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Condenser Performance
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ABSTRACT

This study is the first to model and experimentally validate refrigerant inventory of R134a in small-channel cross-flow condensers†. This heat exchanger is used in automotive applications and uses smaller internal volumes than conventional heat exchangers to perform the same task. Since the cost of refrigerants continues to rise due to the phase out of chlorofluorocarbons (CFCs), internal volume becomes a key design parameter.

The model is a one-dimensional, two-fluid model which divides the condenser into several segments and modules. This model accurately predicts the rate of heat transfer and refrigerant pressure drop: the heat transfer was predicted within ±10% of the experiment and the pressure drop was predicted within ±30% for the majority of the data.

More importantly, the model predicts refrigerant inventory within ±10% of the experiments for ninety five percent of the data. In the inlet header, the slip ratio was correlated to the Reynolds and Froude numbers, and the homogeneous liquid volume fraction. In the small channels, the Reynolds and Weber numbers, and the homogeneous liquid volume fraction were used to correlate the slip ratio. For the outlet header, the dispersed liquid in the core was modeled using an unsteady gravity model and the annulus was modeled using the liquid-film Reynolds number and inverse viscosity.

Finally, the flow regimes were documented for the pipes, headers and small channel condenser tubes. Intermittent flow was the predominate flow regime in the small channels which is consistent with the Damianides flow map and Kelvin-Helmholtz stability criteria. Through the inlet header the flow transitioned from a dispersed liquid to a bubble flow regime. The flow regime in the outlet header was always a dispersed "gravity driven" liquid in the core with a thin liquid annulus on the wall. Visual data collected for the headers were in qualitative agreement with the refrigerant inventory model.

† Small-channel heat exchanger technology is the subject of United States and foreign patents applied for and issued to Modine Manufacturing Company, Racine, Wisconsin, USA. Current United States patents include 4,615,385, 4,688,311 and 4,998,580.
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NOMENCLATURE

# .......... number
A .......... area
chan ... channel
C .......... product of specific heat and mass flow rate
C_p ....... specific heat
CS ..... control surface
CV .... control volume
d ....... derivative operator
D .......... diameter
e .......... specific internal energy
E ....... internal energy
Ext .... external
F ...... force
g .......... acceleration due to gravity
h......... enthalpy
htc ...... heat transfer coefficient
H ....... height of fluid within channel
k.......... thermal conductivity
KE ...... kinetic energy
L ...... length of region of interest in direction of flow
m ...... mass
NOMENCLATURE (continued)

\( \dot{m} \) .... mass flow rate
\( \hat{n} \) .... unit vector normal to control surface
\( N \) .... dimensionless number
\( P \) .... pressure
\( p_1, p_2 \) .. parameters in void fraction correlation
\( pp \) .... momentum
\( \dot{Q} \) .... rate of heat transfer
\( \text{Res()} \) .. residual
\( \text{RH} \) .... relative humidity
\( S \) .... slip ratio \( S = \frac{U_g}{U_L} \)

\( \text{seg} \) .... segment
\( t \) .... time
\( \text{th} \) .... tube thickness
\( T \) .... temperature
\( u \) .... uncertainty
\( U \) .... velocity
\( U_A \) .... overall heat conductance
\( U_r \) .... relative velocity of fluid particle with respect to the control surface
\( V \) .... volume of region of interest
\( \dot{V} \) .... volumetric flow rate
Greek symbols

\( \alpha \) ....... void fraction \( \alpha = \frac{A_G}{A_L + A_G} \)

\( \beta \) ....... homogeneous void fraction (volumetric quality)

\( \delta \) ....... film thickness

\( \Delta \) ....... difference operator

\( \epsilon \) ....... effectiveness

\( \eta \) ....... overall surface efficiency

\( \lambda \) ....... wavelength

\( \mu \) ....... dynamic viscosity

\( \rho \) ....... density

\( \sigma \) ....... surface tension

\( \theta \) ....... angle

\( v \) ....... specific volume

Subscripts

\( ac \) ....... after-condenser

\( amb \) .... ambient

\( avg \) .... average

\( A \) ....... air

\( c \) ....... cross section perpendicular to flow

\( cond \) .... condenser

\( corr \) .... corrected

\( d \) ....... droplet

\( f \) ....... film

\( fric \) .... frictional

\( G \) ....... gas phase
NOMENCLATURE (continued)

Subscripts

Go .... gas only

grav ... gravitational

h ........ hydraulic

H ....... homogeneous

in........ inlet

L ........ liquid phase

meas .. measured

min .... minimum

mom ... momentum

mod ... module

n......... number of units

out...... outlet

O ....... overall

pc ...... pre-condenser

R ........ refrigerant

si ...... inside wetted surface

so ...... outside surface

surf .... surface

t......... total for mixture

tot ...... total

tt ....... turbulent-turbulent

w ........ water

Superscripts:

S ........ Superficial
NOMENCLATURE (continued)

Dimensionless numbers

\[ \beta \quad \text{volumetric quality} \quad \frac{1}{1 + \left( \frac{1-x}{x} \right) \left( \frac{\rho_g}{\rho_L} \right)} \]

Fr \quad \text{Froude number}

Ga \quad \text{Galileo number}

j \quad \text{Colburn j-factor} \quad \frac{\text{Nu}}{\text{Re} \sqrt{\text{Pr}}} \quad \frac{1}{N_f} \quad \text{inverse viscosity}

Pr \quad \text{Prandtl number}

Re \quad \text{Reynolds number}

We \quad \text{Weber number}

x \quad \text{thermodynamic vapor quality} \quad x = \frac{\dot{m}_g}{\dot{m}_L + \dot{m}_g}

X \quad \text{Lockhart-Martinelli parameter}

y \quad \text{homogeneous volumetric fraction}
CHAPTER 1
INTRODUCTION

1.1 Background

In 1974, Rowland and Molina published a paper linking man-made chlorofluorocarbons (CFCs) with the destruction of ozone in the stratosphere. Since that time, evidence continues to implicate CFCs. In the late 1980s and early 1990s governments have initiated aggressive campaigns to phase-out these compounds. Consequently, manufacturers and users of CFC's are profoundly affected. For instance, the short phase-out schedules have caused competitors to collaborate in an effort to solve their common problems. The University of Illinois has been instrumental in providing a framework for this collaboration through the formation of the Air Conditioning and Refrigeration Center (ACRC) in 1989.

The main goal of the ACRC is to contribute to the base of knowledge required to create new equipment that makes use of ozone-safe compounds. The ACRC has over twenty industrial sponsors that contribute both finances and staff. In addition to sending a representative to the semiannual Industrial Advisory Board meetings, the sponsor may designate a technical monitor for each project. This interaction with industry fosters a cooperative research environment and facilitates technology transfer. The research is divided into three areas: 1) processes (i.e. fundamental heat transfer and property studies), 2) components (i.e. evaporators, condensers, capillary and orifice tubes and suction line heat exchangers) and 3) systems (i.e. mobile and stationary air conditioning and domestic refrigerators). This project was within the component area. It made use of the findings from the processes area and provided a detailed component model to the systems area.

1.2 Objective

The ultimate goal of this work was to provide designers of vapor-compression systems with a tool they could use to predict refrigerant inventory in cross-flow
condensers. To accomplish this task, a computer simulation of a small-channel cross-flow condenser was developed and experimentally validated over the range of conditions typically found in automotive air conditioning systems. The small-channel condenser is a state-of-the-art heat exchanger which uses small refrigerant channels in a parallel-flow arrangement to enhance heat transfer and reduce weight. The working fluid used in this study was ozone-safe Refrigerant 134a (1,1,1,2-tetrafluoroethane).

Predicting refrigerant inventory in a condenser, especially in the two-phase flow region is a complex task. A large gap still exists between theory and experimental results. For example, the output from existing refrigerant inventory models can vary by as much as a factor of ten (Rice, 1987). An extensive literature search revealed that none of the models have been developed or applied to channels as small as the ones used in this study or to the headers which feed these channels. This study is the first one to model and experimentally validate refrigerant inventory of R134a in small-channel cross-flow condensers.

One key factor in determining refrigerant inventory is the flow regime, which is the form and structure of liquid and vapor. The amount of liquid in a passage can vary significantly from one flow regime to the next. An important contribution of this study is to document the flow regimes in cross-flow heat exchangers having small-channels arranged in parallel paths and connected with headers. This information can be used to predict the flow regime of fluids contained in a similar geometry and exposed to similar operating conditions.

Predicting refrigerant inventory is not only dependent on the flow regime and slip ratio but also requires accurate heat transfer modeling. The rate of heat transfer establishes the refrigerant quality along the length of the condenser tubes. Since refrigerant inventory is a strong function of refrigerant quality, (see Eqs. 2.01 through 2.06) any errors in the rate of heat transfer significantly affect the amount of refrigerant in the heat exchanger. Previous work in the ACRC has focused on modeling the heat
transfer and pressure drop of R134a (Weber (1991), Ragazzi (1991), Orth (1993) and Dobson (1993)). The work of these researchers is extended to small-channel condensers in the current study.

1.3 Motivation

There are three design issues related to vapor-compression systems which would benefit from an accurate refrigerant inventory model. A refrigerant inventory model will:

• Allow for optimization of refrigerant charge

Refrigerant charge has a significant effect on performance. Farzad (1990) studied the effect of refrigerant charge on system performance for air conditioners. For a three-ton (10.5 kW) air conditioner equipped with a capillary tube, the seasonal performance was degraded by 21 percent when the system was undercharged by 20 percent.

Since the amount of refrigerant affects the performance, it is important to find the quantity where the system will perform optimally. Currently, the refrigerant inventory for vapor compression systems is determined using costly trial and error experiments. After developing an air conditioner, many manufacturers will vary the amount of refrigerant in the system until the optimum performance is attained. An accurate computer model of the refrigerant in a system would reduce the time and expense of this process.

This study provides important information for a full-system model by focusing on the condenser. It is the two-phase region which provides the greatest uncertainty in predicting refrigerant inventory and the condenser is one of two components in a vapor compression system where two-phase flow is present.

• Provide needed constraint to system model

After applying the conservation laws of energy and momentum and the appropriate constitutive relations to a vapor compression system, there remains an
extra unknown variable. Many system models address this problem by specifying the state of the refrigerant at some point within the loop. In modeling a vapor compression system, this is undesirable since it is impossible to know this condition ahead of time.

A better way to find this extra variable is to provide another independent equation through the conservation of mass law. To apply this law to the system, an accurate model is needed to predict the mass of refrigerant in each of the components. Oak Ridge National Laboratory (ORNL) provides a version of their heat pump model which uses the continuity equation. This model requires refrigerant inventory as an input in place of one the refrigerant states within the heat pump. Farzad (1990) successfully demonstrated the capability of this model by comparing the output with experimental data.

- Improve heat transfer coefficient correlations

  Liquid refrigerant inhibits the heat transfer process in condensing flows. Predicting the quantity of liquid at each section of the heat exchanger will improve heat transfer coefficient correlations.

The next chapter reviews the literature on void fraction correlations and flow regime mapping. Chapter 3 presents the features of the experimental facility used for model validation. The simulation model is described in Chapter 4. The results of the flow regime and modeling are set forth in Chapters 5 and 6. Finally, Chapter 7 discusses the conclusions which were drawn from this work as well as recommendations for future work.
CHAPTER 2
THEORY AND LITERATURE REVIEW

2.1 Theory

This chapter begins by discussing theory relevant to interpreting the results of this study. Next, literature related to refrigerant inventory and flow regime transitions are reviewed.

2.1.1 Two-Fluid Model

The two-fluid flow model is used in this research to predict the amount of refrigerant in the two phase region of a small-channel condenser. This flow model treats the vapor and the liquid as two separate streams. (In most cases these two streams flow at different average velocities.) The liquid and vapor are also assumed to be in thermodynamic equilibrium with each other.

To use this model, it is necessary to develop an empirical relationship between the void fraction and the independent variables of the flow (e.g. mass flow rate, temperature and mass vapor quality). The following equations use the two-fluid flow model to predict the mass of refrigerant in a heat exchanger:

\[ m_g = A_c \int_0^L \rho_g \alpha dz \]  
\[ m_L = A_c \int_0^L \rho_L (1 - \alpha) dz \]  
\[ m_i = m_g + m_L \]

Figure 2.01 Illustration for Two-Fluid Flow Model
The cross sectional area and length of the channel are inputs to the model and therefore are known variables. The density of the gas and liquid can be determined from the property routines at the pressure and temperature of the refrigerant. The pressure and temperature are predicted by the heat transfer and pressure drop routines. Therefore, the only unknown in the equation above is the average void fraction (i.e. area occupied by gas/total cross-sectional area) along the length of the heat exchanger. Using the conservation of mass equation for each of the fluid streams a relationship for the void fraction can be derived as shown in Equation 2.06.

\[
\dot{m}_L = \rho_L A_L U_L \\
\dot{m}_G = \rho_G A_G U_G
\]

Dividing Equation 2.05 by Equation 2.04 and applying the definitions of quality, void fraction and slip ratio the following relationship results:

\[
\alpha = \frac{1}{1 + \left(\frac{1-x}{x}\right) \frac{\rho_G \cdot S}{\rho_L}}
\]

The mass vapor quality of a particular module can be accurately predicted from a physically based computer model for mobile air conditioning condensers (Zietlow et al., 1992). This model was updated to include correlations from Dobson (1993) and applied to small channel condensers. The densities were determined from property routines developed by the National Institute of Standards and Technology (NIST).

Therefore, the only variable left to be determined is the slip ratio. From the momentum equation (Collier (1980)) this slip ratio is affected directly by inertial, viscous, and gravitational forces. Surface tension forces can be important in determining the shape of the interface between the two streams especially in small channels. The shape of this interface affects the interfacial area and the interfacial friction factor (Wallis (1994)).
2.1.2 Evaluation of Dimensionless Forces

The dimensionless groups (Eqs. 2.07 to 2.10) suggested by Hughmark (1962) were used to evaluate the relative importance of the forces mentioned in the previous section. Table 2.01 summarizes the range of these dimensionless groups over the experimental test envelope. Refer to Figure 4.01 for a schematic of the condenser.

Table 2.01 Range of Dimensionless Groups for Each Section of Condenser

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<th>Reynolds(1000)</th>
<th>Froude</th>
<th>Weber</th>
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<td>Inlet pipe</td>
<td>34-224</td>
<td>2-115</td>
<td>Not Calculated</td>
</tr>
<tr>
<td>Pipe-outlet</td>
<td>4-224</td>
<td>0-117</td>
<td>Not Calculated</td>
</tr>
<tr>
<td>Inlet Header-top</td>
<td>15-140</td>
<td>.3-20</td>
<td>Not Calculated</td>
</tr>
<tr>
<td>Inlet Header-middle</td>
<td>6-68</td>
<td>.08-5</td>
<td>Not Calculated</td>
</tr>
<tr>
<td>Inlet Header-bottom</td>
<td>1-11</td>
<td>0-.17</td>
<td>Not Calculated</td>
</tr>
<tr>
<td>Condenser Channels</td>
<td>2-12</td>
<td>40-1520</td>
<td>4-120</td>
</tr>
</tbody>
</table>

Since these numbers can be interpreted as the ratio of forces they were used as a guideline to determine the relative importance of the forces. The Reynolds number is the ratio of inertial to viscous forces. In comparing the different sections of the condenser to each other it looks as if the viscous forces will be most important in the condenser channels and least important in the pipes. This agrees with the parameters found in the optimized model as displayed in Table 6.01 The Froude number is the ratio of inertial to gravitational forces. Gravitational forces are most important in the bottom of the inlet header and least important in the condenser channels. It is interesting to note that both viscous and gravitational forces become more important as the fluid travels down the inlet header. The reason for this was due to a reduction in the inertial forces as mass flow left the header though the condenser channels. The Weber number is the ratio of inertial to surface tension forces. Based on a discussion with Wallis (1994) the Weber number was used in place of the Froude number for the void fraction correlation applied to the condenser tubes. This gives a qualitative sense of which groups should be included in a
correlation for slip ratio. The Reynolds and Froude number for the pipes and inlet header and the Reynolds and Weber number for the condenser channels. The outlet header was not included in this analysis since the flow visualization results suggested a different type of analysis. This is presented in section 6.4.

2.2 Existing Correlations

Many empirical correlations have been developed to estimate void fraction in two-phase flows. Unfortunately, empirical relationships are often applicable only under conditions similar to the ones for which the correlation was developed. Attempts have been made at applying several of these correlations to refrigeration applications. Below is a summary of the relevant work.

2.2.1 Rice

Rice (1987) performed numerical experiments using 10 different void fraction models. Comparisons were made between the mass inventory predictions for heat exchangers over the operating range of residential heat pumps. He discovered significant differences in the results from these correlations. The maximum variation in refrigerant inventory ranged from a factor of 1.7 to a factor of 10.

Rice summarized each of the correlations and the conditions under which they were developed. He also identified which models were best suited for refrigeration applications. Based on the experimental work of others he recommends the Hughmark, Premoli, Tandon and Barcozy correlations for refrigeration applications. These models were examined further in this study.

2.2.2 Hughmark

The Hughmark (1962) correlation was developed for air-water systems flowing upward in vertical pipes. After developing the correlation, it was validated against
experimental results for steam-water flow in vertical upward flow and with several different air-liquid systems in horizontal co-current flow. In addition to water, the liquids included oil blend 1, kerosene, diesel fuel oil, benzene and oil. The pressures ranged from 0.103 to 20.6 MPa (15 to 2,990 pounds per square inch). The pipe diameters ranged from 16 to 140 mm (5/8 of an inch to 5 1/2 inches).

The Hughmark correlation is a function of the quality, density ratio of the two phases, mass flux and the viscosity of each phase. The correlation was used to predict the amount of refrigerant in three experimental studies (Kuijpers(1987), Otaki(1973), and Farzad (1990)) on refrigeration systems with fair agreement. These studies are discussed in further detail in Section 2.3 on experimental validation for refrigeration systems.

Perhaps the most valuable contribution from Hughmark's work is the dimensionless groups used in the void fraction correlation. Hughmark accounted for the forces which affect the slip ratio in the following dimensionless groups:

$$N_{Re} = \frac{D_a G}{(1 - \alpha_L) \mu_L + \alpha_G \mu_G}$$ \hfill (2.07)

The shear forces in the denominator of the Reynolds number are weighted by the void fraction. When there is a large amount of vapor present the viscosity of the gas governs the slip between the two-phases. As the amount of liquid increases (low void fractions) the viscosity of the liquid becomes more important. Calculating this dimensionless group is an iterative procedure since the void fraction is not known ahead of time.

$$N_{Fr} = \frac{\overline{U_H}^2}{g D_a} = \frac{1}{g D_a} \left( \frac{G \bar{\varepsilon}}{\beta \bar{\rho}_G} \right)^2$$ \hfill (2.08)

The Froude number represents the ratio of inertial to gravitational forces. The velocity in the Froude number is calculated assuming the velocities of the vapor and liquid are the same.

$$y_L = \frac{\dot{m}_L v_L}{\dot{m}_L v_L + \dot{m}_G v_G} \Rightarrow \frac{A_L}{A_c} \quad \text{(2.09)}$$
The volumetric fraction can be reduced to one minus the homogeneous void fraction if the velocities of the liquid and vapor are assumed equal. Surface tension forces were evaluated using a Weber number but were found not to be significant in the size pipes that were tested by Hughmark. It was defined as:

$$N_{we} = \frac{D_n \rho g U_H^2}{\sigma g_e}$$

(2.10)

2.2.3 Premoli

The Premoli (1971) correlation was developed for a large variety of conditions for two-phase mixtures flowing upward in vertical adiabatic channels. The channels consisted of different configurations with equivalent diameters ranging from 3 mm (.12 inches) to 51 mm (2 inches). The fluids included water/steam, liquid water/argon, acetone/argon and alcohol/argon. The pressures ranged from 2.8 to 6.9 MPa (400 to 1000 psi) for the water/steam data and .2 to 2.8 MPa (30 to 400 psi) for the other data. The correlation uses Reynolds and Weber (ratio of inertial to surface tension forces) numbers. Therefore, there is no mechanism to account for differences in gravitational forces. Gravitational forces are especially important in large diameter pipes where they promote a stratified flow.

2.2.4 Baroczy

The Baroczy method differs from the other three because it does not account for changes in mass flux. It is based on the Lockhart-Martinelli parameter. According to Tandon (1985), void fraction has a dependence on mass flux. Since this correlation does not have a mechanism to account for changes in mass flux it will not be studied further since in the headers of the condenser there was a significant change in mass flux as the refrigerant divides into the separate tubes. The flow visualization results demonstrate the significant effect this change in mass flux has on flow regime and consequently refrigerant inventory and slip ratio.
2.2.5 Tandon

Tandon (1985) developed a model for void fraction prediction by assuming the flow regime is annular. The model was compared with two different sets of data. The first set was obtained using the gamma ray absorption technique for steam-water during annular flow in a vertical tube with an inside diameter of 22 mm (.87 inch). The second set was collected on the boiling of heavy water in a vertical round duct with an inside diameter of 6.10 mm (.240 inches). The heated length of the duct was 2.5 m (8.20 feet).

The correlation is a function of the liquid Reynolds number and the Lockhart-Martinelli parameter. As with the Premoli correlation, gravitational forces are not accounted for in this correlation.

2.3 Experimental Validation for Refrigeration Systems

2.3.1 Kuijpers

The Kuijpers (1987) study concludes that for small refrigerator condensers the Premoli correlation produces the best agreement with experimental measurements. The experiments were conducted on a complete refrigeration system where the weights of the refrigerant in the heat exchangers were measured on-line.

For the condenser, the tests covered the range of mass fluxes from 55-75 kg/m²-s (11-15 lbm/ft²-s) at a condensing temperature of 45 °C (113 °F). There was a large degree of uncertainty in the data (~30 %) because the starting point of the subcooling region could not be determined.

2.3.2 Otaki

The Otaki (1973) study uses the Hughmark correlation to predict the amount of refrigerant in evaporators and condensers. The study concludes that the Hughmark correlation is within ten percent of the experimental results for a wide range of refrigeration machines. This study looked at 17 different units which included a
refrigerator, room air conditioners, packaged air conditioners, heat pumps, a bus air conditioner and a refrigeration unit for transportation.

Although these results look promising, they are by no means conclusive. The Otaki paper does not describe the method used to measure the weight of refrigerant during operation nor does it describe the range of test conditions that were used in the experiment.

2.3.3 Farzad

Well-documented experimental work for refrigeration systems has been done by Farzad (1990). Eight different correlations (five mass flux independent and three mass flux dependent) were used in a system simulation and compared with experimental test results at one set of environmental conditions for a three-ton air conditioner. The simulation and experimental results were compared using several different system variables (e.g. superheat leaving evaporator, sub cooling leaving condenser, refrigerant flow rate and capacity) over a range of refrigerant charge conditions from 20% undercharged to 20% overcharged. Out of the eight void-fraction correlations, the system model predicted these variables closest to the measured values with the Hughmark correlation. This was an overall system test and the weight of refrigerant in each of the components was not measured.

The condenser of this system consisted of tubes with an inside diameter of 8.05 mm (.317 inches), while the evaporator had tubes with an inside diameter of 8.81 mm (.347 inches). The environmental conditions were held at 27 °C (80 °F) dry bulb (DB) and 19 °C (67 °F) wet bulb (WB) for the inlet to the evaporator, and 28 °C (82 °F) DB and 14 °C (57 °F) WB for the inlet to the condenser.
2.3.4 Correlation Summary

The following table summarizes the relevant work dividing the literature into two categories: development and validation studies. Development studies are those where void fraction correlations were generated while validation studies are those where the existing correlations were applied to either a refrigeration system or other experimental data.

Table 2.02 Summary of Void Fraction Correlations

<table>
<thead>
<tr>
<th></th>
<th></th>
<th></th>
<th></th>
<th></th>
<th></th>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td>[mm]</td>
<td>[kg/m²-s]</td>
<td>[MPa]</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Development Studies</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Hughmark</td>
<td>air-water</td>
<td>1.025 [26.0]</td>
<td>4.35-458</td>
<td>36 [2]</td>
<td>Vertical</td>
<td>none</td>
<td>.5-1.0</td>
</tr>
<tr>
<td>This Study</td>
<td>R-134a</td>
<td>.029 [.74]</td>
<td>19-111 [93-541]</td>
<td>160-315 [1.1-2.2]</td>
<td>Horizontal</td>
<td>conden</td>
<td>1.0-.38</td>
</tr>
<tr>
<td>Validation Studies</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Hughmark</td>
<td>steam-water, air with oil blend1, kerosene, diesel fuel oil, benzene</td>
<td>.625-5.5 [16-140]</td>
<td>not avail</td>
<td>15 - 2990 [1-21]</td>
<td>Vertical and Horizontal</td>
<td>.02-.96</td>
<td></td>
</tr>
<tr>
<td>Tandon</td>
<td>water-steam</td>
<td>.87 [22]</td>
<td>not avail</td>
<td>Vertical</td>
<td>boil</td>
<td>.5-.95</td>
<td></td>
</tr>
<tr>
<td>Otaki</td>
<td>Refrigerants</td>
<td>not avail</td>
<td>not avail</td>
<td>not avail</td>
<td>both</td>
<td>1.0-0.</td>
<td></td>
</tr>
<tr>
<td>Farzad</td>
<td>R-22</td>
<td>.317 [8]</td>
<td>not avail</td>
<td>not avail</td>
<td>Horizontal w/ bends</td>
<td>both</td>
<td>1.0-0.</td>
</tr>
</tbody>
</table>
This summary shows that this study covered new territory especially in relation to the hydraulic diameter and fluid. The validation studies suggest the Hughmark and Premoli correlations as the most applicable for calculating refrigerant inventory in refrigeration systems. Out of these two correlations the Hughmark correlation is the most promising candidate since the Premoli correlation does not account for gravitational forces.

2.4 Flow Transition Predictions

Flow regime (configuration of the two phases) can have a significant impact on the slip ratio. In a stratified flow regime the slip ratio is much higher than in a mist flow where the velocities of the liquid and vapor are nearly the same. For this reason some method needs to be employed which is able to predict which flow regime will exist under given flow conditions.

Flow regime maps are one way to present information on two phase flow. These maps use a variety of coordinates to accomplish this task. Since there is no unifying theory that accurately predicts the flow regime, these maps continue to be used to document the phenomena. A few of the more common maps in use are presented below as well as one which approaches the conditions encountered in this study. Also, discussed are some theoretically based criteria which are used to predict certain flow regime transitions.

2.4.1 Mandhane Map

The Mandhane map (Mandhane et al. 1974) used a large data bank (5,935) of flow pattern observations. It was plotted using the superficial gas velocity (velocity of the gas if it were flowing alone in the channel) for the abscissa and the superficial liquid velocity for the ordinate on logarithmic coordinates. The map was developed from air-water data (1178 data points). The remaining data were then compared to this map and it
was found to predict the flow regime transitions of fluids with other properties with reasonable success.

Although Mandhane covered a wide range of fluids and geometry the current study did not fall within these ranges (see Table 2.03) The only property where the data falls within the range of the Mandhane data base was the viscosity of the gas. There was a slight overlap in the liquid density but for most of the operating conditions the density of R134a was higher than the densities contained in the Mandhane data base. The remaining properties, i.e. vapor density, viscosity of the liquid and surface tension at the vapor-liquid interface, also fall outside the Mandhane range.

<table>
<thead>
<tr>
<th>Study</th>
<th>Liquid Density (lbm/ft³)</th>
<th>Gas Density (lbm/ft³)</th>
<th>Liquid Viscosity (centipoise)</th>
<th>Gas Viscosity (centipoise)</th>
<th>Surface Tension (dynes/cm)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Current</td>
<td>62-71</td>
<td>3.3-7.5</td>
<td>0.14-0.17</td>
<td>0.015-0.016</td>
<td>2.6-5.9</td>
</tr>
<tr>
<td>Mandhane</td>
<td>44-63</td>
<td>0.05-3.2</td>
<td>0.3-90</td>
<td>0.01-0.022</td>
<td>24-103</td>
</tr>
</tbody>
</table>

The data were taken from experiments with pipes ranging in diameter from 13 to 170 mm (.5 to 6.5 in). The authors concluded the superficial velocities accurately accounted for any property and diameter effects for the range of geometry and fluids tested. For the range of conditions included in their data base this conclusion was justified. But for conditions outside this envelope Taitel-Dukler (1976) found this conclusion to be tenuous.

2.4.2 Fluid Property Adjustment

The kinetic energy of the gas significantly effects the flow regime. As the kinetic energy of the gas increases so does the interfacial shear stress. This increase in shear
stress promotes Kelvin-Helmholtz instabilities (these will discussed more fully in section 2.4.5) which are the precursors to slug, annular or mist flow.

Most flow regime maps have been developed from air/water systems. The density of vapor refrigerant is 40 to 100 times higher than the density of air. Therefore, the kinetic energy (see Equation 2.11) is higher if both fluids are traveling at the same velocity. Hanratty (1994) suggests one way to modify data with different vapor densities is to calculate the equivalent the superficial velocity of air to match the kinetic energy of the refrigerant vapor. For an incompressible fluid in irrotational flow the kinetic energy along a streamline is expressed as follows:

\[ KE = \frac{\rho U^2}{2} \]  

Equating the kinetic energies of air and refrigerant gas and solving for the velocity of the air results in :

\[ U_A = \left( \frac{\rho_g}{\rho_A} \right)^{\frac{5}{2}} U_G \]  

This velocity is the equivalent velocity of air required to produce the same kinetic energy as the refrigerant gas. With this adjustment, refrigerant data may be comparable to flow regime maps produced with air-water data.

2.4.3 Taitel-Dukler

While the Mandhane map is a compilation of empirical observations Taitel and Dukler (1976) proposed an analytically based map. They started from the integral form of the momentum equation for a stratified flow in an inclined pipe. They identified dimensionless groups based on theories for the mechanism of each flow regime transition. For example they used the Kelvin-Helmholtz stability criteria to determine the criteria for the transition from stratified to intermittent or annular-dispersed regimes.
The resulting flow map agrees with the Mandhane map. They went on to demonstrate that changes in fluid properties and pipe diameter and inclination can make significant shifts in the flow regime transition.

In 1983 Barnea et. al. recognized that the original Taitel-Dukler model was of limited application. Their experiments revealed the deviation between the original theory and the experiment became greater at the transition from stratified to intermittent as the diameter of a pipe was reduced. Assuming that surface tension forces become more important as pipe diameter decreases the criteria for this transition was modified to account for surface tension. The new theory was compared to experimental observations of air-water flowing through horizontal pipes ranging in diameter from 4 mm to 12.3 mm. It predicted the transition from stratified to intermittent with reasonable accuracy.

2.4.4 Damianides-Westwater

In 1988 Damianides and Westwater, recognizing the need for flow regime data in ever smaller pipes, published data for a compact heat exchanger and glass pipes ranging in diameter from 1 to 5 mm (.04 to .20 in.). Water and air were their working fluids. They observed that the stratified flow regimes disappeared in the 1 mm tube. Even though the latest Taitel-Dukler theory tries to account for surface tension when compared to these data there was poor agreement for all boundaries. Still the authors concluded that surface tension plays an important role in small tubes.

2.4.5 Kelvin Helmholtz Stability Criteria

The Kelvin-Helmholtz model can be used to predict the existence of instabilities in a stratified flow. These instabilities are the precursor to flow regime transitions in low viscosity fluids. The Kelvin-Helmholtz stability model assumes that the flow is inviscid and the velocity profiles are flat (Drazin and Reid, 1981). When the destabilizing
Bernoulli effect is counteracted by surface tension and gravity the result is the following neutral stability criteria:

\[
(Y_{\frac{\lambda}{\lambda}})^2 \rho_L \rho_g \coth \left( \frac{H_L}{\lambda} \right) \coth \left( \frac{H_G}{\lambda} \right) (U_L - U_G)^2 = \\
\left[ (Y_{\frac{\lambda}{\lambda}})^2 \rho_L - \rho_g + \sigma (Y_{\frac{\lambda}{\lambda}})^3 \right] \left( \rho_L \coth \left( \frac{H_L}{\lambda} \right) + \rho_G \coth \left( \frac{H_G}{\lambda} \right) \right)
\] (2.13)

The liquid height was calculated using the results from the Taitel-Dukler (1976) momentum analysis for stratified flow. Andritsos and Hanratty (1987) experimentally observed Kelvin-Helmholtz instabilities, in air-water systems, at lower gas velocities than the analysis predicts. It is likely that this holds true for R134a since its viscosity is lower than water. Therefore, this stability analysis is a conservative estimate of whether instabilities exist in R134a.
CHAPTER 3
EXPERIMENTAL FACILITY

From the survey of literature it is clear that there is a need for experimental work on small-channel condensers. The work that has been done on refrigeration systems (Kuijpers, Otaki and Farzad) has been limited in terms of range of operating conditions and inconclusive in terms of the contribution of individual components (i.e. condenser) to the overall system refrigerant inventory. This chapter describes the apparatus and techniques used to collect the experimental data. An analysis of the uncertainty in these data is also presented.

3.1 Flow Visualization Apparatus and Techniques

The test condenser was specially designed to examine the flow in the headers and condenser channels. The condenser channels were standard stock but the headers were custom made to accommodate sight plugs at various locations as shown in Figure 3.01. Light was introduced through fiber optic light guides. During each experiment the flow regimes in the headers and condenser tubes were recorded. To aid in the interpretation of the visual data a bore scope was used to magnify the image. For difficult to interpret data, a video camera/recorder was used to record the images for later evaluation. The combination of the bore scope, video camera and recorder is referred to as the visual recording system and is shown schematically in Figure 3.02.

At the time of the observations, the flow regimes predicted by other maps were not known providing an unbiased observation of the flow. After the observations were made, then the points were compared to other maps. For data with a video image, the video was reviewed without reference to the flow conditions or previous notations and a second assessment was made of the flow regime.
Figure 3.01 Flow Visualization Condenser

Figure 3.02 Visual Recording System
A special technique was used to determine the flow regime in the channel. Light from one of the fiber optic light sources was used to provide back-lighting for the channel of interest. The light intensity was recorded at the other end of the channel with the visual recording system. A shortcoming of this method is the uncertainty of how much the flow in the header affects the light intensity. In spite of this shortcoming, when the fluctuation in light intensity was compared with the visual observations of the structure of the flow exiting the channels they were in agreement. For example when the light intensity fluctuated either droplets or intermittent jets were observed at the exit of the channels.

3.2 Experimental Procedure

3.2.1 Calorimeter

The following schematic, Figure 3.03, illustrates the instrumentation and components used in the experimental apparatus for the condenser. The apparatus had the capability to control the fluid flow rates and inlet conditions. The water-cooled pre- and after-condensers provide the capability for partial condensing. These water-cooled condensers are a significant feature of the apparatus because they were able to maintain a constant known quality in the test condenser during the recording of flow visualization and refrigerant inventory data. A refrigerant turbulator was used at the entrance to the test section to promote thermodynamic equilibrium between the liquid and the vapor.

An air blender and honeycomb were installed in the air stream at the inlet to the test condenser in order to insure a uniform air velocity. A traverse of the air velocity before the test section confirmed that the velocity was uniform to within ±10% of the average. The air blenders at both the inlet and outlet were effective in reducing the temperature stratification. With the duct heater in operation, the stratification at the inlet could be as high as 28 °C (50 °F). After installing the air blenders this stratification was reduced to within 1 °C (2 °F).
Figure 3.03 Schematic of Experimental Apparatus for Testing Condensers
The most troublesome operating problem encountered with this apparatus was overheating the refrigerant evaporator. This evaporator consisted of electrical heaters wrapped around 19 mm (3/4") piping. When the power was increased the refrigerant flow rate would drop significantly. On a few occasions this drop in refrigerant flow rate was large enough that the heater overheated and shorted at the lead wires. The energy from this short was large enough to create a hole in the pipe.

The solution to this problem was to install a feedback control for the refrigerant pump which would increase the speed of the pump when the flow rate decreased. After this control was installed there were no more problems with overheating.

3.2.2 Data Acquisition and Processing

The data acquisition system was built upon the one described by Weber (1990) and Marin (1992) with the exception of additional instrumentation for measuring the heat transfer rates in the pre- and after-condensers and the weight measurement. Another addition was the equipment used to collect, store, digitize and process visual data.

Data were recorded by three multiplexers and a data logger once every 15 seconds. These data were then uploaded each 15 seconds to a desktop computer for storage and display. Except for the temperature measurements the data were stored as voltages. At the end of a day these data were uploaded to a workstation where the calibration information was applied to each voltage measurement and processed into the desired quantities (i.e. heat transfer rate, refrigerant inventory). The analysis program used to perform these calculations is listed in Appendix A. One of the outputs of the analysis program was an data file used as input to the model for the purpose of validation.

At the beginning of each day ten data strings were recorded before any equipment was turned on. The purpose of this was two-fold. First, these data could be used to adjust the offset for sensors where the zero would drift (i.e. differential pressure transducers, static pressure transducer and flow meters). Secondly, the calibration of the
refrigerant pressure and temperature measurements could be compared for agreement with saturation conditions. This comparison was made only if two-phase refrigerant was observed in the adjacent sight glasses.

3.2.3 Calibration and Uncertainty

The following section discusses the accuracy of the instrumentation and how their uncertainties propagate through to the desired results.

3.2.3.1 Heat Balance

A heat balance was performed between the refrigerant and the two water-cooled condensers of the apparatus. The apparatus was equipped with a pre- and after-condenser to study the refrigerant inventory at various qualities. These condensers were equipped with instruments (see Figure 3.03) to measure the rate of heat transferred to both water flow streams. The summation of the rate of heat added to the water streams was compared with the rate of heat removed from the refrigerant as shown in Figure 3.04. Excellent agreement, within ±5%, was obtained for all experiments where there was no air flow through the test section.

When this heat balance was first performed the heat transfer rate based on water measurements was systematically lower than that based on refrigerant measurements. This result was due to the additional heat losses to the environment from inadequately insulated refrigerant pipes. This heat loss was modeled using a one-dimensional (in the radial direction) heat transfer model accounting for conduction and convection heat transfer (See Equation 3.01). Since there was one layer of insulation on the pipes, the inside resistance was assumed to be insignificant compared to the sum of the insulation and the outside resistances. All the components between the temperature measurement at the inlet to the pre-condenser to the midpoint of the test condenser were included in the overall conductance (UA) calculation for the inlet control volume (See Figure 3.05).
Correspondingly, all the components from the midpoint of the test condenser to the midpoint of the receiver were included in the overall conductance calculation for the outlet control volume. Energy balances were performed on the control volumes to determine the unknown enthalpies at the inlet and outlet of the test condenser. Equations 3.02 and 3.03 are the result of these energy balances and show how the ambient losses were accounted for in the analysis software.

Figure 3.04 Heat Balance Between Refrigerant and Water
As testing proceeded the heat loss to the ambient was reduced by adding another layer of insulation to the piping. This was done in two stages on July 11 and July 20, 1994. Table 3.01 summarizes the overall conductances calculated for each of the control volumes at each of the stages of insulation.

Table 3.01 Overall Conductance Values for Heat Losses to Ambient

<table>
<thead>
<tr>
<th>Time Period</th>
<th>$U_{A_{in}}$ ($W/°C$)</th>
<th>$U_{A_{out}}$ ($W/°C$)</th>
<th>$U_{A_{in}}$ (Btu/hr-°F)</th>
<th>$U_{A_{out}}$ (Btu/hr-°F)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Before</td>
<td>2.469</td>
<td>12.104</td>
<td>4.681</td>
<td>22.943</td>
</tr>
<tr>
<td>7/11/94</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>7/11 to 17/94</td>
<td>2.469</td>
<td>7.170</td>
<td>4.681</td>
<td>13.591</td>
</tr>
</tbody>
</table>
In addition to identifying systematic errors, the heat balance analysis was effective in identifying and tracking down other problems with the instrumentation. One problem it uncovered was a thermocouple where the insulation was worn and a junction had formed just outside the sheath of the probe. That was a particularly difficult problem to find since the thermocouple did respond to changes in water temperature. Another problem was a leak in the tubing between the air flow nozzle and the differential pressure sensor. That caused the low pressure side of the sensor to read lower than it should, resulting in a higher air flow rate. That was also a difficult problem to track down since the sensor did give a reasonable response to changes in air flow rate. These problems were solved by replacing the temperature probe and sealing the leak thereby resulting in a good heat balance over the entire range of testing.

The heat balance between the air and refrigerant, however, was not as satisfactory (See Figure 3.06). This may be due to the temperature stratification in the outlet duct. The outlet air temperature consisted of the average of 9 thermocouples located within 1 meter (3.3 feet) of the test condenser. This array measured a stratification which ranged from 2 to 9.3 °C (3.6 to 16.8 °F). Another possible explanation of the systematic error in the air-side measurements is leakage from the air ducts. Although the duct was checked thoroughly for leaks and several were found, there may still exist some leaks which were undetectable. The aforementioned reasons and higher uncertainty in the air-side instruments (See section 3.2.3.2 for a detailed discussion of the components used to estimate these uncertainties) suggest that the refrigerant-side measurements were more reliable. Another fact which supports this conclusion was the excellent agreement obtained from the refrigerant to water heat balance.
3.2.3.2 Experimental Uncertainty

With any experiment there are errors associated with the measurements. Except during calibration, the actual error for any particular measurement is unknown since its actual value is unknown. However, a reasonable prediction of the range within which the actual value lies can be made by an uncertainty analysis.
The first step in this process was done by either calibrating (comparing the instrument output with a known input) or using the manufacturer's calibration for a given instrument. Table 3.02 summarizes the uncertainty of the instruments used in the experimental apparatus.

Copper-Constantan (Type T) thermocouples (TC) were used for many of the temperature measurements, otherwise RTD's were used. For the uncalibrated thermocouples the maximum possible uncertainty was used in the analysis. The uncertainties in the order of importance were attributed to the following causes:

- Thermocouple output (difference between actual output and standards published by NIST) - For a type T thermocouple in the 0 to 350 °C (32 to 662 °F) temperature range the special limit of error specified by the American National Standards Institute (ANSI) is ± 0.5 °C (0.9 °F) or ±0.4% of reading, whichever is greater. All of the uncalibrated thermocouples were ordered with special limits of error. The temperature reading would need to be above 125 °C (257 °F) before the ±0.4% error would apply. Since all the thermocouple readings were below this limit, the ±0.5 °C limit was used in the analysis.

- Reference junction measurement - A resistance temperature detector (RTD) was used to measure the temperature of an ice bath. This RTD was not calibrated so the manufacturer's uncertainty of ± 0.55 °C (1.0 °F) was used.

- Voltage measurement - The data logger has an uncertainty in voltage measurement of 0.1% of full scale. The 0-5 mV scale was used for the thermocouple readings. This converts to a temperature error of .11 °C (.2 °F).

- Thermocouple polynomials (difference between the NIST curves and the polynomials used in the data logger to convert the voltages to temperature) - The manufacturer of the data logger reports a limit of error of ± 0.001 °C (0.002 °F) when the measured temperatures are between -100 and 100 °C (-148 and 212 °F).
The polynomials were used to convert the reference temperature to a voltage and then the thermocouple output voltage to a temperature.

The uncertainties applied to uncalibrated thermocouples are summarized in Table 3.03.

<table>
<thead>
<tr>
<th>Variable</th>
<th>#</th>
<th>Abs Uncert.</th>
<th>Abs Uncert.</th>
<th>% Unc.</th>
<th>Source</th>
</tr>
</thead>
<tbody>
<tr>
<td>$T_{\text{in,cond,R}}$</td>
<td>7</td>
<td>0.32 °F</td>
<td>0.18 °C</td>
<td>—</td>
<td>Instrument Calibration Sheet Max Difference</td>
</tr>
<tr>
<td>$T_{\text{out,ac,R}}$</td>
<td>8</td>
<td>0.30 °F</td>
<td>0.17 °C</td>
<td>—</td>
<td>Instrument Calibration Sheet Max Difference</td>
</tr>
<tr>
<td>$T_{\text{in,nozzle}}$</td>
<td>11</td>
<td>1.0 °F</td>
<td>0.55 °C</td>
<td>—</td>
<td>Estimate from Omega Catalog</td>
</tr>
<tr>
<td>$P_{\text{out,superhr}}$</td>
<td>14</td>
<td>0.10 psi</td>
<td>0.69 kPa</td>
<td>—</td>
<td>Pressure Transducer Calibration Certificate</td>
</tr>
<tr>
<td>$P_{\text{in,cond,R}}$</td>
<td>15</td>
<td>—</td>
<td>—</td>
<td>0.025</td>
<td>Instrument Calibration Sheet</td>
</tr>
<tr>
<td>$P_{\text{out,ac,R}}$</td>
<td>16</td>
<td>0.16 psi</td>
<td>1.10 kPa</td>
<td>—</td>
<td>Pressure Transducer Calibration Certificate</td>
</tr>
<tr>
<td>$\Delta P_{\text{R}}$</td>
<td>17</td>
<td>0.025 psi</td>
<td>0.17 kPa</td>
<td>—</td>
<td>Setra Calibration Certificate</td>
</tr>
<tr>
<td>$P_{\text{out,pump}}$</td>
<td>18</td>
<td>0.17 psi</td>
<td>1.17 kPa</td>
<td>—</td>
<td>Pressure Transducer Calibration Certificate</td>
</tr>
<tr>
<td>$\dot{V}_{\text{R}}$</td>
<td>20</td>
<td>—</td>
<td>—</td>
<td>1</td>
<td>Manufacturer's Data</td>
</tr>
<tr>
<td>$\Delta P_{\text{nozzle}}$</td>
<td>21</td>
<td>0.025&quot; H20</td>
<td>6.2 Pa</td>
<td>—</td>
<td></td>
</tr>
<tr>
<td>$P_{\text{static,in,noz}}$</td>
<td>22</td>
<td>0.025&quot; H20</td>
<td>6.2 Pa</td>
<td>—</td>
<td></td>
</tr>
<tr>
<td>$T_{\text{in,cond,A}}$</td>
<td>58</td>
<td>2.1 °F</td>
<td>1.2 °C</td>
<td>—</td>
<td>Summation of Thermocouple Errors</td>
</tr>
<tr>
<td>$T_{\text{out,cond,A}}$</td>
<td>67</td>
<td>2.1 °F</td>
<td>1.2 °C</td>
<td>—</td>
<td>See Table 3.03</td>
</tr>
<tr>
<td>RH_{\text{in, A}}$</td>
<td>76</td>
<td>—</td>
<td>—</td>
<td>2</td>
<td>General Eastern Catalog</td>
</tr>
<tr>
<td>RH_{\text{out, A}}$</td>
<td>77</td>
<td>—</td>
<td>—</td>
<td>2</td>
<td>General Eastern Catalog</td>
</tr>
<tr>
<td>$m_{\text{R}}$</td>
<td>78</td>
<td>0.005 lbm</td>
<td>2.3 g</td>
<td>—</td>
<td>See Table 3.04</td>
</tr>
<tr>
<td>$T_{\text{out,superhr}}$</td>
<td>79</td>
<td>0.252 °F</td>
<td>0.14 °C</td>
<td>—</td>
<td>Applying worst error of the Thermocouples</td>
</tr>
<tr>
<td>$T_{\text{out,cond,R}}$</td>
<td>85</td>
<td>0.18 °F</td>
<td>0.10 °C</td>
<td>—</td>
<td>Instrument Calibration Sheet - Used Max Diff of 4 pts</td>
</tr>
<tr>
<td>$T_{\text{in,pc,w}}$</td>
<td>86</td>
<td>0.144 °F</td>
<td>0.08 °C</td>
<td>—</td>
<td>Instrument Calibration Sheet - Used Max Diff</td>
</tr>
<tr>
<td>$T_{\text{out,pc,w}}$</td>
<td>87</td>
<td>0.198 °F</td>
<td>0.11 °C</td>
<td>—</td>
<td>Instrument Calibration Sheet - Used Max Diff of 3 pts</td>
</tr>
<tr>
<td>$T_{\text{in,ac,w}}$</td>
<td>88</td>
<td>0.252 °F</td>
<td>0.14 °C</td>
<td>—</td>
<td>Instrument Calibration Sheet - Used Max Diff of 3 pts</td>
</tr>
<tr>
<td>$T_{\text{out,ac,w}}$</td>
<td>89</td>
<td>0.126 °F</td>
<td>0.07 °C</td>
<td>—</td>
<td>Instrument Calibration Sheet - Used Max Diff of 3 pts</td>
</tr>
<tr>
<td>$\dot{V}_{\text{pc,w}}$</td>
<td>90</td>
<td>—</td>
<td>—</td>
<td>0.436</td>
<td>Turbine Flow Manuals</td>
</tr>
<tr>
<td>$\dot{V}_{\text{ac,w}}$</td>
<td>91</td>
<td>—</td>
<td>—</td>
<td>0.825</td>
<td>Turbine Flow Manuals (Took Avg. of Max and Min)</td>
</tr>
<tr>
<td>$P_{\text{amb}}$</td>
<td>92</td>
<td>0.02953 &quot;Hg</td>
<td>100 Pa</td>
<td>—</td>
<td>Calibration Certificate - Took error at 101.3 kPa</td>
</tr>
<tr>
<td>$D_{\text{nozzle}}$</td>
<td>94</td>
<td>0.01 &quot;</td>
<td>0.25 mm</td>
<td>—</td>
<td>Calipers</td>
</tr>
</tbody>
</table>
Table 3.03  Summary of Uncalibrated Thermocouple Uncertainty

<table>
<thead>
<tr>
<th>Source of Error</th>
<th>Error (°C)</th>
<th>Error (°F)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Reference Temperature</td>
<td>0.55</td>
<td>1.0</td>
</tr>
<tr>
<td>TC Output</td>
<td>0.5</td>
<td>0.9</td>
</tr>
<tr>
<td>Voltage Measurement</td>
<td>0.11</td>
<td>0.2</td>
</tr>
<tr>
<td>Polynomial Curve Fit</td>
<td>0.002</td>
<td>0.004</td>
</tr>
</tbody>
</table>

The measured heat transfer rates were calculated based on temperatures, pressures and flow rates. Since the output of these instruments were only used for the calculation of heat transfer when the fluids were in the single phase region then they were independent measurements. Therefore, the propagation of the uncertainty to the desired result can be estimated by Equation 3.04.

\[
u_o = \sqrt{u_1^2 + u_2^2 + u_3^2 + \ldots + u_n^2}
\]  

(3.04)

The impact on the heat transfer rate was determined in the analysis program. The numerical derivative was calculated for each variable used in the calculation of the desired output. That was then multiplied by the appropriate uncertainty from Table 3.02. These uncertainties were then used in Equation 3.04 to determine the overall uncertainty. The results of this analysis are contained in the uncertainty bars displayed in Figure 3.06.

3.2.3.3 Application Uncertainty

The test section was divided into five different regions for modeling purposes: 1) inlet and outlet pipe, 2) inlet header, 3) condenser tubes, 4) outlet header and 5) stub. Figure 4.01 is a schematic of the condenser from which these regions can be discerned. Regions 1 and 5 are unique to the test stand. The uncertainty associated with these two regions and its impact on the refrigerant inventory in the condenser was estimated.

To predict the uncertainty in the pipes the output of the final model was compared to the output of the homogeneous model (which assumes there is no slip between the
vapor and the liquid). This was an extreme assumption for the lower limit since flow was either wavy or wavy-annular (based on observations through sight glasses (see Appendix D) in these pipes. There can be a significant amount of slip between fluids in these flow regimes. So this analysis provides the maximum uncertainty in the pipes with a high probability that the actual uncertainty will be significantly lower. The uncertainty was calculated for the range of pressures (psia), qualities and mass flow rates (lbm/hr) shown in Figure 3.07. In SI units the pressure ranged from .69 to 3.4 MPa and the mass flow rate ranged from 45 to 272 kg/hr. These values exceed the range of operating conditions expected for condensers used in mobile air conditioning applications.

The uncertainty attributed to the stub was determined experimentally. The refrigerant was heated to superheated conditions and then passed through an adiabatic test section. The model assumed the stub was full of liquid and the remainder of the test section was superheated vapor. There were two experimental data points eliminated from the data set because the superheat was high enough to evaporate some of the liquid in the stub. All the points with superheat values less than 14°C (25°F) were included in the uncertainty analysis. The model agreed with the experiment within 1.8 grams (.004 lbm) with an average error of .46 grams (.001 lbm) for 6 data points which covered mass flow rates of 47-133 kg/hr (104-294 lbm/hr). When this uncertainty was added to the uncertainty in the pipes it did not noticeably increase the overall application uncertainty as demonstrated in Figure 3.08.
Figure 3.07 Application Uncertainty Due to Pipes Alone
3.2.3.4 Weight measurements

The uncertainty in the weight measurement was attributed to the following five factors: 1) the difference in closing time of valves, 2) the load cell, 3) density of the vapor remaining in the test section, 4) leakage of refrigerant from the test section during extraction and 5) refrigerant loss in the dead space between the stub and sampling cylinder shut-off valves. These factors were estimated and are summarized in Table 3.04
Table 3.04 Summary of Uncertainty in Weight Measurement

<table>
<thead>
<tr>
<th>Cause of Uncertainty</th>
<th>Estimated Uncertainty (grams)</th>
<th>(lb_m)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Closing time difference</td>
<td>1.8</td>
<td>.004</td>
</tr>
<tr>
<td>Load Cell</td>
<td>.84</td>
<td>.00185</td>
</tr>
<tr>
<td>Vapor Remaining</td>
<td>.59</td>
<td>.0013</td>
</tr>
<tr>
<td>Leakage</td>
<td>.05</td>
<td>.0001</td>
</tr>
<tr>
<td>Dead Space</td>
<td>.09</td>
<td>.0002</td>
</tr>
<tr>
<td>Total Mean Error</td>
<td>2.3</td>
<td>.005</td>
</tr>
</tbody>
</table>

3.2.4 Test Envelope

An important part of an experimental program is to determine which independent variables are important. Many flow maps of two phase data are constructed with superficial velocities of the two phases as the axes. For a given fluid and geometry the transitions from one flow regime to another is a function of these two velocities. Therefore, these velocities are important variables.

Also, most flow maps were developed under adiabatic conditions. Since there is heat transfer occurring in the heat exchanger, a third variable, the heat flux, is important. Therefore, the three variables which were varied in the test plan were the superficial velocity of the vapor, the superficial velocity of the liquid and the heat flux.

The superficial velocities are related to measurable variables in the experiment by the following relationships.

\[
U^s_G = \frac{\dot{m}_x}{\rho_G A}
\]

\[
U^s_L = \frac{\dot{m}(1 - x)}{\rho_L A}
\]

(3.05)

Since the headers are essentially adiabatic the change in heat flux only needs to be applied to the condenser tubes. The abbreviation "N/A" was used to indicate not applicable in the table.

The ranges of the three independent variables covered in the experiments are contained in Table 3.05.
Table 3.05 Test Envelope Based on Superficial Velocities and Heat Flux

<table>
<thead>
<tr>
<th>Variable</th>
<th>Inlet Header</th>
<th>Condenser Tube Exit</th>
<th>Outlet Header</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>[SI]</td>
<td>(IP)</td>
<td>[SI]</td>
</tr>
<tr>
<td>Superficial Liquid Velocity</td>
<td>0-.23</td>
<td>0-.75</td>
<td>0-.48</td>
</tr>
<tr>
<td>[m/s] (ftls)</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Superficial Gas Velocity</td>
<td>.13-1.92</td>
<td>.43-6.31</td>
<td>0-4.06</td>
</tr>
<tr>
<td>[m/s] (ft/s)</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Heat Flux [kW/m²] (Btu/hr-ft²)</td>
<td>N/A</td>
<td>N/A</td>
<td>0-11.7</td>
</tr>
</tbody>
</table>

The range of testing can also be expressed in terms of operating variables. Table 3.06 contains a summary of these variables for the adiabatic tests while Table 3.07 does so for the heat transfer tests.

Table 3.06 Test Matrix Based on Operating Variables for Adiabatic Tests

<table>
<thead>
<tr>
<th>Variable at Inlet</th>
<th>Range [SI units]</th>
<th>Range (IP units)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Refrigerant Mass Flow Rate [grams/s] (lbm/hr)</td>
<td>13.1-73.4</td>
<td>104-582</td>
</tr>
<tr>
<td>Refrigerant Pressure [kPa] (psia)</td>
<td>1101-2173</td>
<td>160-315</td>
</tr>
<tr>
<td>Superheat Temperature [°C] (°F)</td>
<td>0-31.3</td>
<td>0-56.4</td>
</tr>
<tr>
<td>Quality [-] (-)</td>
<td>.044-.904</td>
<td>.044-.904</td>
</tr>
</tbody>
</table>

Table 3.07 Test Matrix Based on Operating Variables for Heat Transfer Tests

<table>
<thead>
<tr>
<th>Variable at Inlet</th>
<th>Range [SI units]</th>
<th>Range (IP units)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Refrigerant Mass Flow Rate [grams/s] (lbm/hr)</td>
<td>30.6-34.3</td>
<td>243-272</td>
</tr>
<tr>
<td>Refrigerant Pressure [kPa] (psia)</td>
<td>1350-1730</td>
<td>196-251</td>
</tr>
<tr>
<td>Superheat Temperature [°C] (°F)</td>
<td>0-27.0</td>
<td>0-48.6</td>
</tr>
<tr>
<td>Inlet Quality [-] (-)</td>
<td>.446-.631</td>
<td>.446-.631</td>
</tr>
<tr>
<td>Air Mass Flow Rate [grams/s] (lbm/hr)</td>
<td>120-277</td>
<td>951-2200</td>
</tr>
<tr>
<td>Air Inlet Temperature [°C] (°F)</td>
<td>30.5-37.3</td>
<td>87.0-99.1</td>
</tr>
</tbody>
</table>

The model developed in this study is only applicable within this range of operating conditions which were chosen to cover the range of operating conditions found in mobile air conditioning applications. The model may be applicable beyond these...
ranges of operating conditions since it is based on non-dimensional groups but further experimental testing would need to be done to verify any extension of the model.
CHAPTER 4
SIMULATION DESCRIPTION

As mentioned in the introduction, the refrigerant inventory for vapor compression systems is determined using a costly trial and error procedure. An accurate computer model that predicts the amount of refrigerant in a system would reduce the time and expense of this process. The following chapter presents a computer model with the capability to predict the amount of refrigerant in air-cooled condensers. To accurately predict refrigerant inventory, two important steps are performed. First the heat transfer of the coil is modeled. A modular approach is used to divide the coil into relatively small increments. Second, the slip ratio ($S$) is predicted throughout the condensing region.

The condenser simulation program used for this study was originally developed by Ragazzi (1991). Orth (1993) added refrigerant inventory prediction capabilities to the model. It is based on first principles and is general enough to use with condenser coils of various geometry and for flows of different refrigerants. The first-principles model is programmed in FORTRAN and runs on Apollo workstations. The model along with details on input, output and data structures are in Appendix B.

The simulation program divides the condenser coil into a user-specified number of segments. Segments are defined as sections of a coil which share the same cross-sectional area, mass flow rate of refrigerant and mode of heat transfer (adiabatic or forced convection). Segments in the condensing region are further divided into a number of modules. Each module is treated as an individual heat exchanger. The local refrigerant side heat transfer coefficients are determined based on the average between the module inlet and outlet conditions. The inlet conditions to the first segment are provided by the user through an input file and Newton-Raphson iteration (Ragazzi 1991) is performed to determine the outlet conditions for each module in the segment. The governing equations for each module are based on the conservation of energy and momentum.
The following operating conditions at the condenser coil inlet must be specified for the simulation: the refrigerant pressure, temperature, enthalpy and mass flow rate and the air pressure, temperature, relative humidity and mass flow rate. On the air side, the mass flow rate for each module is determined by a weighting function (the ratio of the length of the module to the total length of the condenser). This weighting function represents a ratio of the air flow area for the module to the total air flow area and assumes that the refrigerant tubes are equally spaced.

4.1 Coil Geometry

For this study, the simulation was used to model a cross-flow heat exchanger, with parallel refrigerant paths. The primary application for this type of coil is automotive air conditioning systems. The circulating refrigerant was ozone-safe R134a. Figure 4.01 below demonstrates how the condenser was divided into segments. The refrigerant enters the first segment through a manually-operated ball-valve. It enters the second segment through a short horizontal pipe in the same configuration as condensers used in industry. The inlet header was divided into 12 segments since the mass flow rate decreases in the downward direction after refrigerant enters each of the parallel-tubes. These tubes were modeled in segment 14 which subdivides the tubes into several modules for accurate calculation of heat transfer. The total mass flow rate was equally divided between refrigerant channels which branch off the inlet header. The outlet header was also divided into 12 segments to account for the increase in mass flow rate as each tube discharges its contents into the header. The refrigerant then exits through the 27th segment. The 28th segment was different from the others because during normal operation no refrigerant flows through this segment. This segment was referred to as the stub. When the condenser was in operation this stub was assumed to be filled with liquid. A sampling cylinder was attached to the stub to extract the refrigerant after it had been trapped.
The pipes (segments 1, 27 and 28) and manifolds were assumed to be adiabatic. However, the pressure drop across these segments were included in the simulation.

4.2 Modeling Equations

The Newton-Raphson variables for the simulation are the outlet enthalpy \( h_{\text{out}} \), the outlet pressure \( p_{\text{out}} \) for each module (see figure 4.02). For coils with cross-counter-flow geometry (see figure 4.03) the average outlet air temperature from the modules is required as a residual equation. This coil geometry occurs when the refrigerant tube is routed from the front to the back of the coil with a return bend and the refrigerant inlet to the tube is located in the back. In this situation, the average air outlet temperature from the front tube is the inlet condition for the back tube. For this case the tube is modeled as
one segment using an equal number of modules in both the front and rear portions of the


Air Outlet
temperature
pressure
humidity

Refrigerant Inlet
pressure
enthalpy
flow rate

pressure drop
rate of heat transfer

Refrigerant Outlet
pressure
enthalpy

Air Inlet
temperature
pressure
humidity
flow rate

Figure 4.02 Typical Module

Figure 4.03 Cross-Counter-flow Geometry

The residual equations for each module in the Newton-Raphson iteration are given by:

\[
\text{Res}(1) = h_{R, out} - h_{R, in} - \frac{\dot{Q}_{\text{mod}}}{m_{R, mod}}
\]

(4.01)

\[
\text{Res}(2) = P_{R, out} - P_{R, in} - \Delta P_{\text{mod}}
\]

(4.02)

\[
\text{Res}(3) = \overline{T}_{A, in, back} - \overline{T}_{A, out, front}
\]

(4.03)

The rate of heat transfer from the module (\(\dot{Q}_{\text{mod}}\)) is calculated using the
effectiveness-NTU method while the pressure drop is calculated by a pressure drop
correlation. The details of these calculations follow in sections 4.3 and 4.4.
The thermodynamic property routines were developed by the National Institute of Standards and Technology (NIST). The NIST routines employ the 32-term Modified Benedict-Webb-Rubin (MBWR) equation of state to calculate refrigerant properties. Air properties are calculated using psychrometric routines from the American Society of Heating, Refrigerating and Air Conditioning Engineers (ASHRAE).

For transient modeling the steady-state algebraic equations become differential equations as shown below in equations 4.04-06. Solution of the transient model consists of the simultaneous numerical integration of these equations over some time period from a specified initial condition.

\[
\frac{dE}{dt} = \dot{m}_R (h_{R, in} - h_{R, out}) - \dot{Q} \quad (4.04)
\]

\[
\frac{dm}{dt} = \dot{m}_{R, in} - \dot{m}_{R, out} \quad (4.05)
\]

\[
\frac{1}{A} \frac{d(pp)}{dt} = (P_{R, in} - P_{R, out}) - \Delta P_{fric} - \Delta P_{mom} - \Delta P_{grav} \quad (4.06)
\]

Additional equations that constrain the system at each time step are shown in equations 4.07-09. They are the definitions of stored energy, mass, and momentum, respectively, for a module. In an actual heat exchanger, energy is stored both in the refrigerant and the condenser wall. However, in this model all the energy was considered to be on the refrigerant side of the heat transfer in Equation 4.07.

\[
E = \dot{m}_{R} \bar{c}_R + \frac{C_p T_{surf}}{L} \quad (4.07)
\]

\[
m = A \bar{\rho} \quad (4.08)
\]

\[
pp = m \left( \frac{\dot{m}}{\rho A} \right)_{avg} \quad (4.09)
\]

Equations 4.07-09 are the static equations and are solved by a Newton-Raphson method at each time step of the transient solution.

There are six unknowns (i.e., \( \dot{m}, P, h, E, m, \) and \( pp \)) in equations 4.04-09. For steady state initialization, the derivatives in equations 4.04-06 are set to zero to solve for \( \dot{m}, P, \) and \( h \). Then the definitions in equations 4.07-09 allow \( E, m, \) and \( pp \) to be directly
calculated. An implicit numerical integration is implemented in the simulation. The implicit method approximated the derivatives at the future time so that all six equations are solved together.

4.3 Heat Transfer

The effectiveness-NTU method, as described in conventional heat transfer texts, is used in the condenser simulation to determine the heat rejected by each module ($\dot{Q}_{\text{mod}}$). The general form of the equation was:

$$\dot{Q}_{\text{mod}} = eC_{\text{min}}(T_{Rin} - T_{Ain}) \quad (4.10)$$

The effectiveness requires the calculation of the overall heat conductance (UA) given by:

$$UA = \frac{1}{\frac{1}{th} + \frac{1}{\eta_R htc_R A_{si}}} \quad (4.11)$$

A two-dimensional finite difference program was written to model the webs inside the condenser tubes to predict the surface efficiency of the refrigerant-side surface. The surface efficiency, as a function of refrigerant-side heat transfer coefficient, see equation 4.12, was included in the simulation.

$$\eta_R = 0.99559 - (0.00033913 \times htc_R) \quad (4.12)$$

Where the heat transfer coefficient is given in Btu/hr-°F-ft². Ragazzi (1991) contains further details about how the effectiveness-NTU method was applied in this simulation.

The air side heat transfer coefficient ($htc_A$) used in the simulation program is based on the Colburn j-factor. The j-factor correlation was determined for the condenser coil using experimental data and an optimization program. This program estimates the parameters for the correlation by minimizing the error between the model and a few
selected experimental data (see Figure 6.05). For the coil used in this study, this 
correlation was found to be,

\[ j = 1.008 \text{Re}_{\text{air}}^{-0.406} \tag{4.13} \]

Experimental points 112, 114, 119, 120 and 123 were used in the optimization. See 
Appendix D for the table containing the operating conditions for these experiments.

The calculation of the local refrigerant side heat transfer coefficient \( (htc_R) \) is 
dependent on the refrigerant phase. In the single-phase region of the condenser, the 
Dittus-Boelter correlation is used to determine the refrigerant heat transfer coefficient for 
the module. The general form of this correlation for cooling is given by:

\[ Nu_R = 0.023 \times \text{Re}_R^{0.8} \times \text{Pr}_R^{0.3} \tag{4.14} \]

For the two-phase region the Dobson (1993) correlation is used. Dobson found 
that the flow regime had a significant effect on the mechanisms for heat transfer. In the stratified and wavy flow regimes, heat transfer is governed by Nusselt-type condensation while in the annular flow regime the heat transfer is governed by forced convection.

In the stratified and wavy flow regimes the heat transfer is modeled as follows:

\[ Nu_L = 0.375 \frac{G a P r}{X_a^{0.25}} \frac{J a}{J a} L \tag{4.15} \]

where,

\[ Nu_L = \frac{h D_L}{k_L} \tag{4.16} \]

\[ G a = \frac{\rho_L (\rho_L - \rho_G) g D_h^2}{\mu_L} \tag{4.17} \]

\[ P r = \frac{h L c_{p,L}}{k_L} \tag{4.18} \]

\[ J a = \frac{c_{p,L} (T_{\text{sat}} - T_w)}{h_{LG}} \tag{4.19} \]

In the annular flow regime the correlation was a function of the Lockhart-Martinelli correlating parameter \( X_{tt} \). Its general form was given by:
\[ \text{Nu}_{L} = 0.023 \text{Re}_{L}^{0.8} \text{Pr}_{L}^{0.4} \left[ 1 + \frac{2.22}{X_{n}^{0.889}} \right] \] (4.20)

where,
\[ X_{n} = \left[ \frac{\rho_{G}}{\rho_{L}} \right]^{0.5} \left[ \frac{\mu_{L}}{\mu_{G}} \right]^{-0.1} \left[ \frac{1 - x_{\text{avg}}}{x_{\text{avg}}} \right]^{0.9} \] (4.21)

Dobson (1993) specifies the following criteria to distinguish between the two flow regimes. The annular flow correlation (Equation 4.20) should be used whenever the Wallis dimensionless gas velocity exceeds 1.8. For values lower than this the stratified flow correlation (equation 4.15) should be used. The Wallis dimensionless gas velocity is given by Equation 4.22.
\[ j_{G}^{*} = \frac{Gx}{\sqrt{gD \rho_{G}(\rho_{L} - \rho_{G})}} \] (4.22)

Note the heat transfer equations were developed for condensation in horizontal tubes with inside diameters ranging from 3.14 to 7.04 mm (.124 to .277 inches). The hydraulic diameter of the channels tested in this study was only .74 mm (.029 inches). Therefore, validation of these correlations will extend their range of application.

4.4 Pressure Drop

In general, there are three components for the module pressure drop: frictional pressure drop, momentum pressure drop and gravitational pressure drop.

\[ \Delta P_{\text{mod}} = \Delta P_{\text{fric}} + \Delta P_{\text{mom}} + \Delta P_{\text{grav}} \] (4.23)

For condenser tubes, the frictional pressure drop and the momentum pressure drop were determined for each module individually. The frictional pressure drop in the single phase region was found using the Fanning friction factor and in the two phase region using the Souza et.al. (1993) correlation. For the gravitational pressure drop, the total elevation change in the condenser was used to determine an overall gravitational pressure drop. This was then multiplied by the weighting function \( L_{\text{mod}} / L_{\text{cond}} \) to provide the
gravitational pressure drop for each module. The momentum pressure drop was obtained through a control volume analysis over the module (Ragazzi 1991). For the other piping components (i.e. pipes and headers) the procedure outlined by Paliwoda (1992) was used.

4.5 Prediction Of Inventory

The total mass of the condenser is given by:

\[
m_{\text{total}} = \sum_{i=1}^{\# \text{seg}} \left( \# \text{chan}_{\text{seg}(i)} \sum_{i=1}^{\# \text{mod}} m_{\text{mod}(i)} \right) \tag{4.24}
\]

In the condensing region equations 2.01 and 2.02 were approximated by dividing the heat exchanger into finite lengths or modules (\(\Delta l_i\)) and summing over the length of interest. Equation 4.25 results from the summation of these two equations after they have been discretized. The bar over the densities and void fraction indicates these values are averaged in the axial direction. In other words, the inlet and outlet values of a given module are averaged together. This assumes the inventory for each module was represented by the mean value for the module. The error in this assumption is eliminated in the limit as \(\Delta l_i\) approaches zero. Practically, however, the error becomes insignificant at some small finite value of \(\Delta l_i\). This value was determined by increasing the number of modules in the condensing segment (#14) until there was negligible change in the result (see Figure 4.06).

\[
m_1 = A \left[ \sum_{i=1}^{\# \text{mod}} \bar{p}_g \bar{\alpha} \Delta l_i + \sum_{i=1}^{\# \text{mod}} \bar{p}_L (1 - \bar{\alpha}) \Delta l_i \right] \tag{4.25}
\]

where,

\[
\bar{\alpha} = \frac{1}{1 + \left( \frac{1 - \bar{x}}{\bar{x}} \right) \frac{\bar{p}_g S}{\bar{p}_L}} \tag{4.26}
\]
The slip ratio \((S)\) is determined in the model using the same correlating form as Hughmark. Hughmark used a variable \((K_H)\) which is defined as the inverse slip ratio.

\[
K_H = \frac{1}{S}
\]  

(4.27)

\(K_H\) is a function of the correlating parameter \(Z\). These values are tabulated in Table 4.01. In the model a linear interpolating algorithm was used to calculate \(K_H\).

<table>
<thead>
<tr>
<th>(Z)</th>
<th>(K_H)</th>
<th>(Z)</th>
<th>(K_H)</th>
<th>(Z)</th>
<th>(K_H)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1.3</td>
<td>0.185</td>
<td>5.0</td>
<td>0.675</td>
<td>20</td>
<td>0.83</td>
</tr>
<tr>
<td>1.5</td>
<td>0.225</td>
<td>6.0</td>
<td>0.72</td>
<td>40</td>
<td>0.88</td>
</tr>
<tr>
<td>2.0</td>
<td>0.325</td>
<td>8.0</td>
<td>0.767</td>
<td>70</td>
<td>0.93</td>
</tr>
<tr>
<td>3.0</td>
<td>0.49</td>
<td>10</td>
<td>0.78</td>
<td>130</td>
<td>0.98</td>
</tr>
<tr>
<td>4.0</td>
<td>0.605</td>
<td>15</td>
<td>0.808</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

The correlating parameter \(Z\) is given by,

\[
Z = \frac{\text{Re}^{p1} F_r^{p2}}{y_L^{1/4}}
\]  

(4.28)

where,

\[
\text{Re} = \frac{D_h G_r}{\bar{\mu}_L + \bar{\alpha}(\bar{\mu}_G - \bar{\mu}_L)}
\]  

(4.29)

\[
F_r = \frac{1}{g D_h} \left[ \frac{G_r \bar{x}}{\beta \bar{\rho}_G} \right]^{2}
\]  

(4.30)

\[
y_L = 1 - \beta
\]  

(4.31)

\[
\beta = \frac{1}{1 + \left[ \frac{1 - \bar{x}}{\bar{x}} \right] \frac{\bar{\rho}_G}{\bar{\rho}_L}}
\]  

(4.32)

This form of the correlation is used to model the pipes and inlet header. For the condenser channels the Froude number is replaced with the Weber number as defined in Equation 2.10. The outlet header is modeled as an unsteady flow process. The details of
this model are contained in Section 6.4. In the outlet pipe when the refrigerant was subcooled the void fraction was assumed to be .5 since the pipe had a downward slope and the Froude number was less than one.

The parameters for each of the dimensionless groups in the correlation were searched for by minimizing the error between the experiment and the model for a selected subset of the data. This procedure is described more fully in section 6.3 along with the results of the search.

4.6 Model Uncertainty

The number of modules used in the simulation has an effect on the output. The fewer the number of modules the greater the uncertainty in the result. This happens because certain quantities (e.g. slip ratio, heat transfer coefficient and frictional pressure drop) were based on the average properties of a module. The larger the module the farther these average properties deviate from the actual properties within the module. To determine the number of modules necessary for accurate results, a series of runs were performed for a given set of operating conditions. The "correct" solution for the set of operating conditions was assumed to be the run with 114 modules. This was chosen as the correct solution because the variation in the outputs were negligible from 50 modules and above and there was no apparent round off error. To calculate the error the output value of interest for this "correct" solution was subtracted from the value for any given run.

The operating conditions were selected so the refrigerant passes through the full condensation process in the heat exchanger. Table 4.02 contains the set of operating conditions used in the simulation. A higher than normal air flow rate was used to provide high heat fluxes. The higher the heat flux the worse the error will be for a given size module.
Table 4.02 Operating Conditions Used to Check the Effect of Number of Modules

<table>
<thead>
<tr>
<th>Refrigerant Inlet</th>
<th>Mass Flow Rate [kg/hr] (lbm/hr)</th>
<th>Temperature [°C] (°F)</th>
<th>Pressure [kPa] (psi)</th>
<th>Relative Humidity [%]</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>90.1 (200)</td>
<td>60 (140)</td>
<td>1379 (200)</td>
<td>Not Applicable</td>
</tr>
<tr>
<td>Air Inlet</td>
<td>1818 (4000)</td>
<td>21 (70)</td>
<td>101 (14.7)</td>
<td>50</td>
</tr>
</tbody>
</table>

The output variables of interest for heat transfer are the outlet subcooling and rate of heat transfer. Figures 4.04 and 4.05 contain the results of the effect of the number of modules on these two variables. As few as 12 modules could be used to predict the subcooling to within 1°C but more than 50 modules would be needed to predict subcooling to within 0.1°C. Likewise, for the total heat transfer as few as 12 modules could predict the total heat transfer rate to within 0.9% while more than 50 modules would be necessary to predict this value within 0.1%.

Figure 4.04 Effect of Number of Modules on Error in Subcooling
Figure 4.05 Effect of Number of Modules on Error in Heat Transfer

Figure 4.06 Effect of Number of Modules on Error in Refrigerant Inventory
Figure 4.07 Effect of Number of Modules on Error in Pressure Drop

The uncertainty in the refrigerant inventory is presented in Figure 4.06. For an accuracy within 1.2% as few as 12 modules may be used. To obtain an accuracy within .2% then more than 50 modules will be needed.

Figure 4.07 demonstrates the effect of the number of modules on the predicted pressure drop. If errors within 5% are acceptable then as few as 12 modules may be used. At least 50 modules would be needed to provide accuracy within 1%.

In conclusion 12 modules appear to be sufficient to model a condenser within experimental error. If better precision is desired then 50 or more modules are recommended. These conclusions are based on operating conditions specified in Table 4.02. If the conditions exceed the values used here it is recommended that the exercise presented in this section be performed under the new set of conditions.
CHAPTER 5

FLOW VISUALIZATION RESULTS

Flow regime transitions are an important part of predicting refrigerant heat transfer and inventory. Dobson(1993) modeled the refrigerant side heat transfer as a function of the flow regime. The slip ratio is also a function of flow regime therefore, knowing which flow regimes are present is an important part of the modeling process.

5.1 Inlet Header

The inlet header provided interesting information which was helpful in qualitatively accessing the accuracy of the simulation (see Section 6.3.3). The flow visualization data were collected through six sight plugs which were installed in the inlet header (See Figure 3.01). To plot the observations, the superficial velocities were calculated assuming equal mass flow rates through each of the condenser tubes.

The flow regime data for three sections of the inlet header are documented in Figure 5.01. These data are contained in three separate bands all labeled at the tail of the data. The reason the data form three separate bands was because the mass flow rate decreases as the refrigerant travels down the header. (The assumed mass flow rate for each band as a fraction of the total mass flow rate was as follows: top=1, sight plug 5(lower middle region)=3/11 and bottom=1/11). On these plots a line representing condensation will begin at the ordinate axis and follow a curve similar in shape to one of the bands of data and end at the abscissa axis of the plot.

This flow regime plot reveals that the top portion of the header the dispersed liquid regime occurred at low qualities (or high superficial liquid velocities). As the superficial velocity of the vapor was increased mist flow became more prevalent. The vapor velocity was adjusted so this map could be compared with air water data. This adjustment is described in section 2.4.2.
Figure 5.01 Flow Map for Top of Inlet Header (Sight Plug 2)

For comparison, Barnea et al. (1982) presented data for a 2.5 cm pipe in the vertical downward flow arrangement. They observed annular flow at all superficial vapor velocities below .6 m/s. They did not mention whether or not there was any significant entrainment of liquid in the vapor core. It is unlikely that they did since their analysis assumed that all the liquid was contained in the annulus. Their test apparatus had an L/D ratio of 400 as compared to the header with an L/D ratio of only 6.3. The proximity of the entrance to the header may explain why a majority of the liquid was entrained in the core in this study as opposed to forming an annulus.
The transition from bubble to dispersed flow was clearly delineated in the data for the lower-middle region of the header. The bubble regime occurred when the liquid velocities were high and the gas velocities were low. With an increase in quality, there was a short transition to churn and then a significant region of dispersed liquid flow. At higher vapor velocities some annular flow was observed but no clear boundary could be established.

Bubble flow was the predominate flow regime at the bottom of the header. There was a wide transition range to churn flow and a fairly clear line of demarcation for the dispersed liquid regime. The dispersed liquid regime occurred at high qualities and only a small pool of liquid remained in the cavity below the entrance to the bottom condenser tube. Above this pool there was dispersed liquid flowing in a continuous gas phase.

Another interesting way to look at these data is to fix the quality and travel with the refrigerant down the header. Except for the highest qualities the flow regime will transition from a dispersed liquid flow to a dispersed bubble flow. Traveling down the header under the same operating conditions the velocities drop and with it the Froude number, as refrigerant enters the condenser tubes. Gravity separates the liquid from the flow stream as the inertial forces diminish. For experiment 136 there was dispersed liquid flow at the top of the header (superficial liquid velocity = .0678 m/s and adjusted superficial vapor velocity = 2.15 m/s). At the bottom of the header there was a liquid rich region where bubble flow was predominant (superficial liquid velocity = .00668 m/s and adjusted superficial vapor velocity = .196 m/s). A profile of this phenomena is shown in Figure 5.02. This figure was compiled of video images taken of the different sight plugs installed on the inlet header. To distinguish between liquid and vapor please note that dispersed vapor will form fair sized bubbles in a continuous liquid phase while dispersed liquid forms in sheets, jets or small drops in a continuous gas phase. In Section 6.5 the output from the model for this same experiment predicts the same trend as these images.
Inlet Header
Internal Diameter: 19 mm (.75 inches)
Fluid: R134a
Mass Flow Rate: 96.1 kg/hr (212 lbm/hr)
Pressure: 1.40 MPa (203 psia)
Quality: .208

Figure 5.02 Flow Visualization of Inlet Header Under Steady State Conditions
5.2 Small Channels

5.2.1 Kelvin-Helmholtz

The Kelvin-Helmholtz stability criteria (see section 2.4.5) was applied to the condenser channels to determine if unstable waves were present. These instabilities can lead to intermittent, annular or mist flows. Intermittent flow is supported by this instability as waves coalesce to form slugs. Annular or mist flows are supported from this instability as drops are stripped from the waves and entrained in the vapor. These entrained drops can then deposit on the walls to form a liquid annulus.

To perform this analysis it was necessary to calculate the height of the liquid in the channel. The integral momentum analysis of Taitel-Dukler (1976) was adapted to the geometry of this channel which was modeled as a parabola (see Figure 5.03). To determine the equation of the parabola measurements were taken from an image of the channel. Measurements from the image were compared to those provided by the manufacturer to calculate a magnification factor. The magnification factor of 25 produced good agreement between the measurements as shown in Table 5.01 below.

<table>
<thead>
<tr>
<th>Dimension</th>
<th>Manufacturer</th>
<th>Image (25x)</th>
<th>% Difference</th>
</tr>
</thead>
<tbody>
<tr>
<td>Channel Height</td>
<td>.815</td>
<td>.800</td>
<td>-1.84</td>
</tr>
<tr>
<td>Hydraulic Diameter</td>
<td>.737</td>
<td>.759</td>
<td>+2.99</td>
</tr>
</tbody>
</table>

In their analysis Taitel-Dukler made five assumptions 1) separated flow, 2) equal pressure drops in both phases, 3) the hydraulic gradient in the liquid is negligible, 4) a smooth interface and 5) both phases were turbulent. In addition to these assumptions the channel was assumed to be horizontal for this study (this was verified with a level). With these assumptions the momentum equation yields:

$$X_n \left( \frac{(\bar{U}_L \bar{D}_L)^{-2} \bar{U}_L^3 \bar{S}_L}{A_L} \right) = \left( \frac{\bar{U}_g \bar{D}_g}{A_g} \right)^{-2} \bar{U}_g^2 \left( \frac{\bar{S}_g}{A_g} + \frac{\bar{S}_l}{A_L} \frac{\bar{S}_l}{A_g} \right)$$  \hspace{1cm} (5.01)
Figure 5.03  Channel Geometry Used in Stability Analysis
The sections of Equation 5.01 are composed of the following relationships. Assuming the liquid flows in an open channel and the gas flows in a closed channel the corresponding hydraulic diameters can be calculated as suggested by Agrawal et al. (1973):

\[
\bar{D}_L = \frac{4\bar{A}_L}{\bar{S}_L} \quad \text{and} \quad \bar{D}_G = \frac{4\bar{A}_G}{\bar{S}_G + \bar{S}_i}
\]  

(5.02)

Integrating between the liquid level and the parabola one obtains:

\[
\bar{A}_L = 2\left(\bar{H}_L\bar{W}_L - a\frac{\bar{W}_L}{3}\right)
\]

(5.03)

And then integrating between the top of the channel and the parabola:

\[
\bar{A} = 2\left(\bar{H}\bar{W} - a\frac{\bar{W}^3}{3}\right)
\]

(5.04)

where, by definition

\[
\bar{W}_L = \frac{W_L}{D_h}
\]

(5.05)

\[
\bar{H}_L = \frac{H_L}{D_h}
\]

(5.06)

The equation for a parabola through the origin yields:

\[
a = \frac{\bar{H}_L}{\bar{W}_L^2}
\]

(5.07)

Integration along the line of contact between the perimeter of the channel and each of the two fluids provides the following expressions:

\[
\bar{S}_L = \bar{W}_L\sqrt{\left(1 + \left(2a\bar{W}_L\right)^2\right)\frac{\sinh^{-1}(2a\bar{W}_L)}{2a}}
\]

(5.08)

\[
\bar{S}_G = \bar{W}\sqrt{\left(1 + \left(2a\bar{W}\right)^2\right)\frac{\sinh^{-1}(2a\bar{W})}{2a}} + 2\bar{W} - \bar{S}_L
\]

(5.09)
The length of the interface between the gas and liquid is:

\[ \tilde{S}_i = 2\tilde{W}_L \]  

(5.10)

The continuity equation and definition of superficial velocities furnishes the dimensionless velocities based on dimensionless areas.

\[ \frac{U_k}{U_L^*} = \tilde{U}_L = \frac{\tilde{A}}{A_L} \]  

(5.11)

\[ \frac{U_g}{U_g^*} = \tilde{U}_g = \frac{\tilde{A}}{A_g} \]  

(5.12)

For each of the liquid levels in Figure 5.04 the mass flux was varied beyond the range of conditions encountered in mobile air conditioning applications to insure that all operating extremes were covered in the analysis. It was varied in increments of 20 kg/m²-s. Each point in the figure represents a different mass flux.

The \( h_L/D_k \) of .8 corresponds to a flow quality of .09. At these low qualities the channel is nearly full of liquid and the gas velocities are low. The stability analysis shows that the waves may be stable under these conditions. Since there is so much liquid in the channel, however, a small wave can bridge the gap between the liquid level and the top of the channel. So there is likely to be plug or slug flow under these conditions even though it is not initiated by the coalescence of unstable waves.

As the liquid level decreases to .4 (x=.507) and less, even small waves are unstable. This means that unstable Kelvin-Helmholtz waves are likely to be present. This may lead to coalescence which can promote slug flow or entrainment and deposition which leads to annular and mist flows.
Figure 5.04 Kelvin-Helmholtz Neutral Stability Results for a Small Parabolic Channel
5.2.2 Flow Map

The experimental data were used to calculate the superficial liquid and vapor velocities for all the experiments assuming the mass flow rate was equally distributed among the condenser tubes. Then, the superficial vapor velocity was adjusted to account for the difference in densities of refrigerant vapor and air (see section 2.4.2). This property is the only one that was adjusted. With the adjustment, the flow regime observations made for this study could be compared with the air-water data observed by Damianides et al. (1988)

The data were collected using the flow visualization equipment described in section 3.1. The nature of the flow exiting the small channels was used to determine the flow regime inside the channel. Before the onset of mist conditions and at moderate mass flow rates, it was easy to observe the intermittent nature of the flow as liquid slugs left the channel. These slugs would alternate from one channel to another. As the mass flow rate increased these slugs impacted the opposite wall. At the highest mass flow rates, the amount of liquid impacting the sight plug made it difficult to observe the exit of the channels. These points are indicated with a square symbol in Figure 5.05. During a majority of mist conditions video images were recorded for later analysis since it was difficult to observe the exit of the channel. To aid in the image analysis, a light source was provided at the opposite end of the channel. For the few cases where it was difficult to tell if the flow in the channel was annular or not a frame by frame analysis was performed. If the light level diminished for a period equal to or greater than the residence time of a slug traveling at the superficial velocity of the vapor then it was assumed the flow was intermittent. Some examples of this technique are contained in the next section of the report.

In the legend "SPL" refers to single phase liquid and "I" to intermittent. Where SPL is indicated, there was heat transfer in the condenser tubes. The channels towards
the front of the condenser, according to air flow, had single phase liquid exiting while those in the back had intermittent flow.

![Flow Regime Data at Exit of Channel Compared to Damianides Map](image)

Figure 5.05 Flow Regime Data at Exit of Channel Compared to Damianides Map

From the previous figure it is apparent that the flow regime data from this study for small channels were in agreement with the Damianides map when the refrigerant gas velocity was adjusted to account for the differences between the density of refrigerant vapor and air. The predominate flow regime was intermittent flow.
5.2.3 Flow Regime Detection

An example of the visual data collected during mist conditions is recorded in Figure 5.07. Since the mist was too thick to observe the nature of the flow exiting the condenser channels then one channel was back-lit as described in section 3.1. This channel was the eleventh channel of nineteen from the front of the tube in the direction of air flow, and was located in the topmost condenser tube. Its geometry is best described as a parabolic trough with a cover. Figure 5.06 contains the digital image of the channel during zero flow conditions and illustrates the region contained in the images of figure 5.07. These images highlight the intermittent nature of the flow in the condenser. Contour plots of constant light intensity were superimposed on the digital images to highlight the change in light intensity from one frame to the next. For this experiment (#104) the superficial liquid velocity was .045 m/s and the adjusted superficial vapor velocity was 21.1 m/s. The predicted residence time of a slug traveling at the actual superficial velocity of the vapor was 3.94 frames.

Figure 5.06  Digitized Image of Channel During No Flow Conditions
To justify back lighting the channel as a technique to detect slugs in the channel, the data point closest to the annular flow regime transition was selected for frame by frame analysis. Data from the images were recorded based on the quality of the light. If the light was cloudy or absent then it was recorded as cloudy and if light showed clearly in at least half the core of the channel then it was recorded as clear. Figure 5.08 is the result of 13 seconds of consecutive video data. Each cycle corresponds to a period with and then without light. The superficial velocity of the liquid was .0434 m/s and the adjusted superficial velocity for the vapor was 26.0 m/s.

For a majority of the time this figure shows that the light intensity was blocked for 3 video frames or longer. This indicates that there may have been slugs in the channel. The cycles where the light is blocked for less than the residence time of a slug could be due to the fact that the slug may not have formed at the entrance to the channel. It could also be caused by liquid in the headers which crosses the path of the light. A secondary
A measure of slug existence is needed to eliminate this uncertainty and validate the reliability of the method.

![Residence Time of Slug = 3.1 Video Frames](image)

Assuming slug is traveling at the superficial vapor velocity and forms at the entrance of the channel. 13 seconds of video data from #87.

**Figure 5.08 Flow Regime Analysis Using Back Lighting**

### 5.3 Outlet Header

In the outlet header, flow regime data were recorded as in the inlet header. There were 6 sight plugs in the outlet header as indicated in Figure 3.01. Dispersed liquid was observed under all operating conditions along with a thin layer of liquid on the wall of the header. The dispersed liquid regime was further subdivided into four regimes depending on the form of the liquid. These regimes were classified as sheet, jet, drop and mist.

The distinction between these regimes was clear at the top of the outlet header where the majority of the header contained vapor with a small amount of liquid. Only when higher vapor velocities were obtained did misting occur which made visual
observation of any liquid in the header difficult. The superficial velocities for Fig. 5.09 were calculated based on 1/11 of the total mass flow rate since only one tube of eleven was emptying its contents into the header.

![Flow Regime Map for Top of Outlet Header](image)

**Figure 5.09** Flow Regime Map for Top of Outlet Header

In the bottom of the outlet header (Fig. 5.10) the distinction between the form of the dispersed liquid was not as obvious as the data at the top of the outlet header. This was due to the higher velocities in this section of the header since the entire mass flow rate was passing through this section. The superficial velocities were consequently
calculated based on the total mass flow rate through the test section. Note there was no pooling observed in the bottom of the outlet header under all operating conditions.

Figure 5.10 Flow Regime Map for Bottom of Outlet Header

As one travels down the header under the same operating condition the velocities increase as refrigerant enters from the condenser tubes. For one set of conditions (superficial liquid velocity = .00668 m/s and adjusted superficial vapor velocity = .196 m/s) at the top of the header there was jet flow. At the bottom of the header (superficial liquid velocity = .0678 m/s and adjusted superficial vapor velocity = 2.15 m/s), sheet flow
was observed. A profile of this phenomena is shown in Figure 5.11. This figure was compiled from video images taken of the different sight plugs installed on the outlet header. The outlet header contained a continuous vapor phase with dispersed liquid. Therefore, as the liquid inventory increases the image of the channels becomes less clear. In Section 6.5 the output of the model for this same experiment predicts the same trend as the images in Figure 5.11.

Another important observation in the outlet header was the trajectory of the liquid leaving the condenser tubes. Its trajectory indicates that gravity was an important factor in the motion of the liquid. This information was critical in the development of the refrigerant inventory model for the outlet header.
Outlet Header
Internal Diameter: 19 mm
Fluid: R134a
Mass Flow Rate: 96.1 kg/hr
Pressure: 1.39 MPa
Quality: .208

Figure 5.11 Flow Visualization of Outlet Header Under Steady State Conditions
CHAPTER 6
SIMULATION RESULTS

The model described in Chapter 4 is designed to predict the rate of heat transfer, the pressure drop of the refrigerant and the refrigerant inventory in air-cooled condensers. This model was validated against experiments which covered the range of conditions (see Tables 3.06-3.07) a condenser would encounter in mobile air conditioning applications. The results of this validation are presented in this chapter.

6.1 Heat transfer

The heat transfer was modeled using a correlation (see Section 4.3) which was dependent on flow regime. The heat flux, based on inside heat transfer area, ranged from 6.0 to 11.7 kW/m² in the validation experiments and covered most of the range encountered in mobile air conditioning systems. The remaining operating conditions used in the heat transfer tests are contained in Table 3.07. In order to cover the different ranges of the thermodynamic vapor quality (x), the data were collected in two groups. One group had superheated (S.H.) inlet conditions with two phase conditions at the outlet. The second group had two phase inlet conditions with outlets that varied from sub cooled (S.C.) to two phase.

The agreement between the simulation and the experiment was excellent with all the predicted points falling within ±10% of the experimental values. Therefore, the heat transfer correlations developed for larger diameter circular tubes successfully predict the rate of heat transfer in small channels with a non-circular cross section.

Since an optimization was performed to determine the heat transfer coefficient on the air side of the condenser one may argue that errors in the refrigerant side heat transfer coefficient may be compensated for by the air side correlation. To see if this could happen an intentional error was applied to the refrigerant side heat transfer coefficient correlation. The error was multiplicative in nature and increased the two phase heat
transfer coefficient by 38%. First the conventional model was run with its corresponding air side correlation. Then a new air side correlation was found based on the erroneous model and this model was run for the same set of data. The error in the heat transfer between the two models was calculated and plotted in Figure 6.02 against the superficial velocity of the liquid exiting the condenser tubes. It is clear from this graph that the optimization procedure for determining the air-side correlation did not remove the imposed systematic error in the refrigerant side heat transfer coefficient.

Figure 6.01 Validation of Heat Transfer Model
The friction factor correlation used in the model was developed from seven single phase vapor tests. The correlation, Equation 6.01, fell between the laminar flow correlation, Equation 6.02, and the smooth tube correlation, Equation 6.03, for turbulent flow when the Reynolds number, based on total flow as liquid, was below 10,000. Above this value the data was near the smooth tube correlation for turbulent flow.

\[ \text{ff} = 45.476 \text{Re}^{-.80} \]  

(6.01)
ff = 64 Re\(^{-1}\) \hspace{1cm} (6.02)

ff = 0.079 Re\(^{-2}\) \hspace{1cm} (6.03)

The reason this correlation did not follow the smooth tube correlation may be because the flow is not fully turbulent. The flow was in the transition region where the liquid only Reynolds number varied from 4,000 to 13,000.

Figure 6.03 Validation of Refrigerant Pressure Drop

The pressure drop experienced by the refrigerant was compared with predicted values for the entire data set. Figure 6.03 contains the results of this comparison. At the lower end of the scale the model systematically under predicts the pressure drop. This
systematic error is likely to be caused by using vapor friction factor data to approximate the liquid friction factor. The friction factor for liquid in a smooth pipe is 7 to 22% higher than that for vapor for a Reynolds number between 1000 and 13000. Overall the results were good with the majority of the data falling between the ±30% error lines. ±30% is a reasonable range of uncertainty for complex geometry because of the empirical nature of the loss coefficients used for the various pipe components.

6.3 Refrigerant Inventory

The agreement between the predicted and measured values of refrigerant inventory was excellent. The entire data set was used for this comparison except for four points: two where the stub was not full of liquid, one where the model did not converge and one where it was noted in the lab book that the inlet shut-off valve was closed too slowly. Ninety-five percent of the data fell within the ±10% range.

The data set includes both tests with heat transfer in the test condenser and without. Tests with heat transfer are noted with a solid square in the figure. The model was able to predict the inventory for these tests just as accurately as it did for the adiabatic tests upon which the model was developed. Besides heat flux, the other operating conditions (i.e. pressure, flow rate, and inlet temperature or quality) were varied as summarized in section 3.2.4. Since the model agrees so well with the experiment over the entire range of these conditions, then the model accurately accounts for the variation of these factors.

Thirteen of the data points, noted with a "+" symbol in Figure 6.04, were selected to obtain the parameters for the dimensionless groups used to predict the slip ratio. Separate correlations were developed for the pipes, inlet header and condenser channels due to the different orientations and flow regimes. The outlet header was modeled separately and is described in Section 6.4. These 13 experimental points were used in the optimization program, see Figure 6.05, to determine the parameters for each of the
dimensionless groups. The exponents for the dimensionless groups are specified in Table 6.01 for each region.

Figure 6.04 Experimental Validation of Refrigerant Inventory Model

The objective function had a narrow valley with a shallow slope. Using the Marquardt technique with the thirteen points grouped together produced several local minimums and it was unclear if the optimum had been found. For this reason, the thirteen points were split into two groups: one where the gravitational forces were higher than the viscous forces and the other for the inverse case. The group of data where the gravitational forces prevailed was used to search for the Froude and Weber number.
parameters. The other group was used to search for the Reynolds number parameters. Alternating between these two groups and the composite set of data provided a robust procedure to locate the optimum solution. Once the data set was divided into these two groups the individual runs directed the search along the valley until the global optimum was found. Figure 6.06 illustrates the contour of the valley near the optimum as a function of one of the six parameters. The other parameters produced similar contours.

![Figure 6.05 Parameter Estimation Procedure](image)

Table 6.01 Optimized Parameters Used in Slip Ratio Correlation for Different Sections of Condenser

<table>
<thead>
<tr>
<th>Section\Group</th>
<th>Re</th>
<th>Fr</th>
<th>We</th>
</tr>
</thead>
<tbody>
<tr>
<td>pipes</td>
<td>.074</td>
<td>.778</td>
<td>Not Applicable</td>
</tr>
<tr>
<td>inlet header</td>
<td>.136</td>
<td>.724</td>
<td>Not Applicable</td>
</tr>
<tr>
<td>condenser channels</td>
<td>.185</td>
<td>Not applicable</td>
<td>.609</td>
</tr>
<tr>
<td>original Hughmark</td>
<td>.167</td>
<td>.125</td>
<td>Not applicable</td>
</tr>
</tbody>
</table>
Figure 6.06 Optimization Path for Refrigerant Inventory Model

For Figure 6.06 all the exponents were adjusted at the same time allowing the optimization to proceed along the trough of the valley. To demonstrate the steepness of the sides of the valley the influence of changing only one of the exponents while holding the rest constant was calculated. This was done for the exponent to the condenser Reynolds number. When this exponent was increased from .185 to .195 the objective function increased by .00014 which is 4 to 5 times steeper than following the trough.

Do any of the existing correlations accurately predict the refrigerant inventory in such a complex series of geometry? To address this question a single void fraction correlation, the one developed by Hughmark (1962), was implemented in the pipes, headers and condenser channels. This correlation was chosen since it included most of the relevant forces which affect slip and it produced the best agreement between experiments and models for other researchers. Figure 6.07 demonstrates that this correlation consistently over predicts the amount of refrigerant in the test section except for points below 30 grams. These points were the stub tests where only single phase vapor existed in the condenser and the void fraction correlation is not relevant.
In the outlet header the void fraction is modeled differently than the other sections of the condenser. The observation of the flow in this header yielded significant insight into the modeling. In section 5.3 it was noted that the flow was predominately dispersed liquid flow with a thin film on the wall. The trajectory of the liquid exiting the condenser tubes suggests that gravity was a significant force on the liquid. Since the Froude number

Figure 6.07 Refrigerant Inventory Model Based on Hughmark Correlation
is developed assuming the flow was steady it does not apply to an accelerating flow.

With accelerating liquid it is possible for the velocity of the liquid to exceed the vapor velocity resulting in a slip ratio less than one. The model form proposed by Hughmark limits the slip ratio to a minimum of one.

To model the outlet header it is assumed the interfacial shear forces were negligible. With this assumption the following equations of motion are used to determine the velocity of the liquid and subsequently the void fraction. The outlet header is divided into 12 segments and 11 nodes. The exit of each condenser tube constitutes a node. The nodes occur at the interfaces between the segments and are used to account for the addition of mass. Figure 6.08 illustrates an individual segment with a node at each end. Within the segment the flow enters at some initial velocity and accelerates until it reaches the bottom of the segment.

\[ L = U_{(n-1)}t + \frac{gt^2}{2} \]  \hspace{1cm} (6.04)

\[ U_n^- = U_{(n-1)} + gt \]  \hspace{1cm} (6.05)

Solving Equation 6.04 for \( t \) and substituting into Equation 6.05 the velocity at the bottom of the segment can be found as shown in Equation 6.06.

\[ U_n^- = -U_{(n-1)} + \sqrt{U_{(n-1)}^2 + 2gL} \]  \hspace{1cm} (6.06)

where,

- subscript \( n \) is the number of the node and the corresponding superscript indicates the velocity just before (-) or just after (+) the flow from the condenser tube is added.

At the nodes the conservation of momentum principle is applied to determine the velocity of the liquid after flow from the condenser tube is added. Assuming the two liquid streams combine into one stream and an even distribution of mass to each of the condenser tubes the conservation of momentum at the node yields Equation 6.07.

\[ U_n^+ = \frac{(n-1)U_n^-}{n} \]  \hspace{1cm} (6.07)
From the continuity equation the cross sectional area occupied by the liquid is given by Equation 6.08.

\[ A_L = \frac{2(1 - x)(n - 1)\dot{m}_L}{n_{\text{max}}\rho_L\left(\dot{U}_{n^{-}} + \dot{U}_{(n-1)^+}\right)} \]  

(6.08)

The void fraction is then determined from Equation 6.09.

\[ \alpha = 1 - \left(\frac{A_L}{A_c}\right) \]  

(6.09)

A film of liquid was also observed on the wall of the outlet header. Wallis (1969) describes a method to predict the thickness of this film. Assuming interfacial shear is negligible, the momentum equation for a thin film, neglecting wall curvature, results in a balance between gravitational and wall shear forces as shown in Equation 6.10

\[ \tau = g(\rho_L - \rho_g)(\delta - y) \]  

(6.10)

where, \( y \) is the distance from the wall.
After applying the constitutive relationship and integrating this relationship, Wallis derived the following relationship for laminar flow.

\[ \delta^* = 0.909 \, Re_T^{0.3} \]  

(6.11)

where for a circular tube,

\[ Re_T = \frac{U_l \rho_L D}{\mu_L} \]  

(6.12)

\[ \delta^* = \frac{\delta}{D} N_f^{2/3} \]  

(6.13)

\[ N_f = \frac{D^3 g (\rho_L - \rho_g) \rho_L}{\mu_L} \]  

(6.14)

For turbulent flow, \( Re_T > 1000 \) the following relationship was determined experimentally.

\[ \delta^* = 0.115 \, Re_T^{0.6} \]  

(6.15)

The model assumes that 50% of the liquid flowed in the core and the remainder in the annulus.

6.5 Refrigerant Distribution

From Section 5.1, the flow visualization data for experiment 136 show the flow enters the header as a dispersed liquid which transitions to a liquid rich dispersed bubble flow at the bottom. The same operating conditions as in the experiment were input into the model to produce the following figure (Figure 6.09). A comparison of this figure with the digitized images in Figure 5.02 demonstrates that the model qualitatively predicts the refrigerant inventory in the inlet header.

To show the sensitivity of the model to the different flow regimes in the header a data point (#111) was selected where there was dispersed liquid flow except for the bottom segment (#13) where there was a pool of liquid with a wavy interface. The volume fraction of liquid in the inlet header is significantly reduced compared to the experiment 136 where there was bubble flow in the lower portion of the header.
Figure 6.09 Predicted Refrigerant Distribution for Inlet Header During Exp. 136

Figure Refrigerant Distribution for Inlet Header During Exp. 111
A comparison of the visual data and the model was made for the outlet header also. Comparing Figure 6.11 with Figure 5.11 shows qualitative agreement between the experiment and the model. The image of the flow shows that the quantity of liquid starts out low and increases as it travels down the header. The same trend was observed for experiment 111 except the liquid volume fraction reached .027 at the bottom of the header rather than .087.

Figure 6.11 Predicted Refrigerant Distribution for Outlet Header During Exp. 136

Figure 6.12 illustrates how the refrigerant mass accumulates as it passes through the condenser for one of the experiments. It is interesting to note the inlet header contains the largest percentage of the mass in the condenser.
6.6 Flow Regime Transitions

To account for flow regime transitions the simulation used an empirical relationship developed by Hughmark (1962) which relates the dimensionless groups to the inverse of slip ratio. Figure 6.13 illustrates this relationship where $Z$ was defined in section 4.5. The parameter $Z$ gives the balance between the inertia forces which decrease slip and viscous, gravitational and surface tension forces, depending on the region of the condenser, which increase slip. As inertial forces dominate the others, $Z$ will increase and the slip ratio decreases.
This dimensionless parameter Z was an excellent discriminator of flow regime as shown in Figure 6.14. The bottom of the inlet header was selected to illustrate this point because there were flow regime transitions and they were distinct. For values of Z below .4 bubble flow could be expected. From .4 to 1.2 the flow transitions from a dispersed bubble flow to a dispersed liquid flow. Finally, above 1.2 the flow is dispersed liquid.
6.7 Application of Simulation

Over the life of the experimental apparatus three significantly different condenser coils were tested for the purpose of model development. This validated model was then used to compare the performance of two of these coils under the same operating conditions (see Table 6.02). A figure of merit was needed which would reward the condenser with a lower quantity of refrigerant. Since the mass of refrigerant in a coil is
directly proportional to the internal volume the rate of heat transfer was normalized by this volume.

Table 6.02 Operating Conditions Used In Coil Comparison

<table>
<thead>
<tr>
<th></th>
<th>Temperature</th>
<th>Pressure</th>
<th>Relative Humidity</th>
<th>Mass Flow Rate</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>[°C]</td>
<td>[°F]</td>
<td>[kPa]</td>
<td>(psia)</td>
</tr>
<tr>
<td>Refrigerant</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>60</td>
<td>140</td>
<td>1380</td>
<td>200</td>
</tr>
<tr>
<td>Air</td>
<td>27</td>
<td>80</td>
<td>101</td>
<td>14.7</td>
</tr>
</tbody>
</table>

The small channel condenser consisted of the same geometry as the coil described in this report with the exception being that there were three passes across the air stream instead of one. The first pass had eleven tubes and the remaining two passes had ten tubes each. The piccolo coil had two small headers with four parallel circuits at the entrance and reducing to two circuits at the exit. Its tubes were 5.18 mm (.204 inches) in diameter. A complete description of this coil is contained in Orth (1993). The air side of both coils were modeled identically so that the only difference between the coils was on the refrigerant side. The length of the tubes were calculated so both coils had the same primary heat transfer area. The primary heat transfer area is defined as the outside perimeter of the refrigerant tube multiplied by its length.

The small channel condenser significantly outperforms the piccolo coil on the basis of heat transfer per unit volume. The major reason for this can be attributed the ratio of primary heat transfer area to internal volume. The small channel condenser provides 1.8 time as much primary heat transfer per unit of internal volume than the piccolo tube arrangement. Figure 6.16 shows the heat transfer rate in the small channel condenser was nearly the same as that in the piccolo condenser over the range of refrigerant flow rates.
Figure 6.15 Normalized Heat Transfer Comparison with Piccolo Condenser
Refrigerant pressure drop in the condenser degrades its performance. This was especially evident in the case of the piccolo condenser. This occurs because in the two phase region temperature and pressure are dependent on each other. As the pressure drops so does the temperature of the refrigerant thereby lowering the temperature difference between the refrigerant and air. Since the pressure drop was included in the calculation of heat transfer rate this effect was accounted for in the comparison of heat transfer rates.

Figure 6.16 Heat Transfer Comparison with Piccolo Condenser
The pressure drop of the two coils on the refrigerant-side was included in Figure 6.17. The pressure drops were normalized by the heat transfer rate to account for the small differences in heat transfer rates. The small channel condenser outperformed the piccolo condenser from refrigerant inventory and pressure drop standpoints.

Figure 6.17 Normalized Pressure Drop Comparison with Piccolo Condenser
CHAPTER 7
CONCLUSIONS AND RECOMMENDATIONS

7.1 Summary and Conclusions

This study is the first one to model and experimentally validate refrigerant inventory of R134a in small-channel cross-flow condensers. The mechanistic model accurately accounts for heat transfer, refrigerant pressure drop and inventory in cross-flow heat exchangers with small refrigerant channels distributed in a parallel-flow arrangement. The unique features, from a modeling standpoint, of this type of condenser are the headers and small channels. An extensive literature search did not uncover any other work that models the refrigerant inventory in headers or in channels as small as the ones studied here.

The agreement between the refrigerant inventory model and the experiment was excellent as 95% of the predictions fell within ±10% of the experimental data. Therefore, a one-dimensional two-fluid model was sufficient for modeling the complex geometry present in this condenser. The Reynolds and Froude numbers, and homogeneous liquid volume fraction were adequate predictors of the slip ratio in the pipes and inlet header. In the small channels, the Reynolds and Weber numbers, and homogeneous liquid volume fraction were sufficient to correlate the slip ratio. For the outlet header where gravity driven flow was observed a model which accounts for the acceleration of the liquid in the core and the balance between gravity and viscous forces in the annulus overcame the problems of using the Hughmark form of the correlation for this type of flow.

The heat transfer and pressure drop were modeled with correlations developed by Dobson et. al. (1993) and Souza et.al. (1992). These correlations were developed for circular tubes ranging in diameter from 3.04 to 7.04 mm (.120 to .277 inches). This study has successfully extended these correlations to small channels (Dh=.74 mm (.029 inches)) with non-circular cross sections. The heat transfer was modeled within ±10% and the pressure drop for the majority of data was within ±30%.
The flow regimes were documented for the pipes, headers and small channel condenser tubes. Intermittent flow was the predominate flow regime in the small channels. A new method was discovered for detecting intermittent flow regimes in channels by using back-lighting of the channel during operation. These images were recorded on video tape and later analyzed on a frame-by-frame basis and the results were consistent with the Damianides flow map and Kelvin-Helmholtz stability criteria. Through the inlet header the flow transitioned from a dispersed liquid to a bubble flow regime. The flow regime in the outlet header was always a dispersed, gravity-driven, liquid in the core with a thin liquid annulus. Visual data collected for the headers were in qualitative agreement with the refrigerant inventory model.

The trap and bypass method of collecting refrigerant samples was superior in accuracy to the continuous method of weight measurement. The repeatability was within +/- 6% and the overall experimental uncertainty was 2.3 grams (.005 lbm). The bypass condenser allowed the apparatus to remain in operation during the sampling process thereby minimizing the time required to reach the next experimental point.

The modular approach to modeling provides sufficient resolution for modeling void fraction in channels containing condensing refrigerant. For this application 12 modules were sufficient to model the heat transfer, pressure drop and refrigerant inventory within the experimental uncertainty. Fifty (50) or more modules reduced the uncertainty from the model to negligible level.

The simulation was applied to a more conventional condenser for the purpose of comparison. The conventional condenser had tubes with an internal diameter of 5.18 mm (.204 inches) which were connected with headers and started with four parallel circuits reducing down to two at the exit. Industry manufacturers commonly refer to this configuration as a piccolo condenser. The air-sides of both condensers were modeled identically; only the refrigerant circuitry was different. The heat transfer performance was similar in both coils. However, the internal volume of the piccolo condenser was
nearly twice the size of the small channel condenser, therefore requiring more refrigerant to perform the same task. The pressure drop was also higher by more than a factor of two in the piccolo condenser. On a refrigerant inventory and pressure drop basis the small channel condenser significantly outperformed the piccolo tube condenser.

7.2 Recommendations

Since the model is a mechanistic model it may be used to optimize the design of small-channel cross-flow condensers used in automotive applications. Care must be taken not to exceed the range of experimental conditions used in developing the model. The model can also be integrated into a system model to perform charge optimization and to provide the information necessary to avoid the specification of a refrigerant state within the system.

Next, based on visual observations of the flow in the inlet header it was apparent that the flow was not evenly distributed among the condenser tubes as the model assumes. More of the mass flow was allocated to the lower tubes than the upper tubes. That observation is in agreement with measurements taken by Collier (1976), Coney (1980) and Miller (1971) on manifolds with air and water. Future testing should separate the condenser tubes from the headers to experimentally determine the magnitude of this maldistribution.

Finally, this condenser was tested with only one fluid. Although the properties of this fluid did change with the operating conditions of the system, testing different fluids with significantly different properties would increase the generality of the model. Also, the refrigerant was tested without oil. Considering most vapor compression systems have oil circulating with the refrigerant the effect of this variable should be studied too.
REFERENCES


