Predicting Refrigerant Inventory of HFC 134a in Air Cooled Condensers

L. Orth, D. C. Zietlow, and C. O. Pedersen

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Department of Mechanical and Industrial Engineering
University of Illinois at Urbana-Champaign, 1993

ABSTRACT

Currently, the refrigerant inventory for vapor compression systems is determined using a costly trial and error procedure. An accurate computer model of the refrigerant in a system would reduce the time and expense of this process. This paper presents a computer model that has the capability to predict the amount of refrigerant in air cooled condensers.

For accurate prediction of refrigerant inventory two important steps are required. The first step is to model the heat transfer of the coil. This is needed to separate the three regions of the coil, de superheating, condensing, and sub cooling and then to further divide the condensing region into increments of quality.

The second step is to predict the void fraction (ratio of area occupied by gas to liquid) throughout the condensing region. Void fraction correlations by Domanski and Didion[11], Hughmark[14], Premoli et al.[12], and Tandon et al. [10] are included in the model.

The simulation is used to model a cross flow heat exchanger, with parallel refrigerant paths, used in mobile air conditioning systems. The model is exercised to illustrate the effects of different operating conditions and void fraction correlations on the refrigerant inventory in condensers. A condenser used in mobile air conditioning applications is modeled. Future work includes experimental validation of the model.
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1.0 Introduction

Accurate prediction of the mass of a system component is important for several reasons. In condenser modeling, accurate charge inventory prediction is required for simulating off design performance and transient behavior. In system modeling, the charge inventory of all components is required for the continuity equation. This work focuses on predicting refrigerant inventory in air cooled tube and fin condensers. In the single phase regions of the condenser, mass prediction is a straightforward calculation. In the two-phase region, inventory is calculated using a void fraction correlation. The void fraction correlation provides a ratio of the tube cross sectional area occupied by refrigerant vapor to the total cross sectional area for a given quality. There are many void fraction correlations available in the literature; Domanski and Didion [11], Baroczy [18], Zivi [17], Smith [19], Premoli et al. [12], Tandon et al. [10], and Hughmark [14]. The purpose of the work presented here is to implement a number of these correlations into a condenser simulation program to determine the mass of the condenser and provide a comparison of the correlations for different operating conditions. Determination of the most accurate correlation requires experimental data of the charge for the condenser coil being modeled. This work will be available in the future and is discussed in section 5.0.

The condenser simulation program used for this study was originally developed by Ragazzi [1]. The steady state simulation was designed to model air-cooled condensers which could be categorized as cross-flow heat exchangers with circular refrigerant tubes with air flowing over them. It is based on first principles and is general enough to use with condenser coils of various geometries and for flows of different refrigerants. A complete description of the program and its adaptation to various condenser coil geometries is provided. The various solution techniques for the charge inventory correlations are also examined and compared.
The results indicate that the prediction of refrigerant inventory is influenced by the selection of void fraction correlation and the predicted length of the subcooling section in the condenser. This length is determined by the simulation program and is dependent upon the correlations used for the refrigerant side heat transfer coefficients and pressure drop. It will be shown that it is as important for the condenser model to properly predict the outlet conditions as it is to predict the overall capacity of the coil. Inventory comparisons are provided for the best combination of heat transfer and pressure drop correlations.

In section 2 of the paper, an overview of the Module Based Condenser Simulation program is provided. It outlines the solution technique and the relevant heat transfer and pressure drop correlations. In addition, a discussion of the thermodynamic property routines used in the simulation is provided. In section 3, a description of the condenser coil used for this work is given. A summary of the assumptions which were made is provided along with a detailed description of how the coil was modeled. The void fraction correlations implemented in the program are discussed in section 4, along with the integral solution techniques and the final results. Future work to experimentally verify the accuracy of the void fraction models is discussed in section 5.
# 1.1 Nomenclature

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Definition</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>( \alpha )</td>
<td>Void fraction</td>
<td></td>
</tr>
<tr>
<td>( A_i )</td>
<td>Inside tube cross-sections area</td>
<td>ft²</td>
</tr>
<tr>
<td>( A_{mean} )</td>
<td>Mean tube cross-sectional area</td>
<td>ft²</td>
</tr>
<tr>
<td>( A_{si(i)} )</td>
<td>Module i inside surface area</td>
<td>ft²</td>
</tr>
<tr>
<td>( A_{so(i)} )</td>
<td>Module i outside surface area</td>
<td>ft²</td>
</tr>
<tr>
<td>( C_{air(i)} )</td>
<td>Module i air heat capacity rate</td>
<td>Btu/hr°F</td>
</tr>
<tr>
<td>( C_{max(i)} )</td>
<td>Module i maximum heat capacity rate</td>
<td>Btu/hr°F</td>
</tr>
<tr>
<td>( C_{min(i)} )</td>
<td>Module i minimum heat capacity rate</td>
<td>Btu/hr°F</td>
</tr>
<tr>
<td>( c_{pA(i)} )</td>
<td>Module i air specific heat</td>
<td>Btu/lbm°F</td>
</tr>
<tr>
<td>( c_{p air} )</td>
<td>Condenser air specific heat</td>
<td>Btu/lbm°F</td>
</tr>
<tr>
<td>( c_{pf(i)} )</td>
<td>Module i refrigerant liquid specific heat</td>
<td>Btu/lbm°F</td>
</tr>
<tr>
<td>( c_{pr(i)} )</td>
<td>Module i refrigerant specific heat</td>
<td>Btu/lbm°F</td>
</tr>
<tr>
<td>( C_{ref(i)} )</td>
<td>Module i refrigerant heat capacity rate</td>
<td>Btu/hr°F</td>
</tr>
<tr>
<td>( C_{ratio(i)} )</td>
<td>Module i heat capacity ratio</td>
<td></td>
</tr>
<tr>
<td>( D_i )</td>
<td>Inside tube diameter</td>
<td>ft</td>
</tr>
<tr>
<td>( D_h )</td>
<td>Air side hydraulic diameter</td>
<td>ft</td>
</tr>
<tr>
<td>( \Delta P_{com} )</td>
<td>Component refrigerant pressure drop</td>
<td>psi</td>
</tr>
<tr>
<td>( \Delta P_{fric(i)} )</td>
<td>Module i frictional refrigerant pressure drop</td>
<td>psi</td>
</tr>
<tr>
<td>( \Delta P_{grav(i)} )</td>
<td>Module i gravitational refrigerant pressure drop</td>
<td>psi</td>
</tr>
<tr>
<td>( \Delta P_{mom(i)} )</td>
<td>Module i momentum refrigerant pressure drop</td>
<td>psi</td>
</tr>
<tr>
<td>( \Delta P_{mod(i)} )</td>
<td>Module i total refrigerant pressure drop</td>
<td>psi</td>
</tr>
<tr>
<td>( \varepsilon_{(i)} )</td>
<td>Module i effectiveness</td>
<td></td>
</tr>
<tr>
<td>( f )</td>
<td>Fanning friction factor</td>
<td></td>
</tr>
<tr>
<td>( F_r )</td>
<td>Refrigerant Froude number</td>
<td></td>
</tr>
<tr>
<td>( f_Q(x) )</td>
<td>Heat flux assumption</td>
<td></td>
</tr>
<tr>
<td>( G_{air} )</td>
<td>Air side total mass flux</td>
<td>lbm/ft²-hr</td>
</tr>
<tr>
<td>( G_{ref} )</td>
<td>Refrigerant mass flux</td>
<td>lbm/ft²-hr</td>
</tr>
<tr>
<td>( H )</td>
<td>Condenser height</td>
<td>ft</td>
</tr>
<tr>
<td>( h_{air} )</td>
<td>Air side total heat transfer coefficient</td>
<td>Btu/hr-ft²-°F</td>
</tr>
<tr>
<td>( h_{air(i)} )</td>
<td>Module i air side heat transfer coefficient</td>
<td>Btu/hr-ft²-°F</td>
</tr>
<tr>
<td>( h_{fg} )</td>
<td>Latent heat of vaporization</td>
<td>Btu/lbm</td>
</tr>
<tr>
<td>( h_{ref(i)} )</td>
<td>Module i refrigerant side heat transfer coefficient</td>
<td>Btu/hr-ft²-°F</td>
</tr>
<tr>
<td>( h_{Rin(i)} )</td>
<td>Module i refrigerant inlet enthalpy</td>
<td>Btu/lbm</td>
</tr>
</tbody>
</table>
\( h_{\text{Rout}(i)} \)  
Module i refrigerant outlet enthalpy  
\( j \)  
Colburn j-factor  
\( k_{\text{air}} \)  
Air thermal conductivity  
\( k_{f(i)} \)  
Module i refrigerant liquid thermal conductivity  
\( k_{\text{ref}(i)} \)  
Module i refrigerant thermal conductivity  
\( k_{\text{tube}} \)  
Tube thermal conductivity  
\( L_{\text{cond}} \)  
Total refrigerant tube length  
\( L_{\text{mod}(i)} \)  
Module i length  
\( L_{\text{seg}} \)  
Segment length  
\( m_{\text{liq}(i)} \)  
Module i refrigerant liquid mass  
\( m_{\text{mod}(i)} \)  
Module i refrigerant mass  
\( m_{\text{total}} \)  
Total condenser mass  
\( m_{\text{vap}(i)} \)  
Module i refrigerant vapor mass  
\( m_{A \text{mod}(i)} \)  
Module i air mass flow rate  
\( m_{\text{Atot}} \)  
Total air mass flow rate  
\( m_{R \text{mod}} \)  
Module refrigerant mass flow rate  
\( m_{R \text{seg}} \)  
Segment refrigerant mass flow rate  
\( m_{R \text{tot}} \)  
Total refrigerant mass flow rate  
\( n \)  
Number of modules in segment  
\( n_{\text{s}} \)  
Air side overall efficiency  
\( n_{\text{tubes}} \)  
Number of tubes in manifold section  
\( \text{NTU} \)  
Number of transfer units  
\( N_{U\text{air}} \)  
Air side Nusselt number  
\( N_{U\text{ref}(i)} \)  
Module i refrigerant Nusselt number  
\( \Phi_{\text{vap}} \)  
Two phase frictional multiplier  
\( p \)  
Number of segments in condenser  
\( \text{Pr}_{\text{air}} \)  
Air side Prandtl number  
\( \text{Pr}_{f(i)} \)  
Module i refrigerant liquid Prandtl number  
\( \text{Pr}_{\text{ref}(i)} \)  
Module i refrigerant Prandtl number  
\( \text{Pr}_{\text{in}(i)} \)  
Module i refrigerant inlet pressure  
\( \text{Pr}_{\text{out}(i)} \)  
Module i refrigerant outlet pressure  
\( Q_{\text{mod}(i)} \)  
Module i total heat transfer  
\( \text{Re}_{\text{air}} \)  
Air side Reynolds number  
\( \text{Re}_{\text{eq}(i)} \)  
Module i equivalent Reynolds number  
\( \text{Re}_{f(i)} \)  
Module i liquid Reynolds number  
\( \text{Re}_{g(i)} \)  
Module i vapor Reynolds number
\( R_{\text{ref}(i)} \)  
Module i refrigerant Reynolds number

\( \text{Res}(n, 2) \)  
Residual equation array for n modules

\( \rho_{\text{avg}(i)} \)  
Module i refrigerant average density  
lbm/ft\(^3\)

\( \rho_f(i) \)  
Module i refrigerant liquid density  
lbm/ft\(^3\)

\( \rho_g(i) \)  
Module i refrigerant vapor density  
lbm/ft\(^3\)

\( \rho_{\text{ref}(i)} \)  
Module i refrigerant density  
lbm/ft\(^3\)

\( \sigma \)  
Refrigerant surface tension

\( \tau_{\text{wall}} \)  
Wall shear stress

\( T_{\text{Air in back}}(i) \)  
Inlet air temperature to back modules of segment  
°F

\( T_{\text{avg out front}}(i) \)  
Average air outlet temperature from front modules of segment  
°F

\( T_{\text{Rin}(i)} \)  
Module i refrigerant inlet temperature  
°F

\( U_A \)  
Overall heat transfer coefficient

\( \mu_{\text{air}} \)  
Air side viscosity  
lbm/ft-hr

\( \mu_f(i) \)  
Module i refrigerant liquid viscosity  
lbm/ft-hr

\( \mu_g(i) \)  
Module i refrigerant vapor viscosity  
lbm/ft-hr

\( \mu_{\text{ref}(i)} \)  
Module i refrigerant viscosity  
lbm/ft-hr

\( V(i) \)  
Module i refrigerant velocity  
ft/s

\( v_f(i) \)  
Module i refrigerant liquid specific volume  
ft\(^3\)/lbm

\( v_g(i) \)  
Module i refrigerant vapor specific volume  
ft\(^3\)/lbm

\( v_{\text{in}(i)} \)  
Module i refrigerant inlet specific volume  
ft\(^3\)/lbm

\( v_{\text{out}(i)} \)  
Module i refrigerant outlet specific volume  
ft\(^3\)/lbm

\( W_{\text{ef}} \)  
Refrigerant liquid Weber number

\( W_g \)  
Heat flux averaged void fraction

\( x_{\text{out}(i)} \)  
Module i refrigerant outlet quality

\( X_{H(i)} \)  
Module i Lockhart-Martinelli correlating parameter
2.0 Simulation Description

2.1 Solution Technique

The Module Based Condenser Simulation program is a general, first principles condenser simulation which was developed to model air-cooled cross-flow condenser coils of various geometries. The general technique used by the model is to divide the condenser coil into a user specified number of segments which in turn are divided into a number of modules. The inlet conditions to the segment are provided and then each module in the segment in treated as an individual heat exchanger and a Newton-Raphson solution is performed to determine the outlet conditions for each module. This procedure is used to analyze all the segments in the condenser coil. The governing equations for each module are the conservation of energy and momentum equations. These equations provide a set non-linear equations which are then solved with a Newton-Raphson iteration technique. A description of this technique is provided in Stoecker [2].

The operating conditions at the condenser coil inlet which must be specified for the simulation are the refrigerant inlet pressure, temperature and mass flow rate and the air inlet pressure, temperature, mass flow rate and relative humidity. These are what will be referred to as the inlet test conditions. As stated earlier, each module is treated as an individual heat exchanger so these same inlet variables must be specified for each module. Certain assumptions are made to get this information for each module, and hence, the accuracy of the solution is inherent on the validity of these assumptions. For example, the condenser coil modeled by Ragazzi consisted of two parallel, serpentine, finned tubes with air flowing normal to the tubes. The inlet test conditions to the condenser coil were known (from experiments). The coil was divided into two segments, the front tube and the back tube. So the inlet test conditions on the refrigerant side for each segment were assumed to be the same as those for the inlet to the condenser, with the exception of
the refrigerant mass flow rate which was assumed to be one-half of the total refrigerant mass flow rate. This would also be the refrigerant mass flow rate for each module.

\[ \dot{m}_{R_{\text{mod}}} = \dot{m}_{R_{\text{seg}}} = 0.5 \times \dot{m}_{R_{\text{tot}}} \]  \hfill [2.1]

For the air side inlet conditions, there is a slightly different situation. The inlet air temperature and pressure for each module, in the first tube of the coil modeled by Ragazzi, are equal to the inlet pressure and temperature for the test point. The inlet air temperature to the second tube (segment) is set equal to the average outlet air temperature from the first tube (segment). This is discussed in greater detail in Ragazzi [1]. The air mass flow rate must also be determined. For this example, it was assumed that the air flow rate in the duct was uniform so a weighted average was taken to determine the mass flow rate over each module.

\[ \dot{m}_{A_{\text{mod(i)}}} = \dot{m}_{A_{\text{tot}}} \times \frac{L_{\text{mod(i)}}}{L_{\text{seg}}} \] \hfill [2.2]

Here the weighting function is a ratio of the length of the module to the total length of the segment, which is the total length of one tube. 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. This weighting function can change depending on the geometry of the coil. For example, if a condenser coil had only one refrigerant tube, this tube could be modeled as any number of segments. For this case, the total length of the condenser tube is used in the weighting function instead of the length of one segment. The subscript \((i)\) in equation [2.2] designates the fact that this value will (or may) change for each module and will be utilized throughout this paper. This subscript is not used for the refrigerant module mass flow rate because this value is constant throughout a segment.

One additional determination needs to be made for the condenser simulation, and that is whether the user wants to fix the length of a module or the
outlet quality of the module. The example above would work well with either of these versions. The module length could be specified for each module and the program will solve for the outlet quality. If the outlet quality is specified for each module then the program determines the length of each module required to obtain this condition. Both versions have their advantages and disadvantages. For example, the fixed quality version is useful if the length of each of the condenser regions (superheated vapor, two phase and subcooled liquid) is to be determined. For more complicated geometry, however, the fixed length version is more adaptable to the coil geometry as will be shown later. One source of error in the fixed length version is a result of the determination of the local refrigerant side heat transfer coefficients. The heat transfer correlation used for a module is always based on the outlet conditions. For example, if the inlet condition of a module is superheated vapor and the outlet condition of the module is two phase, then the program uses the two phase correlation to determine the heat transfer coefficient for the entire module. In reality, the transition from superheated vapor to two phase occurred somewhere in the module. This problem would only occur at the two transition points (superheated vapor to two phase refrigerant, and two phase refrigerant to subcooled liquid), and can be compensated for somewhat by making the length of the modules smaller.

As stated previously, the simulation uses a Newton-Raphson solution technique for each segment to determine the outlet conditions of each module. The Newton-Raphson variables for both versions of the simulation are the module outlet enthalpy \( h_{\text{Rout}(i)} \) and the module outlet pressure \( p_{\text{Rout}(i)} \). The residuals equations for the Newton-Raphson iteration are given by,

\[
\text{Res}(i, 1) = h_{\text{Rout}(i)} - h_{\text{Rin}(i)} - \frac{Q_{\text{mod}(i)}}{m_{R\text{mod}}} = h_{\text{Rout}(i)} - h_{\text{Rout}(i-1)} - \frac{Q_{\text{mod}(i)}}{m_{R\text{mod}}} \tag{2.3}
\]

\[
\text{Res}(i, 2) = p_{\text{Rout}(i)} - p_{\text{Rin}(i)} - \Delta P_{\text{mod}(i)} = p_{\text{Rout}(i)} - p_{\text{Rout}(i-1)} - \Delta P_{\text{mod}(i)} \tag{2.4}
\]
In equation [2.3], \( Q_{mod(i)} \) represents the total heat transfer from the module and in equation [2.4] \( \Delta p_{mod(i)} \) represents the total pressure drop in the module. The total heat transfer is calculated using an effectiveness-NTU method and the total pressure drop is calculated using a pressure drop correlation.

For certain coil geometries, an additional Newton Raphson variable is used. This variable is the average outlet air temperature from the modules. This situation occurs when the refrigerant tube is wrapped from the front of the coil 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 portion of the tube at the back of the condenser requires the average air outlet temperature from the portion of the tube in the front of the coil to use as its inlet air condition. This is accomplished by modeling the tube as one segment and having an equal number of modules in both the front and rear portions of the tube. The Newton-Raphson residual equation which is added to the iteration is given by,

\[
\text{Res} = T_{\text{air in back}} - T_{\text{avg out front}}
\]  

[2.5]

A more in depth discussion of this procedure is provided in Zietlow [24].

2.2 Effectiveness-NTU Method

The effectiveness-NTU method is used in the condenser simulation to determine the heat rejected by each module \( Q_{mod(i)} \). The general form of the equation is,

\[
Q_{mod(i)} = \epsilon_{(i)} \cdot C_{\text{min}(i)} \cdot (T_{R in(i)} - T_{A in(i)})
\]  

[2.6]

The module effectiveness \( \epsilon_{(i)} \) is defined as the ratio of the actual heat transfer rate to the "ideal" or maximum heat transfer rate. This value is dependent on the ratio of heat capacity rates and the number of transfer units (NTU).

\[
C_{ratio(i)} = \frac{C_{\text{min}(i)}}{C_{\text{max}(i)}}
\]  

[2.7]
The heat capacity rates are calculated for both the refrigerant and the air and the minimum value of these is $C_{\text{min}(i)}$ and the maximum value is designated as $C_{\text{max}(i)}$. The heat capacity rate is the mass flow rate times the specific heat.

\[ C_{\text{ref}(i)} = \dot{m}_{\text{R mod}} \times c_{\text{pR}(i)} \]  \[ C_{\text{air}(i)} = \dot{m}_{\text{A mod}(i)} \times c_{\text{pA}(i)} \]

The variable $\text{UA}$ in equation [2.8] is the overall heat transfer coefficient and is the sum of the convective resistances on the air side and refrigerant side of the condenser coil along with the conductive resistance of the tube wall.

\[ \text{UA} = \frac{1}{\frac{1}{\eta_{\text{air}(i)} A_{\text{so}(i)}} + \frac{1}{k_{\text{tube}}} A_{\text{mean}} + \frac{1}{h_{\text{ref}(i)} A_{\text{oi}(i)}}} \]  \[ \text{[2.11]} \]

The effectiveness is then calculated for one of three cases:

1. $C_{\text{min}(i)} = C_{\text{air}(i)}$ and the refrigerant is single phase

\[ \varepsilon_{(i)} = \frac{1}{C_{\text{ratio}(i)}} \left[ 1 - e^{-NTU \times C_{\text{ratio}(i)}} \right] \]  \[ \text{[2.12]} \]

2. $C_{\text{min}(i)} = C_{\text{ref}(i)}$ and the refrigerant is single phase

\[ \varepsilon_{(i)} = e^{-NTU \times C_{\text{ratio}(i)}} \]  \[ \text{[2.13]} \]

3. Refrigerant is two phase: $\frac{C_{\text{min}(i)}}{C_{\text{max}(i)}} \rightarrow 0$

\[ \varepsilon_{(i)} = 1 - e^{-NTU} \]  \[ \text{[2.14]} \]

The third case is the limiting case which occurs in the condensing region when the refrigerant side specific heat approaches infinity. Once the coil geometry and material are specified, equation [2.11] indicates that there are three unknowns in the equation which must be determined: $\eta_{\text{air}}$, $h_{\text{air}(i)}$, and $h_{\text{ref}(i)}$. The formulation for determining these variables will be provided in the next sections.
2.3 Air Side Heat Transfer Correlation

The air side heat transfer coefficient \( h_{\text{air}} \) used in the simulation program for the charge inventory analysis is the Colburn j-factor. The j-factor is determined for the condenser coil using experimental data and a modified Wilson plot technique [3]. The modified Wilson plot is used to provide the surface efficiency \( \eta_s \) of the coil in addition to the j-factor correlation. If no experimental data are available, Kays and London [13] have tabulated data for various heat exchanger geometries which can be used to determine this correlation. The general formulation for the j-factor is:

\[
j = \frac{\text{Nu}_{\text{air}}}{\text{Re}_{\text{air}} \text{Pr}_{\text{air}}^{1/3}} \tag{2.15}
\]

where,

\[
\text{Nu}_{\text{air}} = \text{Air Nusselt Number} = \frac{h_{\text{air}}D_h}{k_{\text{air}}} \tag{2.16}
\]

\[
\text{Re}_{\text{air}} = \text{Air Reynolds Number} = \frac{G_{\text{air}}D_h}{\mu_{\text{air}}} \tag{2.17}
\]

\[
\text{Pr}_{\text{air}} = \text{Air Prandtl Number} = \frac{\mu_{\text{air}} c_{p \text{ air}}}{k_{\text{air}}} \tag{2.18}
\]

Once the j-factor has been determined for various data points using the modified Wilson plot technique, a correlation for the j-factor can be found as a function of the air side Reynolds number. A least squares fit was utilized in the determination of this correlation. For the coil used in this paper, this correlation was found to be,

\[
j = 0.166 \text{Re}_{\text{air}}^{-0.4} \tag{2.19}
\]

The heat transfer correlation is then found by rearranging equation [2.15] and canceling terms.

\[
h_{\text{air}} = \frac{G_{\text{air}} c_{p \text{ air}} j}{\text{Pr}_{\text{air}}^{2/3}} \tag{2.20}
\]
This is the overall air side heat transfer coefficient. The module heat transfer coefficient is found by multiplying the overall coefficient by a weighting function.

\[ h_{\text{air}(i)} = h_{\text{air}} \times \frac{L_{\text{mod}(i)}}{L_{\text{seg}}} \]  

[2.21]

This is the same weighting function that was used for the example in section 2.1 to determine the module air mass flow rate in equation [2.2]. A more in depth discussion of the modified Wilson plot technique can be found in Weber [4].

### 2.4 Refrigerant Side Heat Transfer Correlations

The calculation of the local refrigerant side heat transfer coefficient \( h_{\text{ref}(i)} \) is dependent upon the refrigerant phase. If the refrigerant in the module is either a superheated vapor or a subcooled liquid, a single phase correlation is used. If the module contains a two phase flow, a two phase correlation is used. In section 2.3, the module air side heat transfer coefficient was found by determining an overall air side coefficient for the coil and applying a weighting function. On the refrigerant side, this procedure is not necessary. There are many heat transfer correlations available for single and two phase flows through circular cross section tubes. This allows the heat transfer coefficient for each module to be calculated independently. Also, one assumption made by the model is that the heat transfer in the return bends of the condenser is negligible since they are not exposed to the air flow.

### Single Phase Heat Transfer Correlation

In the single phase region of the condenser, the Dittus-Boelter [20] correlation was used to determine the refrigerant heat transfer coefficient for the module. This correlation is a general empirical correlation for single phase flow through cylindrical tubes which expresses the dimensionless Nusselt number as a function
of the dimensionless Reynolds and Prandtl numbers. The general form of this correlation for cooling is given by,

$$Nu_{ref(i)} = 0.023 \times Re_{ref(i)}^{0.8} \times Pr_{ref(i)}^{0.3}$$ \[2.22\]

The heat transfer coefficient for the module is related to the Nusselt number by,

$$h_{ref(i)} = \frac{Nu_{ref(i)} \times k_{ref(i)}}{D_i}$$ \[2.23\]

The dimensionless Reynolds number and Prandtl number for the module are given by,

$$Re_{ref(i)} = \frac{\dot{m}_{R_mod} \times D_i}{A_i \times \mu_{ref(i)}}$$ \[2.24\]

$$Pr_{ref(i)} = \frac{\mu_{ref(i)} \times c_p R(i)}{k_{ref(i)}}$$ \[2.25\]

It should be noted that additional single phase correlations are available for use in the simulation program and can be found in Ragazzi [1].

**Two Phase Heat Transfer Correlations**

In the calculation of condenser mass inventory, two refrigerant two phase correlations will be compared, the Cavallini-Zecchin [5] and Dobson [6]. Both correlations were developed for annular flow regimes. The Cavallini-Zecchin correlation is one of the most general empirical correlations. It was developed using a large amount of data from various substances. The Dobson correlation, on the other hand, was developed specifically for R134a. Overall it was found that using the Dobson correlation provided more accurate results for the condenser outlet conditions. This will be discussed in greater detail in the results section.

The general form of the Cavallini-Zecchin correlation is given by,

$$h_{ref(i)} = 0.05 \times Re_{eq(i)}^{0.8} \times Pr_{f(i)}^{0.33} \left[ \frac{k_f(i)}{D_i} \right]$$ \[2.26\]
where,

\[ Re_{eq(i)} = Re_{f(i)} + Re_{g(i)} \left[ \frac{\rho_{f(i)}}{\mu_{f(i)}} \right] \left[ \frac{1}{\rho_{g(i)}} \right]^{0.5} \]

\[ Re_{f(i)} = \frac{G_{ref} \times D_{i}}{\mu_{f(i)}} (1 - x_{out(i)}) \] \[ [2.28] \]

\[ Re_{g(i)} = \frac{G_{ref} \times D_{i}}{\mu_{g(i)}} (x_{out(i)}) \] \[ [2.29] \]

\[ Pr_{f(i)} = \frac{\mu_{f(i)} \times c_{pf(i)}}{k_{f(i)}} \] \[ [2.30] \]

The Dobson correlation which was developed for R134a is a function of the Lockhart-Martinelli correlating parameter \( X_{tt} \). Its general form is given by,

\[ h_{ref(i)} = 2.61 \times h_{f(i)} / X_{tt(i)}^{0.80} \] \[ [2.31] \]

where,

\[ X_{tt(i)} = \left[ \frac{\rho_{g(i)}}{\rho_{f(i)}} \right]^{0.5} \left[ \frac{1}{\mu_{f(i)}} \right]^{-0.125} \left[ \frac{1}{\mu_{g(i)}} \right]^{-0.875} \] \[ [2.32] \]

and \( h_{f(i)} \) is the liquid heat transfer correlation,

\[ h_{f(i)} = \frac{0.023 \times k_{f(i)} \times Re_{f(i)}^{0.8} \times Pr_{f(i)}^{0.3}}{D_{i}} \] \[ [2.33] \]

Equation [2.33], is the Dittus Boelter single phase liquid heat transfer given by equation [2.20], and the liquid Reynolds number and Prandtl number have the same definition as given by equations [2.28] and [2.30].

2.5 Refrigerant Pressure Drop Correlations

As was the case for the refrigerant heat transfer coefficient, the refrigerant pressure drop across the module is dependent on the phase of the refrigerant. The simulation program contains both single phase and two phase pressure drop correlations. In addition to this, there are three components for the module
pressure drop: frictional pressure drop, momentum pressure drop and gravitational pressure drop.

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

[2.34]

The frictional pressure drop and the momentum pressure drop are determined for each module individually. For the gravitational pressure drop, the total elevation change in the condenser is used to determine an overall gravitational pressure drop which is then multiplied by the weighting function \( L_{\text{mod}(i)}/L_{\text{cond}} \) to provide the gravitational pressure drop for each module. This weighting function uses the total length of refrigerant tube. The frictional pressure drop in the single phase region is found using the Fanning friction factor and in the two phase region using the Lockhart-Martinelli correlation. The momentum pressure drop is obtained through a control volume analysis over the module [1].

**Single Phase Pressure Drop Correlation**

The frictional pressure drop in the single phase regions of the condenser coil is determined using the Fanning friction factor which is given by,

\[ f = \frac{\tau_{\text{wall}}}{0.5 \rho_{\text{ref}(i)} V_{(i)}^2} \]  

[2.35]

For laminar flow the following correlation can be used,

\[ f = 16 / Re_{\text{ref}(i)} \]  

[2.36]

For turbulent refrigerant flow,

\[ f = 0.046 \times Re_{g(i)}^{-0.2} \quad \text{for vapor} \]  

[2.37]

or,

\[ f = 0.079 \times Re_{f(i)}^{-0.25} \quad \text{for liquid} \]  

[2.38]

The frictional pressure drop for the module is then given by,

\[ \Delta P_{\text{fric}(i)} = \frac{2 \times f \times G_{\text{ref}} \times L_{\text{mod}(i)}}{D_i \times \rho_{\text{ref}(i)}} \]  

[2.39]
The momentum pressure drop for the module is given by,

$$\Delta P_{\text{mom}(i)} = -G_{\text{ref}}^2 \left( u_{\text{out}(i)} - u_{\text{in}(i)} \right)$$  \[2.40\]

The gravitational pressure drop for the module is given by,

$$\Delta P_{\text{grav}(i)} = \rho_{\text{ref}(i)} g \frac{L_{\text{mod}(i)}}{L_{\text{cond}}}$$  \[2.41\]

**Two Phase Pressure Drop Correlation**

A separated flow model was used in the simulation to calculate the pressure drop in the two phase region. It was developed by Lockhart and Martinelli based on their studies of air-water flows. The general form of the two phase pressure drop correlation is given by,

$$\Delta P_{\text{fric}(i)} = \Phi_{\text{vap}}^2 \left[ \frac{2 \times f_{\text{vap}} \times G_{\text{ref}}^2 \times x_{\text{out}(i)}^2 \times L_{\text{mod}(i)}}{D_i \times \rho_{g(i)}} \right]$$  \[2.42\]

where $\Phi$ is a two phase frictional multiplier which is dependent on whether there is laminar or turbulent flow in the liquid and vapor, and $f_{\text{vap}}$ is the vapor Fanning friction factor given by equation [2.37]. Various forms for $\Phi$ are available in the literature, Soliman et al. [15] and Chisholm [16]. The two-phase frictional multiplier developed by Soliman et al. was for the case of both turbulent liquid and vapor (the turbulent-turbulent case) and is given by,

$$\Phi_{\text{vap}} = 1 + 2.85 X_{\text{lt}(i)}^{0.523}$$  \[2.43\]

where $X_{\text{lt}(i)}$ is the Lockhart-Martinelli parameter given by equation [2.32]. The momentum pressure drop is found by applying a momentum balance to the module which yields,

$$\Delta P_{\text{mom}(i)} = -G_{\text{ref}}^2 \left\{ \left[ \frac{x^2}{\rho_s \alpha} + \frac{(1-x)^2}{\rho_f (1-\alpha)} \right]_{\text{out}(i)} - \left[ \frac{x^2}{\rho_s \alpha} + \frac{(1-x)^2}{\rho_f (1-\alpha)} \right]_{\text{in}(i)} \right\}$$  \[2.44\]
where \( \alpha \) is the void fraction correlation proposed by Zivi [17] and is given by,

\[
\alpha = \frac{1}{1 + \left[ \frac{1 - \frac{x}{P_g(i)}}{x} \right]^{2/3} \left[ \frac{\rho_g(i)}{\rho_f(i)} \right]^{2/3} }
\]  

[2.45]

The gravitational pressure drop is calculated based on a homogeneous flow model and has the same general form as the single phase pressure drop except an average density is used.

\[
\Delta P_{\text{grav}(i)} = \rho_{\text{avg}(i)} g_c \frac{L_{\text{mod}(i)}}{L_{\text{cond}}}
\]  

[2.46]

where,

\[
\rho_{\text{avg}(i)} = \frac{1}{v_f(i) + x(i) \left( v_g(i) - v_f(i) \right)}
\]  

[2.47]

2.6 Property Routines

The thermodynamic property subroutines utilized by the simulation were developed by the National Institute of Standards and Technology (NIST) [7]. The use of these subroutines allows for greater flexibility in the model. There is a choice of two equations of state, the Benedict-Webb-Rubin (BWR) or the Carnahan-Starling-DeSantis (CSD) equation of state. In addition, there are coefficients for the CSD equation of state for up to 26 refrigerants and mixtures of up to five components can be analyzed. The BWR subroutines are more accurate, especially for R134a which is the reference refrigerant [8]. When using the Benedict-Webb-Rubin equation of state, properties for other refrigerants are found using the principle of corresponding states. The CSD equation of state subroutines were chosen for use in the simulation program because they allowed the flexibility of looking at mixtures.

A comparison of the enthalpy change and saturation temperatures for the two equations of state for R134a was made to determine the error that could result through the use of the less accurate CSD equation of state. It was found that there
were errors in the saturation temperatures of less than 0.5 %. For the latent heat, it was found that the error in the CSD equation of state (EOS) ranged from 3.4% to 4.7% for a saturation pressure of 150 psi to 250 psi which is shown in Figure 2.1.

The experimental data taken for the condenser coil is also analyzed with the CSD equation of state subroutines, so this error should not affect the validation of the simulation. An interface program was written for use with the FORTRAN subroutines. This interface allows the subroutines to be implemented into existing programs with relatively few problems.
3.0 Coil Geometry and Modeling

The condenser coil used for the inventory analysis is a cross-flow tube and fin heat exchanger with a fairly complex geometry. This section provides an overview of the coil geometry along with a discussion of the assumptions made in the computer modeling and the changes to the simulation required to include the manifold and return bend pressure drop.

The inlet to the condenser coil is a manifold which is divided into a number of sections where the refrigerant flows in and out of the refrigerant tubes. In each of these sections, a number of tubes circulate the refrigerant through the width of the condenser and return to the manifold where the refrigerant drops into the next section of tubes. Figure 3.1 below, is a top view of the condenser coil which shows the manifold inlet and the top tube configuration.

Figure 3.1 - Condenser Top View

![Diagram of Condenser Top View](image)

Figure 3.2 on the next page, shows the number sections in the condenser coil and the number of refrigerant tubes in each section. It also indicates the air flow direction and whether the refrigerant inlet is in the front of the tube or the rear of tube. This is an important point in the condenser modeling. In section 2.1, it was shown that the modeling of a refrigerant tube whose air inlet was in the front and refrigerant inlet was in the rear required the additional Newton-Raphson variable
which determined the average air outlet temperature from the front of the refrigerant tube to use as the air inlet temperature to the back of the refrigerant tube. This will greatly affect the number of segments required to model this coil.

Figure 3.2 - Condenser Front View

Section 1
Set of 4 tubes
Rear Tube Inlet

Section 2
Set of 4 tubes
Front Tube Inlet

Section 3
Set of 3 tubes
Rear Tube Inlet

Section 4
Set of 3 tubes
Front Tube Inlet

Section 5
Set of 3 tubes
Rear Tube Inlet

Section 6
Set of 3 tubes
Front Tube Inlet

Section 7
Set of 2 tubes
Rear Tube Inlet

Manifold

Air Flow

Return Bends Not Shown

Rear Tubes Not Shown

Drawing not to scale
It should be noted that the distance between manifold sections in figure 3.2 has been exaggerated to enhance the clarity of the figure. This type of geometry requires the use of the fixed length version of the simulation program, because it is not possible to specify before hand the inlet and outlet quality of the segments. These values will be determined by the simulation. For this coil geometry, the condenser coil is divided into ten segments. The manifold sections which have a rear inlet are modeled as one segment (sections 1, 3, 5 and 7), alternatively, the sections with a front inlet are modeled as two segments (sections 2, 4 and 6).

As stated previously the coil is modeled by ten segments. It is assumed that the inlet conditions to all of the tubes in each manifold section are equivalent. This allows the use of only one tube in the Newton-Raphson solution. Table 4.1 below shows the number of modules each segment was divided into for the inventory results presented in section 4.4.

<table>
<thead>
<tr>
<th>Manifold Section</th>
<th>Segment Number</th>
<th>Number of Modules</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>1</td>
<td>10</td>
</tr>
<tr>
<td>2</td>
<td>2</td>
<td>5</td>
</tr>
<tr>
<td></td>
<td>3</td>
<td>5</td>
</tr>
<tr>
<td>3</td>
<td>4</td>
<td>10</td>
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<tr>
<td>4</td>
<td>5</td>
<td>5</td>
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<td></td>
<td>6</td>
<td>5</td>
</tr>
<tr>
<td>5</td>
<td>7</td>
<td>10</td>
</tr>
<tr>
<td>6</td>
<td>8</td>
<td>5</td>
</tr>
<tr>
<td></td>
<td>9</td>
<td>5</td>
</tr>
<tr>
<td>7</td>
<td>10</td>
<td>10</td>
</tr>
</tbody>
</table>

Table 4.2 gives some of the overall condenser dimension needed to run the simulation.
Here the total condenser tube length is used in the weighting function to determine the gravitational pressure drop in the module given by equations [2.41] and [2.46]. The frontal condenser tube length is used to determine the air flow rate over a module given by equation [2.2] and the air side heat transfer coefficient in equation [2.21]. The frontal length is used because this represents the ratio of the area which the module air flow occupies to the total air flow area assuming the refrigerant tubes are equally spaced.

Certain assumptions are made to determine the refrigerant properties along the condenser length. The first assumption made is that refrigerant mass flow rate entering the tubes from the manifold is uniformly divided among the number of tubes giving,

\[ \dot{m}_{R \ mod} = \frac{\dot{m}_{R \ tot}}{n_{\ tubes}} \]  

[3.1]

In addition, the inlet conditions to each tube in the manifold section are assumed to be equal (the inlet pressure, temperature and enthalpy). The return bends and manifold are assumed to be adiabatic since they are not exposed to the airflow. However, the pressure drop across the bends and manifold were included in the simulation.

For example, the inlet pressure and temperature to the condenser is specified by the user. The program determines the condenser inlet enthalpy and the pressure
drop in the first manifold. So the inlet conditions to the first segment are the enthalpy at the inlet of the condenser and the inlet pressure minus the manifold pressure drop. With these two thermodynamic properties set, all the additional segment inlet conditions are found using the thermodynamic property routines. The simulation then solves the Newton-Raphson problem for the first segment which provides the outlet conditions of the first segment. To determine the inlet conditions for the second segment, the inlet enthalpy is equal to the outlet enthalpy of the first segment (the manifold is adiabatic). The inlet pressure is the outlet pressure of the first segment minus the manifold pressure drop. Return bend pressure drops are calculated only in manifold sections which are modeled with two segments.

This procedure continues until all segments of the condenser have been analyzed. The total mass of the condenser is then given by,

$$ m_{\text{total}} = \sum_{m=1}^{p} n_{\text{tubs}(m)} \sum_{i=1}^{n} m_{\text{mod}(i)} $$

[3.2]

If there were only one refrigerant tube per segment then the total mass of the condenser is given by,

$$ m_{\text{total}} = \sum_{i=1}^{n} m_{\text{mod}(i)} $$

[3.3]

The pressure drop correlations for the manifolds and return bends were developed by Paliwoda [23]. He developed a generalized method for determining the pressure drop across pipe components with two-phase flow. This includes, manifolds, return bends, T-junctions, valves, etc. The general component pressure drop correlation for the condenser is given by,

$$ \Delta P_{\text{com}} = \frac{G_{\text{ref}}^2 \times \xi \times \beta_c}{2 \times \rho_g \times g_c} $$

[3.4]
For this condenser's manifold,

\[ \xi = 2.7 \quad [3.5] \]

and,

\[ \beta_c = [\theta + 0.58x(1 - \theta)][1 - x]^{0.333} + x^{2.276} \quad [3.6] \]

where,

\[ \theta = \frac{\rho_g}{\rho_f} \left[ \frac{\mu_f}{\mu_g} \right]^{0.25} \quad [3.7] \]

For the return bends,

\[ \xi = 0.12 \quad [3.8] \]

and,

\[ \beta_c = [\theta + 3.0x(1 - \theta)][1 - x]^{0.333} + x^{2.276} \quad [3.9] \]

Overall it was found that the frictional pressure drop in the return bends was on the same magnitude as was found for the frictional pressure drop in straight modules. The manifold frictional pressure drop on the other hand was quite large in comparison to the pressure drop in straight modules. The next section of the paper discusses the void fraction correlations and the charge inventory equations used to find the mass of the modules.
4.0 Prediction of Inventory

The simulation program solves for the outlet conditions for each module, the temperature, pressure, quality, enthalpy, etc. It is these values which are needed to determine the refrigerant mass in each module using a void fraction correlation. Five different correlations were added to the simulation for comparison purposes. These are the homogeneous void fraction model [21], the Domanski and Didion correlation [11], the Tandon et al. correlation [10], the Premoli et al. correlation [12] and the Hughmark correlation [14]. Section 4.1 presents an overview of the general governing equations needed to calculate the refrigerant inventory. In section 4.2 the specific format for each of the void fraction models is presented. The different solution techniques are discussed in section 4.3 and results are presented in section 4.4.

4.1 General Equations

The general refrigerant inventory equations are provided for both the single phase region and two phase regions of the condenser. In the single phase region, the condenser mass is a function of the density and volume of the module only. In the two phase region, the mass is also a function of the quality which leads to a more complicated integral equation as will be shown. The mass of each module is determined using its inlet and outlet conditions, and then the total condenser mass is found by taking the summation of all the module masses.

\[ m_{\text{total}} = \sum_{i=1}^{n} m_{\text{mod}(i)} \]  \[4.0\]

The general void fraction equations are available in the literature and Rice [9] provides an excellent summary of these equations.
Single Phase Region

In the single phase region of the condenser, the refrigerant mass is a function of the density and module volume.

\[ m_{\text{mod}(i)} = \int_0^L A_i \rho_{\text{(i)}} \cdot dl \]  

[4.1]

where \( L = L_{\text{mod}(i)} \). Since the cross-sectional area of the condenser is constant (the tube inside diameter is constant), this equation can be integrated and the result is:

\[ m_{\text{mod}(i)} = V_{\text{mod}(i)} \times \rho_{\text{avg}(i)} = A_i \times L_{\text{mod}(i)} \times \rho_{\text{avg}(i)} \]  

[4.2]

For better accuracy, the average density across the module is used.

Two Phase Region

In the two phase region of the condenser, the total mass of the module can be found by adding the mass of the liquid and the mass of the vapor together.

\[ m_{\text{mod}(i)} = m_{\text{vap}(i)} + m_{\text{liq}(i)} \]  

[4.3]

To determine the mass of the vapor and liquid in the module the same integral that was used in equation [4.1] is performed for the vapor and liquid refrigerant separately. The mass of the vapor present in the module is given by,

\[ m_{\text{vap}(i)} = \int_0^L \rho_g A_g \cdot dl \]  

[4.4]

and the mass of the liquid present in the module is given by,

\[ m_{\text{liq}(i)} = \int_0^L \rho_f A_f \cdot dl \]  

[4.5]

The void fraction \( \alpha \) is defined as the ratio of the cross-sectional area occupied by the vapor to the total tube cross-sectional area, \( \alpha = A_g / A_i \). Equations [4.4] and [4.5] can then be written in the following form,

\[ m_{\text{vap}(i)} = \rho_g A_i \int_0^L \alpha \cdot dl \]  

[4.6]

and,
The total refrigerant mass in the module is then given by,

\[ m_{\text{mod}(i)} = A_i \left[ \rho_g \int_0^L \alpha \cdot dl + \rho_f \int_0^L (1 - \alpha) \cdot dl \right] \tag{4.8} \]

The void fraction is generally a function of quality so it is desired to transform the integral over the length of the module to an integral over the quality of the module. In order to do this, an assumption regarding the heat flow must be made to determine the relationship between the quality and the tube length \[9\]. The mass integral equation \[4.8\] is then normalized with this function and integrated over the quality. The total heat transferred from a module can be represented by the following equation,

\[ Q = \int_{x_i}^{x_o} m_{\text{mod}} h_f dx = \int_0^L f_Q(x) dl \tag{4.9} \]

Recognizing that the refrigerant mass flow rate and the latent heat of vaporization are constants, the integral in equation \[4.6\] for the mass of the vapor can be normalized and the result is a heat flux averaged void fraction \( W_g \).

\[ W_g = \frac{\int_{x_i}^{x_o} \left[ \alpha(x) / f_Q(x) \right] dx}{\int_{x_i}^{x_o} \left[ 1 / f_Q(x) \right] dx} \tag{4.10} \]

Equation \[4.8\] then becomes,

\[ m_{\text{mod}(i)} = A_i l_{\text{mod}(i)} \left[ \rho_g W_g + \rho_f \left( 1 - W_g \right) \right] \tag{4.11} \]

Now the calculation of the module mass is dependent on evaluating the integral in equation \[4.10\]. In order to do this, as stated earlier, an assumption regarding the heat flux along the length of the tube must be made. The simplest assumption which can be made is that the heat flux is constant, in which case, it drops out of the integral leaving,

\[ W_{g(i)} = \frac{\int_{x_{in(i)}}^{x_{out(i)}} \alpha dx}{x_{out(i)} - x_{in(i)}} \tag{4.12} \]
This assumption is dependent on their being a linear relationship between the quality and tube length. The design of the simulation program is well suited for this assumption if a large number of modules is used. If a large number of modules is used, the change in quality along the length of the module will be small and could therefore, be approximated by a linear distribution without much error. This will be investigated in more detail in the results section. The next section outlines the void fraction models which will be used in this study.

4.2 Void Fraction Correlations

The void fraction correlations available in the literature were reviewed by Rice [9] who categorized them into four groups:

1. Homogeneous
2. Slip ratio correlated
3. $X_{tt}$ correlated
4. Mass flux dependent

The Domanski-Didion correlation can be categorized as $X_{tt}$ correlated while the Premoli and Hughmark correlations are categorized as mass flux dependent. The Tandon correlation is both $X_{tt}$ correlated and mass flux dependent. No slip ratio correlated void fraction models were used in the simulation program.

**Homogeneous**

The homogeneous void fraction is the most simplistic form available. It is a function of the quality and the vapor-liquid density ratio. The general form of the equation is given by [21],

$$
\alpha = \frac{1}{1 + \left( \frac{1 - x}{x} \right) \frac{\rho_g}{\rho_f}}
$$

This void fraction model along with the constant heat flux assumption can be integrated over the quality change of the module to obtain a closed form solution for the homogeneous void fraction. This solution will be provided in section 4.3.
Domanski and Didion

The Domanski and Didion [11] void fraction correlation is based on the Lockhart-Martinelli correlating parameter $X_{tt}$ and the data available from the pressure drop work of Lockhart-Martinelli [22]. It was developed by Domanski and Didion for use in a heat pump simulation. The general formulation is,

$$\alpha = \left[ 1 + X_{tt}^{0.8} \right]^{-0.378}$$

for $X_{tt} \leq 10$ \[4.14\]

and,

$$\alpha = 0.823 - 0.157 \ln X_{tt}$$

for $X_{tt} > 10$ \[4.15\]

where,

$$X_{tt} = \left[ 1 - \frac{x}{x} \right]^{-0.9} \left[ \frac{\rho_g}{\rho_f} \right]^{0.5} \left[ \frac{\mu_f}{\mu_g} \right]^{-0.1}$$ \[4.16\]

Tandon

The Tandon et. al. [10] void fraction equation includes both the effects of frictional pressure drop and mass flux by correlating as a function of the liquid Reynolds number and the Lockhart-Martinelli correlating parameter. The void fraction equations are,

$$\alpha = \left[ 1 - \frac{1.928 \, Re_f^{-0.315}}{F(X_{tt})} + \frac{0.9293 \, Re_f^{-0.63}}{F(X_{tt})^2} \right]$$

for $50 < Re_f < 1125$ \[4.17\]

or,

$$\alpha = \left[ 1 - \frac{0.38 \, Re_f^{-0.088}}{F(X_{tt})} + \frac{0.0361 \, Re_f^{-0.176}}{F(X_{tt})^2} \right]$$

for $Re_f \geq 1125$ \[4.18\]

where,

$$F(X_{tt}) = 0.15 \left[ 1 / X_{tt} + 2.85 / X_{tt}^{0.476} \right]$$ \[4.19\]

and

$$Re_f = \frac{G_{ref} \times D_i}{\mu_f}$$ \[4.20\]
The Premoli et al. [12] correlation is a function of the liquid Reynolds number, Weber number and the surface tension. It is given by,

\[
\alpha = \frac{1}{1 + \left[ \frac{1 - x}{x} \right] \left( \frac{\rho_g}{\rho_f} \right) \times \left[ 1 + F_1 \left( \frac{y}{1 + yF_2} - yF_2 \right)^{0.5} \right]} \tag{4.21}
\]

where,

\[
F_1 = 1.578 \text{Re}_f^{-0.19} \left( \frac{\rho_f}{\rho_g} \right)^{0.22} \tag{4.22}
\]

\[
F_2 = 0.0273 \text{We}_f \text{Re}_f^{-0.51} \left( \frac{\rho_f}{\rho_g} \right)^{-0.08} \tag{4.23}
\]

\[
\text{Re}_f = \frac{G_{ref} \times D_i}{\mu_f} \tag{4.24}
\]

\[
\text{We}_f = \frac{G_{ref}^2 \times D_i}{\sigma \rho_f \rho_g} \tag{4.25}
\]

\[
y = \frac{\beta}{1 - \beta} \tag{4.26}
\]

\[
\beta = \frac{1}{1 + \left[ \frac{1 - x}{x} \right] \left( \frac{\rho_g}{\rho_f} \right)} \tag{4.27}
\]

Here we start to see that the integrals to evaluate \( W_g \) can get quite complicated as the void fraction models become more complex. It is not clear if a closed form solution even exists for these integral equations, therefore, numerical methods of evaluating the integrals is discussed in section 4.3.

The Hughmark void fraction correlation is one the most complex and requires an iterative solution. The correlation is given by,

\[
\alpha = \frac{K_H}{1 + \left[ \frac{1 - x}{x} \right] \left( \frac{\rho_g}{\rho_f} \right)} \tag{4.28}
\]
where $K_H$ is a function of the correlating parameter $Z$. These values are tabulated in Table 4.1. The correlating parameter $Z$ is given by,

$$Z = \frac{Re_a^{1/6} Fr^{1.8}}{y_L^{1.4}}$$  \[4.29\]

where,

$$Re_a = \frac{D_i \varepsilon_{ref}}{\mu_f + \alpha (\mu_g - \mu_f)}$$  \[4.30\]

$$Fr = \frac{1}{g_c \frac{D_i}{\beta \rho_g}}$$  \[4.31\]

$$y_L = 1 - \beta$$  \[4.32\]

$$\beta = \frac{1}{1 + \left( \frac{1 - x}{x} \right) \rho_g / \rho_g}$$  \[4.33\]

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</table>

These are the five void fraction correlations which were added to the condenser simulation program and will used for the comparison of predicted condenser charge. The next section reviews the solution techniques used to evaluate the integral $W_g$.

4.3 Integral Solution Techniques

The calculation of the refrigerant mass in each module of the condenser is dependent on evaluating the integral equation for $W_g$ for each void fraction model. Two solution techniques were explored, a closed form expression and a numerical average of the inlet and outlet void fraction values. A closed form expression for the homogeneous void fraction correlation was found. This is useful in verifying the results obtained for the numerical average approximation. For the other void
fraction correlations, the numerical average is performed. This section describes the procedure and assumptions used for these solution techniques.

Closed Form

The closed form expression is the exact solution for the heat flux averaged void fraction for the module. As the void fraction correlations increase in complexity, however, this expression is more difficult to find if it exists at all. For the homogeneous model, the integration was performed to obtain the following expression,

\[ W_g = \left. \frac{x}{c_2} - \frac{c_1}{c_2} \ln(c_1 + c_2x) \right|_{x_{in}}^{x_{out}} \]  \[ \text{[4.34]} \]

where,

\[ c_1 = \frac{\rho_g}{\rho_f} \quad \text{and,} \quad c_2 = 1 - c_1 \]  \[ \text{[4.35]} \]

This closed form solution will be utilized to determine the accuracy of the numerical average technique.

Numerical Average

The numerical average technique consisted of taking a numerical average of alpha at the inlet and outlet of the module. The assumption made here is that there is a linear quality distribution along the module length. This is the same assumption which was made for the use of the constant heat flux assumption in the formulation of \( W_g \) in section 4.1 and similar reasoning applies here. For a small module length, relative to the total length of the condenser, the change in quality from the inlet to the outlet is small and \( W_g \) can be approximated by an average of the inlet and outlet void fraction. Clearly, the accuracy of this technique is dependent on this assumption being satisfied. If the entire two phase region of the
condenser is being modeled with one module, one might need to investigate additional numerical integration techniques.

4.4 Inventory Results

Simulation Validation

In section 2, a discussion of the two phase refrigerant heat transfer correlations indicated that there were some differences between the Cavallini-Zecchin and the Dobson correlations for both the prediction of the overall coil capacity and the total pressure drop. This is the first comparison provided. In Figure 4.1, a comparison of the total heat transfer is provided for the two correlations. The comparison is against the experimental heat transfer found for the same operating inlet conditions. Overall, the average magnitude of error for the Cavallini-Zecchin correlation was 6.33% while the Dobson correlation was 8.62%. The Cavallini correlation overpredicted the heat transfer in most cases while the Dobson correlation underpredicted the heat transfer. In general, the Cavallini correlation better predicted the heat transfer for this coil. The pressure drop results are shown in Figure 4.2. The average magnitude of error using the Cavallini two phase heat transfer correlation was 33.11%, while for the Dobson correlation it was 21.38%. The underprediction of heat transfer from the Dobson correlation results in a longer two phase region in which a higher pressure drop exists. This increases the overall pressure drop predicted by the simulation. The heat transfer correlations also have a significant impact on the length of the subcooling region. The Cavallini correlation overpredicted the heat transfer which resulted in a long subcooling region. The Cavallini correlation predicted an average of 14.35°F subcooling while the Dobson correlation predicted no subcooling with an average outlet quality of 0.0898. The experimental data indicated an average of 1.5°F subcooling. The Dobson correlation decreases the subcooling length and therefore
provides a better inventory prediction. Figure 4.3 shows a comparison of the charge prediction for the two correlations using the homogeneous close form void fraction.
The comparison in figure 4.3 indicates that use of the Cavallini-Zecchin heat transfer correlation greatly increases the predicted charge (an average 35.4% higher than the Dobson prediction). All further results provided will use the Dobson correlation. Also, a table of the operating conditions for the 46 experimental data points is provided in Appendix B.

Solution Method

This comparison is of the condenser charge predicted using the two integral solution techniques with the homogeneous void fraction correlation. It is desired to determine the accuracy of the predicted charge for the numerical average and the dependence on the number of modules selected for the condenser. Table 4.2 shows that for the averaging method, the error introduced into the charge prediction was equal to or less than 0.05%. The number of modules refers to the total number of modules in the condenser based on the table 3.1. Given these results, the accuracy of the numerical average is sufficient for this case.
Predicted Charge for Void Fraction Correlations

Figure 4.4 shows the predicted condenser charge for the different void fraction correlations. Overall, it is seen that the Hughmark correlations consistently predicts the greatest charge, the next highest predictor is the Premoli correlation, and the Tandon, Domanski and homogeneous correlations seem to be in the same general area.

Inventory vs. Inlet Conditions

A comparison of the predicted charge versus various conditions is made. The charge was compared with the overall heat transfer and pressure drop. In addition,
it was compared with the inlet pressure, refrigerant mass flow, and air mass flow rate. The most significant dependence of the predicted charge is on the inlet refrigerant pressure. This dependence is shown in figure 4.5 below.

Inventory Distribution

This last section of results provides a comparison of the mass distribution along the condenser length for the module vapor, liquid and total module mass. This comparison is based on the results for a single tube. For example, for this coil geometry, the first manifold section contained four tubes. Only the mass of the modules in one of these tubes will be compared. This will provide a clearer picture of what effect the quality distribution has on the predicted module charge and also what effect the refrigerant mass flux has on the module charge. There are two mass flux transitions within this coil. The first occurs between the second and third manifold sections where the number of refrigerant tubes is reduced from four to three. The second transition occurs between the sixth and seventh manifold
sections where the number of refrigerant tubes is reduced from three to two. It is desired to see what effect, if any, these transitions will have.

The comparison was made for three data points at different air mass flow rates. These points were selected for their low outlet qualities which allowed most of the condensing region to be modeled. The data points used were six, ten and forty from the Appendix. Figure 4.6 below shows the distribution for point six which had the highest air mass flow rate of the three. The Hughmark correlation predicts a larger module mass throughout the condensing section. It is also interesting to note, the mass flux transition points had no noticeable effect on the module mass.

Figure 4.7 shows the module mass distribution for point ten. In this diagram the predicted refrigerant mass increased much more sharply towards the end of the condensing region than it did in Figure 4.6. This is due in part to the low refrigerant
mass flow rate in combination with the high air flow rate which provides a faster condensing rate.

![Predicted Module Mass Distribution](image)

Predicted Module Mass Distribution
Figure 4.7 - Air Mass Flow Rate: 4092.6 [lbm/hr]

Figure 4.8, is the module mass distribution for point forty. The predicted module mass for these operating conditions shows a much more gradual increase along the length of the condenser. The refrigerant mass flow rate for this case was about the same as for point 10 while the air mass flow is about half. The first few modules in the figures represent the superheated region of the condenser where the mass is found without the void fraction correlation and so it is the same for all the curves. Figure 4.9, shows the liquid mass distribution along the length of the condenser coil for point six. It is found that the liquid distribution curves are almost the same as the module mass distribution curves. This is due to the dominating effect the liquid has the total module mass.
Predicted Module Mass Distribution
Figure 4.8 - Air Mass Flow Rate: 2197.1 \([\text{lbfm/hr}]\)

Predicted Module Liquid Mass Distribution
Figure 4.9 - Air Mass Flow Rate: 6509.8 \([\text{lbfm/hr}]\)
In Figures 4.10 through 4.12 the vapor distributions are shown for the three data points. One of the interesting results is here is that the Hughmark correlation does not predict the largest module vapor mass as was the case for the liquid mass. The homogeneous void fraction correlation actually predicts the largest vapor mass. This is also shown in Figures 4.11 and 4.12. The expected trends are indicated by the vapor mass distribution curves. The module mass prediction indicated that the module mass sharply increased towards the end of the condensing region for larger air mass flow rates. This same trend is seen in the liquid distribution which indicates that the vapor distribution must more sharply decrease towards the end of condensing region for higher air mass flow rates. This trend is shown in Figures 4.11 and 4.12.

![Predicted Module Vapor Mass Distribution](image)

**Figure 4.10 - Air Mass Flow Rate: 6509.8 [lbm/hr]**

- **Mass Flux Transition**
- **Predicted Module Vapor Mass [lbm]**
- **Condenser Length [ft]**

- **Homogeneous**
- **Domanski**
- **Tandon**
- **Premoli**
- **Hughmark**
Predicted Module Vapor Mass Distribution
Figure 4.11 - Air Mass Flow Rate: 4092.6 [lbm/hr]

Predicted Module Vapor Mass Distribution
Figure 4.12 - Air Mass Flow Rate: 2197.1 [lbm/hr]

Mass Flux Transition

Legend:
- Homogeneous
- Domanski
- Tandon
- Premoli
- Hughmark
Overall, these results reinforce the belief that the accurate prediction of the condenser mass is based upon the accurate prediction of the condenser liquid mass. In order to determine the most accurate void fraction correlation for this purpose it is necessary to measure the condenser mass. This work will be available in the future and is discussed in the next section.
5.0 Future Work

5.1 Overview
The future work consists of an experiment which measures the weight of the refrigerant in a condenser during operation. The experimental measurement will be compared with the simulation results to determine which void fraction correlation produces the best results. The condenser to be tested is different than the one discussed in this paper. It consists of micro channel tubes with headers at the inlet and outlet of the tubes. It will be modeled using the same techniques described in this paper.

5.2 Schematics

The following schematic illustrates the instrumentation and components used in the experimental apparatus for condensers. The apparatus has the capability to control the fluid flow rates and inlet conditions. The pre and after condensers provide the capability for partial condensing. A refrigerant turbulator is used at the entrance to the test section to promote thermodynamic equilibrium between the liquid and the vapor.
Condenser Apparatus

Instrumentation legend
D-differential pressure, F-flow, H-humidity, P-pressure, T-temperature, W-weight

Equipment legend
△ Shut-off valve
T Schrader valve
♀ Manual expansion valve
Capillary tube
Sight glass
Air filter
Air blender
Airflow Measurement Station w/ Nozzles
Supply Fan w/ Speed Controller
Filter
Electric Duct Heater w/ SCR Controller
Filter/Drier
Sight glass
Air filter
Air blender
Air In
Air Out
Subcooling Control
Schrader Valve
Receiver
Air Loop
Test Condenser
Water cooled after-condenser
Water cooled pre-condenser
Refrigerant Turbulator
Airflow Measurement Station w/ Nozzles
Air Loop
Test Condenser
Water cooled after-condenser
Water cooled pre-condenser
Refrigerant Turbulator
Refrigerant Loop
The details of the weight measurement system are given in the following figure.

5.3 Proposed Solutions to Weight Measurement Problems
Belth and Tree [25] describe a design to accurately measure the weight of refrigerant in each of the components of a heat pump. On the following page is a list of problems they encountered, along with the current design's proposed solutions to these problems.

<table>
<thead>
<tr>
<th>Weight measurement problem</th>
<th>Proposed solution</th>
</tr>
</thead>
<tbody>
<tr>
<td>Component weight greater than refrigerant mass</td>
<td>Use a balance beam with a counter weight</td>
</tr>
<tr>
<td>Refrigerant movement causes false readings when using a balance with a single pivot point</td>
<td>Suspend the condenser from above using a cable</td>
</tr>
</tbody>
</table>
Drag and lift forces cause errors in the mass measurement | With no refrigerant flow the lift forces will be measured for different air flow rates
---|---
The stiffness of the piping and duct connections affect the vertical movement of the component | Flexible ducting made of 5 mil plastic will be used between the coil and the fixed duct. Flexible refrigerant hose will be used to connect the component to the rest of the loop.
Rigid mounting of the cantilever beam transducers limited the deflection of the balance beam | Mount the transducer so there can be movement between the transducer and the balance beam
Friction or damping from the pivot points, when bearings were used, caused a hysteresis loop the same magnitude as the expected output | Make all pivot points knife edges
The balance beam vibrated | Immerse the counter weight into an oil bath to dampen the vibrations.

**5.4 Test Plan**

Before testing can begin, the weight measurement system will be calibrated. This calibration will account for two different effects. The first is the effect of the flexible connections on the mass measurement. This will be accounted for by placing a series of known masses, covering the range of expected refrigerant mass, on top of the empty test section and recording the output of load cell. Then the offset and span can be determined from a curve fit of this data.

The second effect is the lift force caused by air flow over the coil. This will be accounted for by recording the output of the load cell at different air flow rates that cover the entire operating range of the condenser. The output of the load cell will be corrected in an analysis program using the relationships developed during the calibration.

The test plan may consist of two phases. In phase one the refrigerant will enter the test section at or near saturated vapor conditions, and exit the test section at or near saturated liquid conditions. In other words, complete condensation will occur
during the testing. After the first phase the data will be analyzed to determine if partial condensing data are needed. Many of the void fraction correlations were developed for a particular flow regime (e.g. Tandon-annular flow) and may not be able to be applied over the full condensing region. If this is the case then the different flow regimes may need to be studied separately.

Since it is not possible to observe the flow regimes in the tubes of the condenser, some criteria will be needed to determine where the flow regime changes. Tests of a compact heat exchanger (Dh=1.74 mm) by Damianides and Westwater [26] do provide a starting point. They tested water-air mixtures at 5.4 atmospheres. The superficial liquid velocity varied from 0.0838 to 8.62, and the superficial gas velocity was from 1.05 to 101.2 m/s. They observed that smooth stratified and wavy stratified flow regimes were absent. Comparing the test envelope with their flow map suggests that there will be two flow regimes: intermittent and annular. If phase two testing is necessary, a dimensional analysis will be conducted to check the validity of the Damianides flow map for this application.

The exit header introduces a significant amount of uncertainty (~35%) in the weight measurement. For this reason some method of flow visualization will be implemented to qualitatively access the amount of liquid in the exit header. With the flow visualization technique we hope to reduce this uncertainty to within 10%.
6.0 References


### Appendix

This appendix provides a table of the operating conditions for the 46 experimental data points used in the results comparison.

<table>
<thead>
<tr>
<th>No.</th>
<th>Ref. Inlet Temp. °F</th>
<th>Ref. Inlet Pressure psi</th>
<th>Air Inlet Temp. °F</th>
<th>Air Inlet Pressure psi</th>
<th>Relative Humid.</th>
<th>Ref. Flow Rate lbm/hr</th>
<th>Air Flow Rate lbm/hr</th>
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