-s to 225 kg/m^2-s). Wall temperature measurements were performed in order to determine the heat transfer coefficients. The results showed that HFC-134a heat transfer coefficients were 18% to 28% higher than those of CFC-12 for both the smooth and finned tubes. For both refrigerants, the enhancement factors for the finned tubes were between 1.5 and 1.75 for the smaller diameter tube, and between 2.0 and 2.3 for the larger diameter tube. HFC-134a gave slightly higher pressure drops than CFC-12 in the finned tubes, but the smooth tubes pressure drop data were inconclusive due to the low pressure drops obtained. The pressure drop results were predicted with a correlation formulated by Kaushik and Azer [1990]. As a result, 64% of the data points were predicted to within ±40% for pressure drops larger than 0.02 psi (0.134 kPa). A correlation by Kaushik and Azer [1988] was used to predict the heat transfer coefficients, but it was not a good predictor of the experimental results. So using the experimental results from this study, new constants were determined for the
same correlation. As a result, 90% of the data points were predicted to within ±20% and 100% of the data points were predicted to within ±30%. The new correlation was (see Kaushik and Azer [1988]):

\[ h = 317.95 \frac{k}{D_i} Pr^{1/3} Re_e^{0.357} \left( \frac{\Delta x D_i}{L} \right)^{0.954} Pr_{red}^{-0.737} F_1^{-0.442} F_2^{-1.116} \]  

(2.11)

5,000 < Re_e < 80,000

Chiang [1993] investigated the heat transfer and pressure drop of R-22 during condensation in axial and helical micro-finned tubes. Four types of micro-finned tube were tested. The first tube had an o.d. of 0.4" (10 mm), a helix angle of 0 degrees (axial fins) and 72 fins. The second tube had an o.d. of 0.4" (10 mm), a helix angle of 18 degrees and 60 fins. The third tube had an o.d. of 0.3" (7.5 mm), a helix angle of 0 degrees and 60 fins. The fourth tube had an o.d. of 0.3" (7.5 mm), a helix angle of 18 degrees and 43 fins. The condensing temperature was about 105 °F (40.5 °C) and the mass flux ranged from 198 klb_m/ft²-hr to 807 klb_m/ft²-hr (270 kg/m²-s to 1100 kg/m²-s). A modified Wilson plot technique was used in order to determine the heat transfer coefficients. The average condensing heat transfer coefficients in the axial grooved tubes were 10% to 20% higher than the heat transfer coefficients in the helical grooved tubes of equal o.d., on a nominal area basis. The pressure drop in the 0.4" (10 mm) o.d. helical grooved tube was approximately 15% higher than the pressure drop in the 0.4" (10 mm) o.d. axial grooved tube. No pressure drop data for the 0.3" (7.5 mm) o.d. tubes were reported. Quasi-local heat transfer coefficients were also studied. As a result, an almost linear increase in the heat transfer coefficient with quality increase was observed.
2.2 Internal condensation in smooth tubes

2.2.1 Heat transfer

In recent work performed on this project by Dobson et al. [1994], two heat transfer correlations were developed: a wavy flow correlation and an annular flow correlation. Figure 2.1 shows the most typical two-phase flow regimes observed in condensation. The wavy regime is a gravity driven regime, and the mechanism of heat transfer in it is primarily conduction through the liquid film at the top of the tube, but there is also heat transfer in the pool at the bottom of the tube by forced convective condensation. Because of this, the wavy flow correlation takes into account filmwise condensation at the top of the tube and forced convective condensation at the bottom of the tube. The wavy flow correlation predicts the experimental data of this study very well, with a mean deviation of 6.6%. The annular regime is a shear dominated regime, where the interfacial shear stresses dominate the gravitational forces, producing an almost symmetric annular film in the tube. In this regime, forced convective condensation is the main mechanism of heat transfer. The annular flow correlation is a two-phase multiplier correlation. This type of correlation is based on a single-phase correlation and is corrected for two-phase flow. There are other approaches to predicting heat transfer in annular flow, such as shear-based approaches and boundary layer approaches, which are more theoretical and complicated. The annular flow correlation developed in this study predicted the experimental data very accurately, with a mean deviation of 4.5%, 67% of the data points were predicted within ±5% and 96% of the data points were predicted within ±15%. The final form of the annular flow correlation is the following:

\[
\text{Nu} = 0.023 \text{Re}^{0.8} \text{Pr}^{0.4} \left[ 1 + \frac{2.22}{X^{0.889}} \right]
\]  

(2.12)
where

\[
X_u = \left( \frac{\rho_v}{\rho_l} \right)^{0.5} \left( \frac{\mu_l}{\mu_v} \right)^{0.1} \left( \frac{1-x}{x} \right)^{0.9}
\]  

(2.13)

\(X_{tt}\) is the turbulent-turbulent Lockhart-Martinelli parameter (Lockhart and Martinelli [1947]). The density and viscosity ratios have been correlated as a function of the reduced pressure for a number of refrigerants (Wattelet et al. [1994]). The correlation is the following:

\[
\left( \frac{\rho_v}{\rho_l} \right)^{0.5} \left( \frac{\mu_l}{\mu_v} \right)^{0.1} = 0.551(P_{red})^{0.492}
\]  

(2.13b)

It is obvious that Eq. (2.12) is based on Dittus-Boelter's single-phase correlation (Incroprera and DeWitt [1990]), which has the following form:

\[
Nu = 0.023Re_i^{0.8}Pr_i^{0.4}
\]  

(2.14)

The other term in Eq. (2.12) is the two-phase multiplier.

As mentioned previously, the wavy flow correlation takes into account filmwise condensation at the top of the tube and forced convective condensation at the bottom of the tube. In the final form of the wavy correlation, which is presented next, the first term is the Nusselt number of the liquid film at the top of the tube, and the second term accounts for the forced convective condensation at the bottom of the tube:

\[
Nu = \frac{0.23Re_{vo}^{0.12}}{1 + 1.11X_u^{0.58}} \left[ \frac{Ga Pr}{Ja} \right]^{0.25} + (1 - \theta_1/\pi)Nu_{forced}
\]  

(2.15)
where

\[ \theta_1 = \text{angle subtended from the top of the tube to the liquid level} \]

\[ \text{Nu}_{\text{forced}} = 0.0195 \text{Re}_l^{0.8} \text{Pr}_l^{0.4} \phi_1(X_{lt}) \]  \hspace{1cm} (2.16)

where

\[ \phi_1(X_{lt}) = \sqrt{1 + \frac{c_1}{X_{lt}^{c_2}}} \]  \hspace{1cm} (2.17)

For 0 < Fr_l ≤ 0.7:

\[ c_1 = 4.172 + 5.48 F r_l - 1.564 F r_l^2 \]  \hspace{1cm} (2.18)

\[ c_2 = 1.773 - 0.169 F r_l \]  \hspace{1cm} (2.19)

For Fr_l > 0.7:

\[ c_1 = 7.242 \]  \hspace{1cm} (2.20)

\[ c_2 = 1.655 \]  \hspace{1cm} (2.21)

The liquid level angle, \( \theta_1 \), can be related to the void fraction by the following equation, if the area occupied by the condensate film is neglected:
\[
\alpha = \frac{\theta_1}{\pi} - \frac{\sin(2\theta_1)}{2\pi} \quad (2.22)
\]

A good approximation to Eq. (2.22), formulated by Jaster and Kosky [1976], which is much easier to use is:

\[
1 - \frac{\theta_1}{\pi} \approx \frac{\arccos(2\alpha - 1)}{\pi} \quad (2.23)
\]

Dobson et al. [1994] also recommended the conditions under which each of the two correlations (Eq. (2.12) and Eq. (2.15)) should be used. The parameter used to predict the flow regime is the Froude number, as defined by Soliman [1982]:

\[
Fr_{so} = 0.025Re_{1}^{1.59} \left(\frac{1 + 1.09X_{u}^{0.038}}{X_{u}}\right)^{1.5} \frac{1}{Ga^{0.5}} \text{ for } Re_{1} \leq 1250 \quad (2.24)
\]

\[
Fr_{so} = 1.26Re_{1}^{1.04} \left(\frac{1 + 1.09X_{u}^{0.038}}{X_{u}}\right)^{1.5} \frac{1}{Ga^{0.5}} \text{ for } Re_{1} > 1250 \quad (2.25)
\]

Eq. (2.12) and Eq. (2.15) should be used according to the following recommendations:

For \( G \geq 365 \text{ kbl}_m/\text{ft}^2\cdot\text{hr} \) (500 kg/m\(^2\)-s):

Use Eq. (2.12)

For \( G < 365 \text{ kbl}_m/\text{ft}^2\cdot\text{hr} \) (500 kg/m\(^2\)-s):

Use Eq. (2.15) if \( Fr_{so} < 20 \)

Use Eq. (2.12) if \( Fr_{so} > 20 \)
2.2.2 Pressure drop

Souza et al. [1992, 1993] developed a frictional pressure drop correlation for two-phase flow. This correlation uses the method proposed by Lockhart and Martinelli [1947] which establishes that the two-phase pressure drop is equal to the pressure drop that would be experienced by the liquid or vapor, multiplied by a two-phase multiplier ($\phi^2$). Using the liquid phase as the base, the correlation looks like this:

$$\Delta P_{\text{fric}} = \Delta P_L \phi_L^2$$

(2.26)

where $\Delta P_{\text{fric}}$ is the frictional pressure drop and $\Delta P_L$ is the pressure drop that would be obtained for the liquid phase only, which is defined as:

$$\Delta P_L = \frac{2f_L G^2 (1-x)^2 L}{\rho_L D}$$

(2.27)

where

$$f_L = \frac{0.079}{Re_L^{0.25}}$$

(2.28)

The two-phase multiplier is defined (as in the wavy flow correlation):

$$\phi_L^2 = (1.376 + \frac{c_1}{X_{\text{TT}}^2})$$

(2.29)

The constants in the expression for the turbulent-turbulent Lockhart-Martinelli parameter are slightly different from the ones in Eq. (2.13). In this case, the Lockhart-Martinelli parameter is defined:
Finally, the constants $c_1$ and $c_2$ are defined in the same way as in the wavy flow correlation:

For $0 < Fr_L \leq 0.7$:

$$c_1 = 4.172 + 5.48 Fr_L - 1.564 Fr_L^2$$  \hspace{1cm} (2.31)$$

$$c_2 = 1.773 - 0.169 Fr_L$$  \hspace{1cm} (2.32)$$

For $Fr_L > 0.7$:

$$c_1 = 7.242$$  \hspace{1cm} (2.33)$$

$$c_2 = 1.655$$  \hspace{1cm} (2.34)$$

This correlation has proven to be a good predictor of two-phase pressure drop. In fact, Dobson et al. [1994] found that this correlation consistently predicted the data in that study with a mean deviation of less than 20%.

There are two more causes of pressure drop inside a tube: acceleration and gravity. For single tube, horizontal heat exchangers, the gravitational component of the pressure drop is zero. The acceleration component of the pressure drop ($\Delta P_{\text{ACC}}$) is defined:

$$\Delta P_{\text{ACC}} = \frac{16 \pi^2 \rho}{\rho} \left\{ \frac{x^2}{\rho \alpha_o} + \frac{(1-x_o)^2}{\rho (1-\alpha_o)} \right\} - \left\{ \frac{x_i^2}{\rho \alpha_i} + \frac{(1-x_i)^2}{\rho (1-\alpha_i)} \right\}$$  \hspace{1cm} (2.35)$$
where the subscript $i$ denotes inlet while the subscript $o$ denotes outlet.

The void fraction ($\alpha$), can be calculated using the correlation by Zivi [1964]:

$$\alpha = \frac{1}{1 + \left(\frac{1-x}{x}\right)\left(\frac{\rho_v}{\rho_L}\right)^{0.87}}$$

(2.36)
Figure 2.1 Typical flow regimes in condensation processes (from Dobson et al. [1994]).
CHAPTER 3
EXPERIMENTAL FACILITY

This chapter describes the experimental facility used to obtain the condensation data for this study. Earlier versions of this facility have been described by Hinde et al. [1992], Dobson et al. [1994], Gaibel et al. [1994] and Kenney et al. [1994]. The facility version used by Gaibel et al [1994] and Kenney et al. [1994] is very similar to the present facility, so this chapter describes the general characteristics of the experimental apparatus emphasizing the recent modifications.

The purpose of the experimental apparatus is to provide refrigerant at different conditions to the test-condenser, where the heat transfer coefficients and the pressure drops of the refrigerant are measured.

3.1 Experimental apparatus

The experimental apparatus is composed of a refrigerant loop, a water loop and the data acquisition system. Sections 3.1.1, 3.1.2 and 3.1.3 describe these three components, as well as the instrumentation used in the two loops. The test-condenser is part of both the refrigerant and the water loops. However, it is described in the section corresponding to the refrigerant loop (Section 3.1.1).

3.1.1 Refrigerant loop

A schematic of the refrigerant loop is shown in Fig. 3.1. Most of the refrigerant loop, including the test-condenser, is insulated with armaflex insulation. The pump drives the refrigerant through the loop. The pump is a MicroPump™ three-gear, variable speed, 0.77 gpm (0.049 l/s) pump. It is driven by a 1/3 hp (0.25 kW), three-phase motor and the speed of the motor is controlled with an AC inverter. It requires no lubrication, so experiments with pure refrigerants can be conducted. Varying the pump speed is the most
accurate way of controlling the flow rate around the refrigerant loop. The other way of varying the flow rate, particularly at low values, is by sending some refrigerant around the pump in the pump bypass. This flow is controlled with a needle valve in the bypass line.

After the pump, the refrigerant flow rate is measured in one of two flow meters, depending on the flow rate. Flow rates of 2 lb/min (0.9 kg/min) or less are measured in a Micro Motion D6™ mass flow meter, with a range of 0-2 lb/min (0-0.9 kg/min). This flow meter has a 0-10 V output and an uncertainty of ±0.1%. Higher flow rates are measured in a Max Machinery™ positive displacement flow meter, with a range of 0-1 gpm (0-3.8 l/min), a 0-10 V output and an uncertainty of ±0.31%. The two flow meters are connected in parallel.

Following the flow meters, the refrigerant passes through the refrigerant heater. The heater is used to heat the subcooled liquid refrigerant to the desired conditions at the inlet of the test-condenser. Those conditions are usually a particular vapor quality in the case of two-phase tests, or a particular temperature in the case of single-phase tests. The heater consists of five 5.91 ft (1.8 m) long passes of 0.375" (9.52 mm) o.d. copper tubing. Each pass is wrapped with four 180 Ω resistance heater tapes. The total power of the heater is 21,840 Btu/hr (6.4 kW). The heaters are wrapped on the surface with shrink tape and insulated with armaflex insulation. The first nine heaters are always turned on, and are powered by a 0 to 240 V variable voltage transformer (Variac), which allows adjustment of the power output from 0 to 9827 Btu/hr (0-2.88 kW). The rest of the heaters are controlled with on/off switches and are used only in cases when the heat requirements are higher than the maximum output of the first nine heaters. A schematic of the refrigerant heater is shown in Fig. 3.2. The power input into the refrigerant heater is measured with two Ohio Semitronics™ watt-hour transducers (one for the first nine heaters and one for the rest). Their uncertainty is 0.2%.

After exiting the heater, the refrigerant flows through a long section of insulated (adiabatic) tubing to obtain fully developed flow before reaching the test-condenser.
The refrigerant flows through a sight glass next. The sight glasses serve two purposes. First, they are used to observe the refrigerant flow patterns, regimes, etc. Second, they serve as a safety device. They allow confirmation that normal flow is circulating around the loop, and that the data acquisition system is showing the right data. The sight glasses are 5" (12.7 cm) long with an o.d. of 0.5" (12.7 mm) and an i.d. of 0.277" (7.04 mm), which was the closest available size to the test-condenser i.d.. The sight glasses are annealed and can withstand a pressure of up to 500 psi (3450 kPa).

Following the sight glass, the refrigerant passes through an adiabatic section. It serves the same function as the section of tubing at the outlet of the refrigerant heater, but two differential pressure transducers connected in parallel across it measure the adiabatic pressure drop. One transducer is a Sensotec™ 0-5 psi (0-35 kPa), with a 0-5 V output and an estimated uncertainty of ±0.07 psi (±0.5 kPa) (Dobson et al. [1994]). The other transducer is a Setra™ 0-1 psi (0-7 kPa) with a 4-20 mA output, a 24 VDC supply and an estimated uncertainty of ±0.03 psi (±0.2 kPa).

Next, the refrigerant flows through the test-condenser. Here is where the heat transfer and pressure drop data are obtained. The test-condenser is a counter-flow heat exchanger, with refrigerant flowing in the inner tube, and water flowing in the outer annulus. Fig. 3.3 is a schematic of the test-condenser and the adiabatic section.

The water annulus is part of the water loop, which will be described later. This annulus is made of transparent plastic, and it has an o.d. of 0.75" (19 mm) and an i.d. of 0.625" (16 mm). Nylon washers with an inside diameter equal to the outside diameter of the inner tube, and with an outside diameter equal to the inside diameter of the water annulus, hold together the water annulus and the inner tube. Small holes are drilled in the washers to allow the water to flow through them and to mix the water to avoid temperature stratification. The inlet and outlet temperatures of the water in the test-condenser are measured with type-T thermocouple probes that expose the thermocouple beads directly to the water. The error in the temperature readings of these thermocouples is less than 0.2 °F.
(0.1 °C). A detailed description of the construction process of these probes is presented in Dobson et al. [1994]. The water flow through the test-condenser is controlled by a needle valve in a rotameter which is located upstream of the test-condenser. However, the water flow rate is not measured with the rotameter, but by collecting a timed sample in a cylinder downstream of the test-condenser, and weighing the cylinder before and after the sample is taken. The uncertainty in the water flow rate measurement was estimated at less than 1.5% by Dobson et al. [1994]. Pressure ranging from 10 psig to 20 psig (70 to 140 kPa) is maintained in the water annulus with a needle valve located downstream of the test section, to prevent the formation of air bubbles that could affect temperature readings and the heat transfer from the refrigerant to the water.

The water annulus used for these experiments is very similar to the ones used in earlier versions of the experimental apparatus. There is, however, one important improvement in its construction. While in earlier versions copper tees were inserted in the annulus to allow the thermocouples to be taken out of the test-condenser (to connect them to the data acquisition system), in this version small holes were drilled in the plastic annulus for that purpose. The holes were later sealed with glue. This saved construction time, reduced the amount of materials required for the construction of the test-condenser (particularly glue), and reduced the number of water leaks.

The inner tube of the test-condenser is the component in which most of the main differences between this apparatus and earlier ones exist. In earlier studies the inner tubes were smooth tubes, while in this study the inner tube is a 0.375" (9.52 mm) o.d. micro-finned tube. The main two physical differences between the smooth and the micro-finned tubes are the texture of the inside surface and the thickness of the tube wall. The inside surface of the smooth tube is smooth, while the inside surface of the micro-finned tube has many small fins. The wall thickness of the smooth tube is much larger than that of the micro-finned tube. For example, the wall thickness of a 0.375" (9.52 mm) o.d. smooth tube used in earlier studies was around 0.05" (1.3 mm), while the wall thickness of the
micro-finned tube used in the present study is 0.012" (0.3 mm). This difference in wall thickness creates the necessity of changing the method used to mount the wall thermocouples on the tube. Those differences will be explained later in this chapter.

The micro-finned tube is manufactured by Modine\textsuperscript{TM}, it is made of copper, it has an o.d. of 0.375" (9.52 mm), a maximum i.d. of 0.351" (8.91 mm) and a minimum i.d. of 0.336" (8.53 mm). It has 60 fins with a height of 0.007" (0.18 mm) and a helix angle of 18 degrees. A schematic of the micro-finned tube is shown in Fig. 3.4.

The absolute pressure at the inlet of the test-condenser and the pressure drop across it are measured (on the refrigerant side). The absolute pressure at the inlet of the test-condenser is measured with two BECT\textsuperscript{TM} pressure transducers. The first transducer has a 0-500 psi (0-3445 kPa) range, an output of 0-5 V and a 24 VDC supply. The second transducer has a 0-300 (0-2100 kPa) psi range, an output of 4-20 mA and a 115 VAC supply. The uncertainty for both of these transducers was estimated by Dobson et al. [1994] at ±1 psi (±7 kPa). The pressure drop across the test condenser is measured with a Sensotec\textsuperscript{TM} 0-5 psi (0-35 kPa) differential pressure transducer. Its output is 0-5 V and its uncertainty is ±0.07 psi (±0.48 kPa). The pressure measurements are made through pressure taps installed at the inlet and outlet of the adiabatic section and the test condenser (see Fig. 3.3). A detailed description of the characteristics and construction of these pressure taps is presented in Gaibel et al. [1994].

To measure the wall temperature of the micro-finned tube, type-T, copper-constantan thermocouples were soldered to the outside of the tube (see Fig. 3.3). As in the latest version of the test-condenser used before the present one, six thermocouples were soldered to the tube outside the water annulus (three stations with one thermocouple at the top and one thermocouple at the bottom of the tube). The stations located outside the water annulus are located at the inlet of the adiabatic section, at the inlet of the test-condenser and at the outlet of the test condenser. The distribution of the thermocouples soldered to the tube inside the water annulus is also similar to that of the previous versions: five stations
with four thermocouples each, located at the following circumferential locations: 0°, 60°, 180° and 240° from the top. For more information on the location and installation of the thermocouples see Dobson et al. [1994] and Gaibel et al. [1994].

However, the thermocouples had to be mounted on the tube using a different technique from the one previously used. As mentioned earlier, the smooth tubes have much thicker walls than the micro-finned tubes. In earlier test-condensers, which used smooth tubes, the thermocouples were installed in grooves made in the tube itself (Gaibel et al. [1994] provide a detailed description of the thermocouple mounting technique), because the walls of the tubes were thicker than the thermocouples, assuring that the thermocouples would not stick out in the boundary layer of the water, which would affect the temperature readings. This would be impossible to do with the micro-finned tube used in this study, because its wall thickness is 0.012" (0.3 mm) while the diameter of the thermocouples is 0.020" (0.5 mm).

Polaski [1993] studied different thermocouple mounting techniques, and the accuracy of their wall temperature readings. The method that produced the best results was one that simulated the grooves cut in the tubes, used in earlier versions of the test-condenser, by soldering two pieces of copper shim stock on the tube, leaving a groove between them, where the thermocouple was placed and soldered to the tube.

An attempt was made of building a test-condenser using that technique. However, it was very difficult to obtain good contact between the tube and the shim stock when using a shim stock thicker than the thermocouple diameter. So another technique, which was based on this technique was tried. Instead of shim stock, a copper coupling with an inside diameter equal to the outside diameter of the tube was used. Grooves were cut in the coupling, which was later soldered to the tube. Then, the thermocouples were placed in the grooves and soldered to the tube. This mounting technique is illustrated in Fig. 3.5.

Before building the test-condenser, this technique was tried in a water-to-water heat exchanger built specifically for this purpose. In the heat exchanger, one thermocouple was
soldered in a groove cut in the tube, while another was soldered using the copper coupling technique in the opposite side of the tube, but in the same axial location. Hot water flowed in the inner tube and cold water flowed in the outer annulus. The temperature difference between the hot water and the cold water was around 22 °F (12 °C). The heat exchanger was operated at different flow rates and temperatures, but in every case, the difference between the wall temperature measurements of the two thermocouples was 1% or less of the temperature difference between the hot water and the cold water. So in effect, using this mounting technique for the wall thermocouples does not increase the uncertainty in the wall temperature measurements with respect to the groove technique.

The rest of the mounting procedure of the wall thermocouples is similar to the procedure used in earlier test-condensers, which, as mentioned earlier, is described in detail in Gaibel et al. [1994].

After exiting the test-condenser, the refrigerant flows through another sight glass, identical to the one described earlier. Following the sight glass at the exit of the test-condenser, the refrigerant passes through an after-condenser. The after-condenser is a Refrigeration Research™ water cooled heat exchanger with a 24,000 Btu/hr (7 kW) capacity. The purpose of the after-condenser is to return the refrigerant to a subcooled liquid state. It is necessary to subcool the refrigerant for two reasons: the pump cannot effectively operate if vapor flows through it, and the flow meters would give wrong readings if vapor flowed through them (particularly the positive displacement flow meter).

From the after-condenser, the refrigerant flows to the receiver. The receiver is a cylinder immersed in a temperature-controlled water tank. The receiver-water tank system is used to control the pressure in the refrigerant loop. The pressure in the loop can be adjusted by varying the temperature in the tank or by varying the amount of refrigerant that flows around the receiver in the receiver bypass.

After the receiver, the refrigerant passes through a water-cooled counter-flow heat exchanger. The purpose of this heat exchanger is to condense and subcool any vapor that
might have formed as the refrigerant passed through the receiver. Finally, the refrigerant passes through a filter/drier to remove water and impurities before it reaches the pump once again.

Some other pressure and temperature measurements, that have not been mentioned yet are performed in the refrigerant loop. Thermocouple probes measure the refrigerant temperatures at the heater inlet, adiabatic section inlet, test section outlet and after-condenser outlet. All of these are type-T thermocouples as well. The temperature measurement at the heater inlet is necessary to determine the enthalpy of the refrigerant at that point. The temperature measurement at the adiabatic section inlet and at the test section outlet are necessary to determine the refrigerant enthalpy at those points during one-phase tests. Some of these thermocouples are also used to verify the readings of other thermocouples during two-phase tests (redundant measurements), and to help locate problems, sources of bad data, etc. Absolute pressure measurements are also made at the heater inlet, at the test section outlet and between the after-condenser and the pump. All of these measurements are done with 0-1000 psi (0-6900 kPa) Setra™ transducers. The pressure measurement at the heater inlet is required to calculate the refrigerant enthalpy. The other two are used mainly for leak detection and for safety purposes.

3.1.2 Water loop

The water loop is an important part of the experimental apparatus. Fig. 3.6 is a schematic of the water loop. The water from the building supply line splits into three lines. The first line supplies water to the after-condenser after the water flows through a rotameter. The rotameter is not used to measure the water flow, but sometimes, when the full capacity of the after-condenser is excessive for the conditions at which the system is being operated (usually at very low refrigerant flow rates), a valve upstream of the rotameter is used to reduce the water flow, effectively reducing the capacity of the after-
condenser. In those cases, the rotameter gives an approximate indication of how much the water flow is being reduced.

The second line supplies water to the heat exchanger. These first two lines go to the waste line directly after passing through the after-condenser and heat exchanger respectively.

The third line passes first through a rotameter. This rotameter has a needle valve which is used to regulate the water flow rate in the test-condenser, but as mentioned earlier, it is not used to measure the water flow rate. Then, the water flows through a water heater. The water heater has a 5120 Btu/hr (1.5 kW) heating power, and it is controlled by a Variac. This heater-Variac system is used to obtain the desired water temperature at the inlet of the test-condenser, through which the water flows next. After flowing through the test-condenser, the water is discharged into a waste tank, which is emptied into the building waste line with a pump activated by a float switch.

3.1.3 Data acquisition system

The last component of the experimental apparatus is the data acquisition system. All the experimental data were transmitted to a Macintosh IIci™ computer, with a National Instruments NB-MIO-16L™ data acquisition board which was installed in the computer, and a Campbell Scientific 21X™ datalogger, which was connected to the serial port of the computer. The National Instruments™ board was used to receive the data from the refrigerant flow meters and the differential pressure transducers, because both the differential pressure and the refrigerant flow rate vary very fast, and this board has the capability of reading data at a very fast rate (40 kHz). The National Instruments™ board can receive up to eight analog input channels.

The Campbell Scientific™ datalogger is capable of reading data every 10 seconds. It was connected to two Campbell Scientific AM64™ multiplexers, making it capable of
receiving 64 analog signals. It was used to receive the data from all the thermocouples and the absolute pressure transducers because this data were usually very steady.

All the information was processed and displayed in the computer with LabView 2© software from National Instruments™. The data were later analyzed in a separate computer using Excel 4.0©.
Figure 3.1 Schematic of the refrigerant loop
To test condenser

From flowmeters

Figure 3.2 Schematic of the refrigerant heater

Figure 3.3--Schematic of the test-condenser and the adiabatic section
Figure 3.4 Schematic of the micro-finned tube
Figure 3.5 Schematic of the wall thermocouple mounting technique
Figure 3.6 Schematic of the water loop
CHAPTER 4
EXPERIMENTAL PROCEDURES

This chapter describes the procedures used to obtain the experimental data. First, the operation of the experimental facility is described. Then, the methods used to calculate experimental results, such as heat transfer coefficients, enhancement factors and penalty factors, as well as the methods used to calculate important parameters, such as heat losses and experimental uncertainties, are described. Finally, the experimental conditions over which the experiments were conducted are presented.

4.1 Experimental facility operation

After the construction and installation of the test-condenser was finished, the next step consisted in verifying that both the water loop and the refrigerant loop had no leaks. The examination of the water loop was relatively easy, because usually the only place in which leak problems were encountered was the test-condenser. The water leaks in the test-condenser were detected by turning the water flow on, pressurizing the water side and observing where the water was coming out of. Fixing the water leaks was also easy: after spotting the leaks, the area was dried and more glue was applied.

Finding and fixing the leaks in the refrigerant loop was not only harder than in the water loop, but also more critical because even though a small water leak did not affect the operation of the system significantly, a small refrigerant leak caused a large loss of refrigerant because of the high pressure that it is under, making the operation of the system impossible. To find refrigerant leaks, the first step was to evacuate the refrigerant loop with a vacuum pump. Vacuuming the refrigerant loop instead of pressurizing it with a compressed gas, such as nitrogen, had the advantage of avoiding pressure fluctuations due to temperature variations. Then, some valves in the refrigerant loop were closed, dividing it into three sections, each of which had one or more absolute pressure transducers that
allowed pressure monitoring. If the pressure in a section increased over time, that meant that there was a leak in it. After determining with this procedure the general location of the leaks, the exact location of the leaks was determined using one of two methods. The first method consisted in filling the loop with compressed nitrogen, and finding the leaks by applying a solution of water and soap. The second method consisted in filling the loop with HFC-134a vapor and finding the leaks with a refrigerant leak detector. Finally, the leaks were fixed depending on their source. Usually, leaks were found in threaded fittings, which were fixed by applying teflon tape and tightening them, and in compression fittings, which were fixed by tightening them or by replacing the ferrules.

Once the refrigerant loop was free of leaks, it was charged with refrigerant (HFC-134a in this case). To charge the system with refrigerant, first it had to be evacuated again with the vacuum pump, so that no air remained trapped in it, because any air remaining in the system could affect the experimental data. Then, a refrigerant bottle was connected to a charging valve in the refrigerant loop with a charging hose. The bottle was heated with a heat gun in order to generate a large enough pressure difference between the bottle and the refrigerant loop so that the refrigerant would flow from the bottle to the loop. A total charge of refrigerant ranging from 5.5 lbm to 6 lbm (2.5 - 2.7 kg) was required for the operation of the system.

The procedure described up to this point was conducted only once before the actual testing begun. The regular experimental procedures are described next.

Before turning on the heaters, pump, etc., some preparations had to be made. First, the data acquisition program was started. Also, a new ice-bath, used for referencing the thermocouple temperature readings, was prepared and put into place. After the ice-bath and data acquisition program were ready, the heater in the constant temperature water tank was turned on. Once the temperature in the constant temperature water tank had reached the desired value, the refrigerant pump was turned on. In general, the refrigerant mass flow rate was initially low and unstable. To solve this problem, the refrigerant heater was
turned on at a low setting. This produced some vapor which helped the refrigerant flow to the top of the heater and around the loop. Once the refrigerant flow had stabilized, the valve from the building water supply line was opened, supplying water to the after-condenser, the counterflow heat exchanger and the test-condenser.

The next step was achieving the desired conditions in the test-condenser, which meant reaching the target values for the saturation temperature, refrigerant mass flux, vapor quality and temperature difference between the wall and the refrigerant. Simultaneously, a difference of at least 3.6 °F between the inlet and outlet temperatures of the water in the test-condenser was maintained to keep the uncertainty in the difference between these two measurements low. The techniques used to obtain these desired conditions were described in Chapter 3.

After the target conditions were reached and had stabilized, the system was operated for a few more minutes before recording the data, to make sure the conditions did not change while the data were being recorded. The data acquisition lasted for 2 to 3 minutes at each set of conditions. While the data were being recorded, the water flow rate in the test-condenser was measured using the technique described in Chapter 3.

The data were later reduced using the procedures described in Section 4.2.

4.2 Data analysis

The experimental results were reduced using spreadsheet macros written in Microsoft Excel 4.0. This method of data reduction has the advantage of being very flexible and producing an output that facilitates further analysis.

The refrigerant-side heat transfer coefficient in the test-condenser was determined by using the following procedure, which is similar to the procedure followed by Dobson et al. [1994].
4.2.1 Refrigerant-side heat transfer coefficient

First, the refrigerant enthalpy at the inlet of the refrigerant heater was calculated like this:

\[ i_{h,i} = i_l(T_h,i) + u_l(T_h,i)(P_{h,i} - P_{sat}(T_{h,i})) \]  \hspace{1cm} (4.1)

Then, the enthalpy of the refrigerant at the inlet of the test-condenser was determined with an energy balance across the refrigerant heater:

\[ \dot{Q}_h - \dot{Q}_{l,h} = m_r(i_{ts,i} - i_{h,i}) \]  \hspace{1cm} (4.2)

In Eq. (4.2), the heat input in the refrigerant heater, \( \dot{Q}_h \), and the refrigerant mass flow rate, \( m_r \), were measured directly. The refrigerant enthalpy, \( i_{h,i} \), was calculated using Eq. (4.1). So the only remaining unknown quantity required to calculate the enthalpy of the refrigerant at the inlet of the test-condenser is the heat loss in the refrigerant heater, \( \dot{Q}_{l,h} \). The method used to determine that heat loss is explained later in this chapter.

The vapor quality at the inlet of the test-condenser was calculated as:

\[ x_{ts,i} = \frac{i_{ts,i} - i_l(T_{ts,i})}{i_{lv}(T_{ts,i})} \]  \hspace{1cm} (4.3)

An energy balance in the test-condenser gives:

\[ \dot{Q}_r = \dot{Q}_w + \dot{Q}_{l,ts} \]  \hspace{1cm} (4.4)

where
\[ \dot{Q}_r = m_r(i_{r,i} - i_{r,o}) \]  
(4.5)

\[ \dot{Q}_w = m_w c_p w(T_{w,o} - T_{w,i}) \]  
(4.6)

Equations (4.4), (4.5) and (4.6) are used to calculate the refrigerant-side heat transfer in the test-condenser, \( \dot{Q}_r \). All the required quantities are known, except for the heat loss in the test-condenser, \( \dot{Q}_{\text{loss}} \). The method used to determine that heat loss is explained later in this chapter, along with the method used to determine the heat loss in the refrigerant heater.

Finally, the refrigerant-side heat transfer coefficient was calculated with the following equation:

\[ h = \frac{\dot{Q}_r}{A_s (\bar{T}_{\text{sat}} - T_s)} \]  
(4.7)

where

\[ \bar{T}_{\text{sat}} = \frac{T_{\text{sat},i} + T_{\text{sat},o}}{2} \]  
(4.8)

where

\[ T_{\text{sat},o} = T_{\text{sat}}(P_{\text{ts},i} - \Delta P_{\text{ts}}) \]  
(4.9)

The refrigerant-side heat transfer area, \( A_s \), is the real inside heat transfer area of the micro-finned tube. The ratio of this area to the area of a smooth tube having the same inside diameter as the maximum inside diameter of the micro-finned tube used in this study is 1.62. In the remainder of this paper, that ratio will be referred to as the area ratio, \( A_{\text{rat}} \).
The full procedure required to calculate the refrigerant-side heat transfer coefficient is presented in Eq. (4.1) through (4.9), except for the calculation of the heat losses in the test-condenser and in the refrigerant heater.

The heat loss in the test-condenser was determined with subcooled liquid tests. Subcooled liquid refrigerant entered the condenser, where it was cooled by the water. The heat transfer rates in the water side and in the refrigerant side were calculated with Eq. (4.5) and (4.6). By substituting these quantities in Eq. (4.4) the heat loss was determined. This process was repeated for a wide range of water temperatures, and it was observed that the heat losses increased as the difference between the water and ambient temperatures increased. It was also noted that when the water temperature was lower than the air temperature, instead of heat loss to the ambient, a heat gain would be obtained. Therefore, the following overall heat conductance model was implemented:

\[ \dot{Q}_{1,ts} = U_{A1,ts} (T_w - T_a) \]  

where \( U_{A1,ts} \) is the overall heat conductance. The conductance included the convection resistances of the water and air and the conduction resistance of the insulation. The conduction resistance through the insulation is the dominant resistance in the heat conductance. A series of single-phase tests were performed by Dobson et al. [1994], and then a linear curve fit of \( \dot{Q}_{1,ts} \) versus \((T_w - T_a)\) was obtained, with the slope taken as \( U_{A1,ts} \). For the 0.375" (9.52 mm) o.d. tube, the \( U_{A1,ts} \) obtained was 0.86 Btu/hr-R (0.45 W/K). Since the water side of the test-condenser used in the present study is identical to the water side of the 0.375" (9.52 mm) o.d. test-condenser used by Dobson et al. [1994] (the outside diameter of the inner tube is 0.375" [9.52 mm] in both cases and the water annulus has the same dimensions), and the insulation is also identical for the two test-condensers, the conductance values previously obtained were applied in this study too. The conductance is then used to calculate the heat losses in the two-phase tests.
A similar procedure is used to calculate the heat losses in the refrigerant heater. The refrigerant heater used for this study is the same one used by Dobson et al. [1994] for the 0.375" (9.52 mm) o.d. test-condenser, so the conductance value found in that study is used here too.

4.2.2 Enhancement factor and penalty factor

One of the main goals of this study is to compare the performance of the micro-finned tubes to the performance of the smooth tubes. Two parameters are used for this purpose: one that compares heat transfer characteristics and one that compares pressure drop characteristics.

The parameter used to compare heat transfer characteristics is the enhancement factor, EF. The enhancement factor is defined as the ratio of the refrigerant-side heat transfer rate in the micro-finned tube to the refrigerant-side heat transfer rate in the smooth tube, with the same temperature difference between the wall and the refrigerant, or:

\[
EF = \left( \frac{\dot{Q}_{\text{microfin}}}{\dot{Q}_{\text{smooth}}} \right)_{\text{with same } DT_w-r} 
\]

(4.11)

where

\[
\dot{Q}_{\text{microfin}} = (h_{\text{microfin}}A_{\text{microfin}}DT_{w-r}) 
\]

(4.12)

\[
\dot{Q}_{\text{smooth}} = (h_{\text{smooth}}A_{\text{smooth}}DT_{w-r}) 
\]

(4.13)

Since \( DT_{w-r} \) in Eq. (4.12) and (4.13) is the same, the enhancement factor can be re-written as:
\begin{equation}
EF = \left( \frac{h_{\text{microfin}}}{h_{\text{smooth}}} \right) \left( \frac{A_{\text{microfin}}}{A_{\text{smooth}}} \right) \tag{4.14}
\end{equation}

If the area of the smooth tube is taken as the area of a smooth tube with an inside diameter equal to the maximum inside diameter of the micro-finned tube used in this study, then the second term in parenthesis in Eq. (4.14) becomes the area ratio, and as mentioned earlier, for this tube:

\[ A_{\text{rat}} = 1.62 \tag{4.15} \]

So the final expression for the enhancement factor of the micro-finned tube used in this study is:

\begin{equation}
EF = \left( \frac{h_{\text{microfin}}}{h_{\text{smooth}}} \right) A_{\text{rat}} = 1.62 \left( \frac{h_{\text{microfin}}}{h_{\text{smooth}}} \right) \tag{4.16}
\end{equation}

Eq. (4.16) summarizes the advantages provided by a micro-finned tube over a smooth tube. First, the heat transfer rate is increased because of an increase in the heat transfer area. Second, the heat transfer rate is increased because of an increase in the heat transfer coefficient due to the turbulence created by the micro-fins. Of course, the extra turbulence also causes a higher pressure drop in the condenser.

The parameter used to compare the pressure drop characteristics of the micro-finned tube to those of a smooth tube is the penalty factor, PF. The penalty factor is defined as the ratio of the pressure drop in the micro-finned tube to the pressure drop in a smooth tube, with both tubes having equal lengths, or:

\begin{equation}
PF = \left( \frac{\Delta P_{\text{microfin}}}{\Delta P_{\text{smooth}}} \right)_{\text{equal length}} \tag{4.17}
\end{equation}
The heat transfer coefficients in Eq. (4.16) and the pressure drops in Eq. (4.17) were experimentally determined for the micro-finned tube. For the smooth tube, the heat transfer coefficients were determined with Eq. (2.12) and (2.15), and the pressure drops were determined with Eq. (2.26). The diameter used in those equations is an equivalent flow diameter, \( D_{\text{eq,flow}} \). The equivalent flow diameter is related to the cross-sectional inside area of the micro-finned tube by the following equation:

\[
A_{\text{cross-sect}} = \frac{\pi}{4} (D_{\text{eq,flow}})^2 \tag{4.18}
\]

### 4.3 Uncertainty analysis

Dobson et al. [1994] performed a detailed uncertainty analysis in calculating the heat transfer coefficient, using the method described by Moffat [1988]. That method calculates the uncertainty in a variable \( y \) that depends on \( N \) independent variables \( (x_i) \) for each of which the uncertainty \( (\delta_i) \) is known. The uncertainty in \( y \) is calculated:

\[
\delta y = \sqrt{\sum_{i=1}^{N} \left( \frac{\partial y}{\partial x_i} \delta x_i \right)^2} \tag{4.19}
\]

That study produced the following expression to calculate the uncertainty in the heat transfer coefficients:

\[
\frac{\delta h}{h} = \left[ \left( \frac{m_w c_{p,w}}{(T_{w,i} - T_{w,o})} \right)^2 \left( \frac{\delta T_{w,i}^2 + \delta T_{w,o}^2}{m_w c_{p,w}(T_{w,o} - T_{w,i})^2} + \delta m_w^2 \right)^2 + \delta Q_i^2 \right]^{0.5} + \ldots \tag{4.20}
\]
The instrumentation used in the present study is identical to the instrumentation used by Dobson et al. [1994] so the uncertainty in each of the measurements is the same. The only uncertainty that could have changed is that of the wall temperature measurements in the test-condenser, but as discussed in Chapter 3, it did not change. Because the mass flow rates, temperature ranges, heat losses and dimensions contained in Eq. (4.20) are very similar for both studies, the uncertainty results obtained in that study are applicable to the present study. Therefore, the uncertainty in the experimental heat transfer coefficient values obtained in the present study can be expected to range from 5% to 15%, which was the range calculated in the previous study.

4.4 Test conditions

The test conditions for this study are summarized in Table 4.1. As mentioned in Chapter 3, all the experiments were conducted in a 0.375" (9.52 mm) o.d. micro-finned copper tube with 60 fins and an 18 degree helix angle. The average vapor quality range of 0.10-0.90 presented in Table 4.1 is approximate. In some instances, that range was altered slightly due to practical considerations. A typical quality change in the test-condenser for one data point was 15%.
### Table 4.1 Test conditions

<table>
<thead>
<tr>
<th>Quantity</th>
<th>Conditions</th>
</tr>
</thead>
<tbody>
<tr>
<td>Refrigerant</td>
<td>HFC-134a</td>
</tr>
<tr>
<td>Mass flux klb m/ft²-hr (kg/m²-s)</td>
<td>55 (75)</td>
</tr>
<tr>
<td></td>
<td>110 (150)</td>
</tr>
<tr>
<td></td>
<td>165 (225)</td>
</tr>
<tr>
<td></td>
<td>220 (300)</td>
</tr>
<tr>
<td></td>
<td>295 (400)</td>
</tr>
<tr>
<td></td>
<td>330 (450)</td>
</tr>
<tr>
<td>Average vapor quality</td>
<td>0.10-0.90</td>
</tr>
<tr>
<td>Saturation temperature  °F (°C)</td>
<td>95 (35)</td>
</tr>
<tr>
<td>Wall-refrigerant temperature difference °F (°C)</td>
<td>Mass flux = 55 klb m/ft²-hr (75 kg/m²-s):</td>
</tr>
<tr>
<td></td>
<td>3.6 (2)</td>
</tr>
<tr>
<td></td>
<td>5.4 (3)</td>
</tr>
<tr>
<td></td>
<td>All other mass fluxes:</td>
</tr>
<tr>
<td></td>
<td>3.6 (2)</td>
</tr>
</tbody>
</table>
CHAPTER 5
EXPERIMENTAL RESULTS

This chapter presents the experimental results of this study. The effects of mass flux and quality on heat transfer, pressure drop, enhancement factor and penalty factor are analyzed. The results are also compared with previously developed correlations. The data used for the graphs contained in this chapter are presented in Appendix A.

Whenever the temperature difference between the wall and the refrigerant is not specified in this chapter, a temperature difference of 3.6 °F (2 °C) is implied.

5.1 Heat transfer results

5.1.1 Results from the present study

Figure 5.1 presents the experimental values of the Nusselt number versus vapor quality for all the mass fluxes tested. As mentioned earlier, the Nusselt number (heat transfer coefficient) was determined based on the real inside heat transfer area of the micro-finned tube. The graph shows similar trends to those observed in smooth tubes in previous studies by Dobson et al. [1994] and Gaibel et al. [1994]. The Nusselt number increases with quality, but the slopes for the lower mass fluxes are lower than those of the higher mass fluxes. This is explained by the fact that at the lower mass fluxes the dominant flow regime is the wavy regime, which is characterized by a low dependence of Nusselt number on quality. On the other hand, at higher mass fluxes, the dominant flow regime is the annular regime, which is characterized by a strong dependence of Nusselt number on quality.

However, an interesting difference between the trends in the micro-finned tube and the smooth tubes was observed at the higher mass fluxes. For the smooth tubes, both of the mentioned studies found an increase in the Nusselt number as the mass flux was
increased, for the whole mass flux range. This was also the case for the micro-finned tube at the lower mass fluxes. However, at the higher mass fluxes, the Nusselt number showed very little dependence on mass flux. This phenomenon can be observed better in Fig. 5.2 and Fig. 5.3.

Figures 5.1 through 5.3 show the actual experimental Nusselt numbers, but no information on the performance of the micro-finned tube with respect to the performance of the smooth tubes can be obtained from them. That information can be observed in Fig. 5.4 where the enhancement factor, as defined in Eq. (4.16), is plotted against quality for all the mass fluxes. An important characteristic observed in the enhancement factors plotted in Fig. 5.4 is that all of them are higher than 1.0, but some are lower than 1.62. This means that there is always an increase in the heat transfer rate when using a micro-finned tube instead of a smooth tube (with dimensions as specified in Chapter 4), but at certain conditions, that increase is not even as large as the increase in the heat transfer area.

Fig. 5.4 shows a general decrease in the enhancement factor as the mass flux is increased. This is explained by the fact that as the mass flux is increased, more turbulence is generated in the flow, reducing the relative effect of the micro-fins.

Some other results observed in Fig. 5.4 can be explained by the flow regime prevailing at each set of conditions. At the lowest mass flux, 55 klb/ft²-hr (75 kg/m²-s), a high enhancement factor of almost 2.0 is observed. The enhancement factor at this mass flux is almost independent of quality. The flow regime for this mass flux was the wavy-stratified regime at every quality. The main mechanism of heat transfer in this regime is conduction through the liquid film at the top of the tube. The liquid film at the top of the tube is very thin, so the presence of fins reduces the thickness of the film by a considerable amount at the spots where the fins are located, which also reduces the conduction resistance through it. Therefore, a large enhancement factor is obtained under these conditions. And since wavy-stratified flow prevailed for the full quality range, the enhancement factor remained relatively constant as the quality was varied.
For the remaining mass fluxes, some common trends can be observed. At low qualities, the enhancement factors are relatively high. Then, as the quality is increased, the enhancement factors decrease. Finally, a further increase in the quality causes a large increase in the enhancement factors. These trends can also be explained from a flow regime point of view. At low qualities, the wavy-stratified regime is the dominant regime. For the reasons previously explained, this produces high enhancement factors. As the quality is increased, the flow regime changes from wavy-stratified to annular. In the annular regime, an annular film forms around the tube. The thickness of that film is initially large compared to the fin height, which reduces the relative effect of the fins, reducing the enhancement factor. As the quality is increased even further, the thickness of the film is reduced enough so that the fin height becomes important with respect to the film thickness again. This produces high enhancement factors at high qualities. In fact, since the film is now located all around the tube instead of only at the top of the tube, as in wavy-stratified flow, the enhancement factors are often even higher than the enhancement factors obtained in the wavy-stratified regime.

A final important observation from Fig. 5.4 related to the flow characteristics is that at high qualities, the enhancement factor decreases as the mass flux is increased. This is probably due to the fact that at a specific quality, the thickness of the liquid film is increased when the mass flux is increased. This may also explain in part the trend observed in Fig. 5.3, where it is shown that at high mass fluxes, increasing the mass flux has almost no effect on the Nusselt number. A possible explanation of this phenomenon is that there are two offsetting effects that keep the Nusselt number almost constant as the mass flux is increased. First, as the mass flux is increased, an increase in the forced convective heat transfer is expected. But as the mass flux is increased, the thickness of the liquid layer also increases, reducing the effect of the micro-fins.

Figures 5.5 through 5.10 show the effect of quality on the enhancement factor for each mass flux separately, which allows a clearer identification of the trends found at each
mass flux. They also present curve fits of enhancement factor versus quality. Those curve fits are used to integrate the enhancement factors over the 0.10-0.90 quality range and obtain values near the average enhancement factor for each mass flux. This is necessary in order to compare the results of the present study to results of previous studies, since those studies usually perform full condensation, obtaining average Nusselt numbers and enhancement factors.

The "average" enhancement factors obtained by integration are presented in Table 5.1 and plotted against mass flux in Fig. 5.11. The decrease of the enhancement factor as the mass flux is increased is evident in Fig. 5.11 and Table 5.1.

However, there are two exceptions to this trend. First, there is a slight increase in the enhancement factor when the mass flux is increased from 55 klb m/ft²-hr (75 kg/m²-s) to 110 klb m/ft²-hr (150 kg/m²-s). This is probably explained by the fact that for practical reasons (at low refrigerant mass fluxes, larger quality changes occur in each experiment), the Nusselt number was not determined for the 55 klb m/ft²-hr (75 kg/m²-s) mass flux at the highest qualities. The trends observed for the other mass fluxes suggest that a very high enhancement factor would have been obtained for this mass flux at high qualities, increasing the average enhancement factor.

The other exception is observed for the two highest mass fluxes. The average enhancement factors for these two mass fluxes are similar. This may be due to the small difference in mass flux between them. In fact, a slightly lower enhancement factor was obtained for the highest mass flux, but after the numbers were rounded, they came up equal.

5.1.2 Comparison with previous studies

A direct comparison between the results of this study and the results of previous studies is very difficult because of the different geometries used by different investigators and because most of the other investigators studied average (not local) characteristics.
Table 5.1 Average enhancement factors

<table>
<thead>
<tr>
<th>Mass flux</th>
<th>Enhancement factor</th>
</tr>
</thead>
<tbody>
<tr>
<td>klb_m/ft^2-hr (kg/m^2-s)</td>
<td></td>
</tr>
<tr>
<td>55 (75)</td>
<td>1.9</td>
</tr>
<tr>
<td>110 (150)</td>
<td>2.0</td>
</tr>
<tr>
<td>165 (225)</td>
<td>1.9</td>
</tr>
<tr>
<td>220 (300)</td>
<td>1.7</td>
</tr>
<tr>
<td>295 (400)</td>
<td>1.4</td>
</tr>
<tr>
<td>330 (450)</td>
<td>1.4</td>
</tr>
</tbody>
</table>

However, Eckels and Pate [1991] tested HFC-134a in a tube with similar geometric characteristics to the tube used in the present study. Both tubes have the same o.d., the same number of fins and very similar fin height, helix angle and i.d.. For full condensation at 104 °F (40 °C) and mass fluxes of 110 and 220 klb_m/ft^2-hr (150 and 300 kg/m^2-s) they found average enhancement factors of 2.2 and 1.9 respectively. This values are comparable to the integrated average values presented in Table 5.1 of 2.0 and 1.7 for those two mass fluxes. These results are plotted in Fig. 5.12.

Schlager et al. [1989c] also tested R-22 in a tube with a similar geometry to that of the tube used in this study. That study found average enhancement factors of around 1.85, 1.7, 1.6 and 1.55 for mass fluxes of 150 klb_m/ft^2-hr (200 kg/m^2-s), 220 klb_m/ft^2-hr (300 kg/m^2-s), 295 klb_m/ft^2-hr (400 kg/m^2-s) and 365 klb_m/ft^2-hr (500 kg/m^2-s) respectively. The condensation temperature was approximately 104 °F (40 °C). These results are compared to the results of the present study in Fig 5.12.

The results of the three studies presented in Fig. 5.12 are remarkably similar, particularly if it is considered that different refrigerants, tube geometries and experimental
techniques (full condensation versus partial condensation, Wilson plots versus wall temperature measurements) were utilized.

The heat transfer results of this study were compared with the correlation by Koops and Azer [1993] (Eq. (2.11)). However, that correlation is not a good predictor of the data of this study. This may be due to the fact that the tubes used to develop the correlation had fewer and larger fins than the tube used in this study.

The average enhancement factors were compared with the correlation by Schlager et al. [1990b] (Eq. (2.8)), with the 220 klbm/ft²-hr (300 kg/m²-s) mass flux as the "base" mass flux. The average enhancement factors were predicted within ±20% with this correlation. This comparison is presented in Fig. 5.13a and Fig. 5.13b.

5.1.3 Effect of wall-refrigerant temperature difference on wavy flow heat transfer

While most of the experiments were run with a 3.6 °F (2.0 °C) difference between the refrigerant temperature and the average wall temperature, the 55 klbm/ft²-hr (75 kg/m²-s) mass flux experiments were also run with a 5.4 °F (3.0 °C) temperature difference. This was done in order to study the effect of the wall-refrigerant temperature difference on heat transfer in the wavy regime. Dobson et al. [1994] found the film Nusselt number to be proportional to the wall-refrigerant temperature difference to the minus 0.25 power. The reason this mass flux was chosen for this analysis is that it was the only mass flux for which wavy flow was observed at every quality.

Fig. 5.14 shows the effect of quality and wall-refrigerant temperature difference on the Nusselt number. The values of the Nusselt number are a little higher for the 3.6 °F (2.0 °C) temperature difference data.

The result of multiplying the Nusselt numbers of Fig. 5.14 by the wall-refrigerant temperature difference to the 0.25 power is presented in Fig. 5.15. Here the resulting values are very similar for both temperature differences, which means that for wavy flow in
the micro-finned tube there is a similar wall-refrigerant temperature difference dependence of the Nusselt number as in the smooth tube.

The analysis presented above is possible only because the film condensation component is much more important than the forced convection condensation component in wavy flow (the forced convection condensation component does not depend on the temperature difference between the wall and the refrigerant). Therefore, multiplying the Nusselt number by the wall-refrigerant temperature difference to the 0.25 power produces almost the same result as multiplying the film component of the Nusselt number by the same quantity.

5.2 Pressure drop results

5.2.1 Results from the present study

The experimental pressure drops per unit length are presented in Fig. 5.16 for the three highest mass fluxes tested. The uncertainty in the pressure drop measurements for the lower mass fluxes was in the order of the pressure drop itself, so those results are not presented in this paper.

Fig 5.16 shows similar trends to the ones observed in earlier works for smooth tubes, such as Hinde et al. [1992]. The pressure drop increases as the mass flux is increased, due to the higher fluid speed. The pressure drop also increases as quality is increased for most of the quality range. The exception to that trend is at the higher qualities, where the pressure drop decreases as quality increases.

The reason for this behavior can be explained from a flow regime point of view. In the middle of the quality range, the flow regime is the annular regime. The vapor speed is much higher than the liquid speed, and the liquid-vapor interface is characterized by surface waves caused by the high vapor speed. Therefore, the liquid annulus acts like a rough surface for the vapor, producing a high pressure drop. As the vapor quality is increased,
the vapor speed becomes even higher, producing higher pressure drops. But increasing the vapor quality also reduces the thickness of the liquid annulus. At high qualities, the liquid annulus becomes very thin and even disappears at extremely high qualities, causing a reduction in the pressure drop.

Fig. 5.17 shows curve fits of pressure drop versus quality for each mass flux presented in Fig. 5.16. Those curve fits are used to integrate the pressure drop over the 0.1-0.9 quality range and calculate "average" pressure drops, which are later used for comparison purposes. The average pressure drops calculated using this technique are the following:

- For $G=220$ klb/m$^2$/ft$^2$-hr (300 kg/m$^2$-s): $0.120$ psi/ft (2.71 kPa/m)
- For $G=295$ klb/m$^2$/ft$^2$-hr (400 kg/m$^2$-s): $0.204$ psi/ft (4.61 kPa/m)
- For $G=330$ klb/m$^2$/ft$^2$-hr (450 kg/m$^2$-s): $0.263$ psi/ft (5.95 kPa/m)

Multiplying any of these pressure drops by the square of the ratio of any of the other two mass fluxes to the mass flux corresponding to that pressure drop gives approximately the pressure drop corresponding to the other mass flux. For example:

$$0.120 \left( \frac{330}{220} \right)^2 = 0.270 \approx 0.263$$

This is expected since the mass flux is proportional to the velocity and the pressure drop is proportional to the square of the velocity.

The pressure drop characteristics of the micro-finned tube with respect to the smooth tube are presented in Fig. 5.18 where the penalty factor is plotted against average quality (see Chapter 4), for each mass flux. The graph shows that the penalty factor has a low dependence on quality and mass flux.

For the two higher mass fluxes, the penalty factors are almost identical. For both of them the penalty factor decreases slightly with quality at low qualities and becomes almost constant at a quality of approximately 40%.
For the lowest mass flux, the penalty factor is slightly lower than that of the other two mass fluxes. For this mass flux, the penalty factor decreases very slightly with quality throughout the whole quality range.

Fig. 5.19 shows curve fits of penalty factor versus quality for each mass flux.

Fig. 5.20 plots the "average" penalty factors versus mass flux. The average penalty factors were determined with the curve fits presented in Fig. 5.19 and the integration technique used to calculate the average pressure drops and enhancement factors. Those average penalty factors are the following:

- For $G=220 \text{ klb m/ft}^2\cdot\text{hr} \ (300 \text{ kg/m}^2\cdot\text{s})$: 1.19
- For $G=295 \text{ klb m/ft}^2\cdot\text{hr} \ (400 \text{ kg/m}^2\cdot\text{s})$: 1.25
- For $G=330 \text{ klb m/ft}^2\cdot\text{hr} \ (450 \text{ kg/m}^2\cdot\text{s})$: 1.26

These results and Fig. 5.20 show how similar the average penalty factors are for the three mass fluxes.

5.2.2 Comparison with previous studies

As with the heat transfer results, it is difficult to directly compare the pressure drop results of this study with those of previous studies, due to the different geometries and experimental techniques used. In the next few paragraphs, the pressure drop results are compared to the results of previous studies which used tubes with the closest geometry to the tube used in the present study.

Eckels and Pate [1991] (see section 5.1.2) found condensation average penalty factors of approximately 1.5 for a mass flux of $G=220 \text{ klb m/ft}^2\cdot\text{hr} \ (300 \text{ kg/m}^2\cdot\text{s})$ and 1.25 for a mass flux of $G=275 \text{ klb m/ft}^2\cdot\text{hr} \ (375 \text{ kg/m}^2\cdot\text{s})$, for HFC-134a. These results are compared with the results of the present study in Fig. 5.21.

Schlager et al. [1989c] (see section 5.1.2) found average penalty factors of approximately 1.2 for a mass flux of $G=220 \text{ klb m/ft}^2\cdot\text{hr} \ (300 \text{ kg/m}^2\cdot\text{s})$ and 1.22 for mass fluxes of $G=295 \text{ klb m/ft}^2\cdot\text{hr} \ (400 \text{ kg/m}^2\cdot\text{s})$ and $G=330 \text{ klb m/ft}^2\cdot\text{hr} \ (450 \text{ kg/m}^2\cdot\text{s})$ during
condensation of R-22. These results are also compared with the results of the present study in Fig. 5.21.

Fig. 5.21 shows that the three studies compared in it obtained very similar penalty factors, except for the penalty factor obtained by Eckels and Pate [1991] at $G=220 \text{klb}_m/\text{ft}^2\cdot\text{hr}$ (300 kg/m$^2$-s), which is higher than the penalty factors obtained by the other two studies for that mass flux.

The pressure drop results were also compared with the correlation by Kaushik and Azer [1990] (Eq. (2.10)). This correlation is not a good predictor of the pressure drop results of the present study. However, that correlation was developed with data obtained from tubes with fewer and larger fins, and from experiments at lower mass fluxes.

5.3 Analysis of heat transfer and pressure drop results

5.3.1 Comparison of the enhancement factors with the penalty factors

The results presented up to this point show the heat transfer gain and the pressure drop penalty obtained as a result of using micro-finned tubes instead of smooth tubes. However, the relative magnitudes of the heat transfer gain and the pressure drop penalty have not been analyzed. Fig. 5.22 through 5.24 compare the enhancement factor with the penalty factor at each quality for the three highest mass fluxes. The enhancement factor is higher than the penalty at every point except one point for the $330 \text{klb}_m/\text{ft}^2\cdot\text{hr}$ (450 kg/m$^2$-s) mass flux, where both quantities are almost equal. The difference between the enhancement factor and the penalty factor tends to decrease as the mass flux is increased. This is due mainly to the decrease in the enhancement factor.

5.3.2 Condenser simulation

A quick look at the results of Section 5.3.1 may suggest that the advantages in terms of heat transfer are greater than the disadvantages in terms of pressure drop when
using a micro-finned tube instead of a smooth tube. However, this is not necessarily the case. Other things have to be taken into account in order to make that judgment.

First, the design criteria of each application should be analyzed. For example, in some applications the size of the heat exchangers may be the critical design parameter, so a good enhancement factor may be desired regardless of the penalty factor. In other applications, the cycle efficiency may be critical, so low pressure drops are desired.

Another important consideration is the air/water side resistance. If the air/water side resistance is the dominant resistance in the heat exchanger, enhancing the refrigerant side heat transfer will not have an important effect on the overall thermal resistance and heat transfer.

In order to understand the effects of using a micro-finned tube instead of a smooth tube on the overall performance of a heat exchanger, a simulation program that combines the heat transfer and pressure drop data from this study, the heat transfer correlations presented in Dobson et al. [1994], the pressure drop correlation of Souza et al. [1992] and the return bend pressure drop correlation of Christoffersen et al. [1993], was written using a Microsoft Excel 4.0© spreadsheet.

This program is very similar to the one described by Dobson et al. [1994] so it will only be described briefly. The only difference between the present version of the program and the earlier one is that the curve fits of enhancement factor and penalty factor versus quality (presented in Fig. 5.8, 5.9 and 5.19) are included in this version of the program as multiplicative factors for the smooth tube heat transfer and pressure drop correlations.

The heat transfer rate in the heat exchanger is calculated with the following equation:

\[ \dot{Q} = \frac{L(T_{\text{sat}} - T_a)}{R_{\text{ref}} + R_a} \]  

(5.1)

where
\[ R_{\text{ref}} = \frac{1}{\pi D h_{\text{ref}}} = \frac{1}{\pi N \text{u}_{\text{ref}} k_1} \]  

(5.2)

\( R' \) is the thermal resistance per unit length. The air side resistance per unit length, \( R_{\text{a}}' \), is treated as a known constant. The refrigerant side heat transfer coefficient, \( h_{\text{ref}} \), is calculated by multiplying the smooth tube heat transfer coefficient by the enhancement factor. As discussed in Chapter 4, the enhancement factor includes both the effect of the heat transfer area increase and the heat transfer coefficient increase, so the value obtained for \( h_{\text{ref}} \) with the above technique is not exactly the heat transfer coefficient for the micro-finned tube. However, this technique produces the same result as multiplying the heat transfer coefficient by the heat transfer coefficient ratio, and the heat transfer area per unit length, \( \pi D \), by the area ratio.

The program divides the heat exchanger into elements with a 5% vapor quality change in each, and calculates the length and pressure drop in each of them. The total length and pressure drop are then calculated.

The inputs to the program are: refrigerant mass flow rate, tube i.d., air temperature, inlet refrigerant temperature, return bend diameter, distance between return bends and air/water side resistance per unit length. The effect of the return bends was neglected in this analysis in order to simplify it and concentrate on the effects of the different tube types.

The first part of the simulation analysis consisted in four sets of conditions: two air/water side resistances and two refrigerant mass fluxes. The high air/water side resistance is typical of an air cooled residential air conditioner. The low air/water side resistance is typical of a water cooled condenser. At each set of conditions the program was run for a smooth tube (using the version by Dobson et al. [1994]) and for a micro-finned tube. The goals of this analysis are:
- Comparing the pressure drop and heat exchanger length of the micro-finned tube and smooth tube at each set of conditions, emphasizing the effects of the air/water side resistance.

- Examining the effect of refrigerant mass flux on the performance of the micro-finned tubes with respect to the smooth tubes.

The input conditions for this part of the analysis are presented in Table 5.2, while the results of the analysis are presented in Table 5.3.

<table>
<thead>
<tr>
<th>Table 5.2 Simulation conditions</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Quantity</strong></td>
</tr>
<tr>
<td>Air/water side resistance</td>
</tr>
<tr>
<td>Refrigerant mass flux</td>
</tr>
<tr>
<td>Inlet saturation temperature</td>
</tr>
<tr>
<td>Air temperature</td>
</tr>
</tbody>
</table>

For the 220 klb_m/ft²-hr (300 kg/m²-s) mass flux and the high air/water side resistance, there is about a 9% increase in the total pressure drop and about a 9% decrease in the total heat exchanger length when a micro-finned tube is used instead of a smooth tube. For that same mass flux and the low air/water side resistance, there is about a 6% decrease in the total pressure drop, and about a 23% decrease in the total heat exchanger length. This results show that when the air/water side resistance is decreased, and the refrigerant side resistance becomes more important, enhancing the refrigerant side heat transfer has a very important effect on the overall heat exchanger performance. For the low
air/water side resistance case, not only was the length of the heat exchanger reduced when the smooth tube was substituted with a micro-finned tube, but the pressure drop also decreased. The reason for this is that the length of the heat exchanger was reduced by a very significant amount, so even though the pressure drop per unit length for the micro-finned tube is larger than that of the smooth tube, the total pressure drop for the micro-finned tube is lower.

As mentioned earlier, the high air/water side resistance used in this analysis is typical of an air cooled residential air conditioner. But even though that resistance is much larger than that of a water cooled condenser, it is lower than that of a typical domestic refrigerator and even lower than that of an automotive air conditioner. From the above discussion it is obvious that as the air side resistance increases, using micro-finned tubes instead of smooth tubes will not produce an important reduction in the heat exchanger length, but the pressure drop will increase significantly.

For the 295 klb\text{m/ft}^2\text{-hr} (400 kg/m\text{^2-s}) mass flux, a general worsening in the performance of the heat exchanger with the micro-finned tube is observed with respect to the 220 klb\text{m/ft}^2\text{-hr} (300 kg/m\text{^2-s}) mass flux. This is explained by the fact that the heat transfer coefficients for those two mass fluxes were very similar, while the pressure drop for the 295 klb\text{m/ft}^2\text{-hr} (400 kg/m\text{^2-s}) mass flux was much higher than that of the 220 klb\text{m/ft}^2\text{-hr} (300 kg/m\text{^2-s}) mass flux.

In the cases where the heat exchanger length is reduced when using a micro-finned tube instead of a smooth tube, there is another "hidden" benefit that is not shown by the results of this analysis. For example, in the case of an air cooled stationary air conditioner, a reduction in the heat exchanger length means a reduction in the fan energy consumption.
Table 5.3 Results of simulation analysis

<table>
<thead>
<tr>
<th>Tube</th>
<th>$R_{a/w'}$ h-ft·°F/Brtu (m-K/W)</th>
<th>$G$ klbm/ft²·hr (kg/m²·s)</th>
<th>$\Delta P$ psi (kPa)</th>
<th>$L$ ft</th>
<th>$\Delta P/\Delta P_s$</th>
<th>$L/L_s$</th>
</tr>
</thead>
<tbody>
<tr>
<td>smooth</td>
<td>0.073 (0.042)</td>
<td>220 (300)</td>
<td>5.04 (34.76)</td>
<td>58.97 (17.98)</td>
<td>1</td>
<td>1</td>
</tr>
<tr>
<td>smooth</td>
<td>0.073 (0.042)</td>
<td>295 (400)</td>
<td>12.29 (84.71)</td>
<td>89.51 (27.29)</td>
<td>1</td>
<td>1</td>
</tr>
<tr>
<td>smooth</td>
<td>0.019 (0.042)</td>
<td>220 (300)</td>
<td>1.88 (12.97)</td>
<td>24.70 (7.53)</td>
<td>1</td>
<td>1</td>
</tr>
<tr>
<td>smooth</td>
<td>0.019 (0.042)</td>
<td>295 (400)</td>
<td>3.96 (27.28)</td>
<td>31.22 (9.52)</td>
<td>1</td>
<td>1</td>
</tr>
<tr>
<td>micro-finned</td>
<td>0.073 (0.042)</td>
<td>220 (300)</td>
<td>5.50 (37.92)</td>
<td>53.92 (16.44)</td>
<td>1.09</td>
<td>0.91</td>
</tr>
<tr>
<td>micro-finned</td>
<td>0.073 (0.042)</td>
<td>295 (400)</td>
<td>15.08 (103.94)</td>
<td>89.94 (27.42)</td>
<td>1.23</td>
<td>1.00</td>
</tr>
<tr>
<td>micro-finned</td>
<td>0.019 (0.042)</td>
<td>220 (300)</td>
<td>1.76 (12.11)</td>
<td>19.02 (5.80)</td>
<td>0.94</td>
<td>0.77</td>
</tr>
<tr>
<td>micro-finned</td>
<td>0.019 (0.042)</td>
<td>295 (400)</td>
<td>4.23 (29.17)</td>
<td>26.30 (8.02)</td>
<td>1.07</td>
<td>0.84</td>
</tr>
</tbody>
</table>

(Note:

$R_{a/w'}$ is the air/water side thermal resistance per unit length, $\Delta P$ and $L$ are the total pressure drop and length of the heat exchanger respectively, $\Delta P/\Delta P_s$ is the ratio of pressure drop in the micro-finned tube to the pressure drop in the smooth tube for the same mass flux and air side resistance and $L/L_s$ is the ratio of the micro-finned tube heat exchanger length to the smooth tube heat exchanger length for the same mass flux and air side resistance.)
The final part of the simulation analysis consisted in trying to optimize the heat exchanger performance by using the local enhancement factor and penalty factor information from this study. For the 220 klb\textsubscript{m}/ft\textsuperscript{2}-hr (300 kg/m\textsuperscript{2}-s) mass flux, high enhancement factors and low penalty factors were observed at high qualities. So the smooth tube and micro-finned tube simulation programs were combined for that mass flux and the high air-side resistance, to simulate a condenser composed of a micro-finned tube for vapor qualities ranging from 1 to 0.7, and a smooth tube for qualities ranging from 0.7 to 0.

The final results were the following:

- Micro-finned tube length: 14.3 ft (4.36 m)
- Micro-finned tube pressure drop: 2.31 psi (15.9 kPa)
- Smooth tube length: 44.0 ft (13.43 m)
- Smooth tube pressure drop: 2.92 psi (20.09 kPa)
- Total length: 58.35 ft (17.79 m)
- Total pressure drop: 5.22 psi (35.99 kPa)

If these results are compared to those of the smooth tube heat exchanger with the same mass flux and air/water side resistance, about a 1% decrease in the total length is observed, while an increase of about 3.5% in the pressure drop is observed. This shows that it is very difficult to obtain both a decrease in the heat exchanger length and the pressure drop when using a micro-finned tube instead of a smooth tube in an air cooled condenser, despite the fact that the enhancement factors are much larger than the penalty factors.
Figure 5.1 Effect of quality and mass flux on Nusselt number

Figure 5.2 Effect of quality and mass flux on Nusselt number at low mass fluxes
Figure 5.3 Effect of quality and mass flux on Nusselt number at high mass fluxes

Figure 5.4 Effect of quality and mass flux on enhancement factor
Figure 5.5 Effect of quality on enhancement factor for $G=55 \text{ klb/ft}^2\cdot\text{hr}$ (75 kg/m$^2$-s)

\[ EF = 1.79 + 0.23x \]

Figure 5.6 Effect of quality on enhancement factor for $G=110 \text{ klb/ft}^2\cdot\text{hr}$ (150 kg/m$^2$-s)

\[ EF = 3.02 - 14.64x + 57.76x^2 - 94.24x^3 + 54.75x^4 \]
Figure 5.7 Effect of quality on enhancement factor for $G=165 \text{ klb/ft}^2\cdot\text{hr}$ (225 kg/m$^2$·s)

Figure 5.8 Effect of quality on enhancement factor for $G=220 \text{ klb/ft}^2\cdot\text{hr}$ (300 kg/m$^2$·s)
Figure 5.9 Effect of quality on enhancement factor for $G = 295 \text{ klb} \text{m/ft}^2 \text{-hr}$ (400 $\text{kg/m}^2 \text{-s}$)

Figure 5.10 Effect of quality on enhancement factor for $G = 330 \text{ klb} \text{m/ft}^2 \text{-hr}$ (450 $\text{kg/m}^2 \text{-s}$)
Figure 5.11 Effect of mass flux on average enhancement factor

Figure 5.12 Comparison of average enhancement factors
Figure 5.13a Comparison of experimental EF with the correlation by Schlager et al. [1990b]

Figure 5.13b Experimental EF versus the correlation by Schlager et al. [1990b]
Figure 5.14 Effect of quality and wall-refrigerant temperature difference on Nusselt number for $G=55$ klb/ft$^2$-hr (75 kg/m$^2$-s)

Figure 5.15 Effect of $DT_{w-r}$ on $Nu^*(DT_{w-r})^{0.25}$ for $G=55$ klb/ft$^2$-hr (75 kg/m$^2$-s)
Figure 5.16 Effect of quality and mass flux on pressure drop

Figure 5.17 Curve fits of pressure drop versus quality
Figure 5.18 Effect of quality and mass flux on penalty factor

5.19 Curve fits of penalty factor versus average quality
Figure 5.20 Effect of mass flux on average penalty factor

Figure 5.21 Comparison of average penalty factors
Figure 5.22 Comparison of EF with PF for $G=220 \text{ klb/ft}^2\cdot\text{hr}$ (300 kg/m$^2$-s)

Figure 5.23 Comparison of EF with PF for $G=295 \text{ klb/ft}^2\cdot\text{hr}$ (400 kg/m$^2$-s)
Figure 5.24 Comparison of EF with PF for $G=330 \text{ klb/ft}^2\cdot\text{hr} \ (450 \text{ kg/m}^2\cdot\text{s})$
CHAPTER 6
CONCLUSIONS AND RECOMMENDATIONS

The purpose of this work was to study the condensation heat transfer and pressure drop characteristics of HFC-134a in a micro-finned tube with an 18 degree helix angle and to compare them to those in a smooth tube.

This chapter presents the main conclusions obtained from the results of the present study and recommends areas for future research.

6.1 Conclusions

As in the smooth tube, the heat transfer coefficient increases with vapor quality. The heat transfer coefficient also increases with mass flux at the lower mass fluxes (55, 110, 165 klb/r^2-hr) (75, 150, 225 kg/m^2-s) but at the higher mass fluxes (220, 295, 330 klb/r^2-hr) (300, 400, 450 kg/m^2-s) the heat transfer coefficient is basically independent of mass flux.

In general, the enhancement factors decrease with mass flux. The enhancement factors are relatively high for wavy-stratified flow. As quality is increased and the flow regime changes to annular flow, the enhancement factors decrease. At high qualities in annular flow the enhancement factors become high again. The local enhancement factors in this study range from about 1.2 to about 2.8.

The calculated average enhancement factors at each mass flux are relatively similar to the enhancement factors obtained by Eckels and Pate [1991] and Schlager et al. [1989c] for tubes with geometric characteristics similar to those of the tube used in the present study.

Attempts to predict the enhancement factors and heat transfer coefficients with existing correlations were unsuccessful. A correlation by Schlager et al. [1990b] was used to predict the average enhancement factors with some success. However, this correlation
only relates average enhancement factors among themselves and their respective mass fluxes, which means that the enhancement factor for at least one mass flux has to be determined with another technique (probably experimentally) in order to determine the average enhancement factors for the other mass fluxes.

In the wavy regime the heat transfer coefficient decreases as the temperature difference between the wall and the refrigerant is increased, for constant mass flux. When the heat transfer coefficients are multiplied by the wall-refrigerant temperature difference to the 0.25 power, similar results are obtained for wall-refrigerant temperature differences of 3.6 and 5.4 °F (2 and 3 °C). These seem to indicate that for the micro-finned tube in the wavy regime, the heat transfer coefficients are proportional to the wall-refrigerant temperature difference to the -0.25 power.

The pressure drop in the micro-finned tube increases with mass flux. The pressure drop also increases with vapor quality, except for the highest qualities, where the pressure drop decreases slightly. In fact, the shapes of the pressure drop versus quality curves for the micro-finned tubes are similar to those of the same curves for smooth tubes.

No pressure drop information for the lower mass fluxes is reported in this study due to the relatively high uncertainty in the pressure drop measurements at those mass fluxes. However, low mass fluxes are usually used in heat exchangers where the air/water side thermal resistance is much higher than the refrigerant side thermal resistance. In those cases, increasing the refrigerant side heat transfer coefficient does not have an important effect on the overall thermal resistance of the heat exchanger, but using micro-finned tubes does increase the pressure drop significantly. Therefore, the usefulness of this type of tube in low refrigerant mass flux heat exchangers is limited, and the pressure drop (and heat transfer) data are not as critical as the high mass flux data.

The penalty factors show little dependence on quality and mass flux. Local penalty factors range from 1.14 to 1.36.
The average penalty factors are similar to the penalty factors obtained by Eckels and Pate [1991] and Schlager et al. [1989c]. Attempts of predicting the experimental pressure drop results with existing correlations were also unsuccessful.

The enhancement factors are higher than the penalty factors for the conditions at which the experiments were conducted in this study.

A simulation program that uses the heat transfer and pressure drop data obtained in this study and pressure drop and heat transfer correlations from earlier studies was developed. The program showed that the relative performance of heat exchangers using micro-finned tubes with respect to heat exchangers using smooth tubes depends not only on the enhancement and penalty factors, but also on other factors such as air/water side thermal resistance. In fact, for low air/water side thermal resistances such as the ones typical of water cooled condensers, using micro-finned tubes instead of smooth tubes reduces not only the length of the heat exchanger but also the total pressure drop.

6.2 Recommendations

The present study should serve as a basis for future work on condensation in micro-finned tubes. More research needs to be conducted in the following areas: geometry, effects of lubricants on heat transfer and pressure drop, and condensation characteristics of zeotropic mixtures.

Regarding tube geometry, different helix angles and diameters should be studied. The results of this study seem to indicate that, in some instances, the heat transfer enhancement is mainly due to the heat transfer area increase, particularly at high mass fluxes. In these cases, a tube with a similar area ratio and straight fins may give a comparable heat transfer performance, while reducing the pressure drop. Work on a micro-finned tube with a similar geometry to the one used in the present study, but with straight fins instead of an 18 degree helix angle, is scheduled to begin in this project in the near future.
Different diameter tubes should be studied for two main reasons. First, a change in diameter may change the heat transfer and pressure drop characteristics, even at the same test conditions (mass flux, quality, saturation temperature, etc.) and for tubes with equal helix angle. Second, a smaller diameter tube will allow to test at higher mass fluxes than the ones tested in this study, with the same experimental apparatus. This is necessary in order to determine if the heat transfer coefficient continues to be independent of mass flux at higher mass fluxes or if it is independent of mass flux only in a specific mass flux range.

The data obtained from tubes with different diameters and helix angles should be combined with the data from the present study in order to formulate heat transfer and pressure drop correlations. No correlations were found that accurately predict the local heat transfer and pressure drop results from this study, and at this point, the data generated by the present study are not enough to formulate correlations.

The effects of lubricants on heat transfer and pressure drop characteristics in a micro-finned tube are of interest because many actual systems have oil circulation in them. The effects of lubricants in micro-finned tubes should be carefully studied because the presence of fins and grooves may make those effects significantly different from the ones observed in smooth tubes. The effects of oil are currently being studied in this project in the same tube used for the present study, and the results of this work will be presented in a future report.

Finally, the performance of zeotropic mixtures in micro-finned tubes should be studied in order to combine the benefits of these types of refrigerants and tubes. However, more research should be done on zeotropic mixtures in smooth tubes before testing them in micro-finned tubes because the heat transfer and pressure drop mechanisms in smooth tubes are not fully understood yet.

When testing with zeotropic mixtures, extreme care has to be taken in order to avoid even minor leaks in the system, because any leakage would tend to change their composition. Charging the system also requires extreme care, because different
compositions will be obtained if the refrigerant goes from the bottle to the system as vapor or as liquid. The best way to charge the system seems to be with liquid refrigerant, because, since most of the refrigerant (by mass) in the bottle is in the liquid state, the composition of the liquid should be very similar to the overall composition of the mixture.
REFERENCES


Moffat, R.J., 1988, "Describing the uncertainties in experimental results", Experimental Thermal and Fluid Science, 1, 3-17.


Schlager L.M, M.B. Pate, and A.E. Bergles, 1988a, "Evaporation and condensation of refrigerant-oil mixtures in a smooth tube and a micro-fin tube", ASHRAE Transactions, 94(1), 149-166.


APPENDIX A
EXPERIMENTAL DATA

Table A.1 presents the experimental data obtained in this study. The column headings and their respective units are:

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In the DP column, ND means not determined.
Table A.1 Experimental data

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APPENDIX B
THERMOPHYSICAL PROPERTIES

The thermodynamic and transport properties of HFC-134a were obtained from technical bulletins published by DuPont. In earlier work conducted in this project, the properties were curve fitted. Those curve fits were used to generate Table B.1.

In Table B.1., the column headings and their respective units are:

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<td>Pressure</td>
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