PARAMETER ESTIMATION FOR HVAC SYSTEM MODELS FROM STANDARD TEST DATA

by

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DEDICATION

I would like to dedicate this thesis study to my family. Without their unconditional support and patience I may not have had the endurance to finish.

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something I have always and will always rely on. I love you all more than I will ever be able to express.

ABSTRACT

Nearly all cooling systems, and an increasing proportion of heating systems, utilize the vapor compression cycle (VCC) to provide and remove heat from conditioned spaces. Even though the application of VCC's throughout the building environment is ubiquitous, effective and accessible models of the performance of these systems remains elusive. Such models could be important tools for VCC designers, building designers and building energy managers as well as those who are attempting to optimize building energy performance through the use of model-based control systems.

Strides have been made in developing lumped parameter models for VCC's. In spite of these contributions, widespread accessibility and use of VCC performance models has yet to be achieved. This work addresses one of the barriers in applying VCC performance models, the identification of model parameter values required to make performance models useful and accurate. A steady state spreadsheet-based model has been developed which, when combined with standard test data provided by system manufacturers, allows the modeler to identify the salient heat transfer parameters that govern the behavior of the condensers and the evaporators.

Performance data provided by the system manufacturer was used to determine model parameter values. Data used from the test conditions for the determination of these parameters include the evaporating and condensing pressures, the input power, the cooling rate and the degrees of superheat and subcool. Most importantly, these data allowed for the computation of the effective heat transfer characteristics within the

moving boundaries, as opposed to heat transfer values calculated strictly from the geometry. Using an effective heat transfer value allows for the spreadsheet-based model to use a broad spectrum of VCC models despite their potential differences in heat exchanger design conditions, that is not dependent on the number and spacing of fins or other optimization design criteria.

To validate the concept, the approach was used to identify parameter values for three different air conditioning units with three different sets of performance specifications. On average the model predicted a heat absorption rate within 1.5% - 3.7% error of what was measured by the manufacturer during testing. This model requires limited sensor information to provide parameters determined under steady state conditions that can be used in a dynamic model to assist in design, control and operation of traditional VCC systems over a range of operating conditions.

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LIST OF ABBREVIATIONS AND NOMENCLATURE

Abbreviations

VCC	Vapor Compression Cycle
TX	Thermal Expansion
EE	Electronic Expansion

Nomenclature

Symbols

A	Area $[ft^2]$
C	Component Coefficient $[dimensionless]$
D	Diameter $[ft]$
h	Refrigerant Enthalpy $\left[\frac{BTU}{lbm}\right]$
L	Total Length $[ft]$
l	Boundary Length $[ft]$
M	Mass $[lbm]$
$\dot{m}$	Flow rate $\left[\frac{lbm}{hr}\right]$
P	Refrigerant Pressure $\left[\frac{lbf}{in^2}\right]$
$\dot{Q}$	Heat Transfer Rate $\left[\frac{BTU}{hr}\right]$
s	Refrigerant Entropy $\left[\frac{BTU}{lbm*R}\right]$
T	Temperature $[F]$

U	Effective Heat Transfer per Unit Length $\left[\frac{BTU}{hr*ft*F}\right]$
$\dot{W}$	Work $\left[\frac{BTU}{hr}\right]$
x	Refrigerant Quality $[dimensionless]$
α	Convective Heat Transfer Coefficient $\left[\frac{BTU}{hr*F}\right]$
$\bar{\gamma}$	Mean Void Fraction $[dimensionless]$
η	Efficiency $[dimensionless]$
ν	Refrigerant Specific Volume $\left[\frac{ft^3}{lbm}\right]$
Υ	Effective Displacement Volume $[ft^3]$
ρ	Refrigerant Density $\left[\frac{lbm}{ft^3}\right]$
ω	Compressor Motor Shaft Speed $\left[\frac{rad}{hr}\right]$

<u>Subscripts</u>

$2, 2', 2'', 3$	Zone Numbers for Condenser
$4, 4', 1$	Zone Numbers for Evaporator
$4f$	Saturated Fluid at State 4
$4g$	Saturated Vapor at State 4
C	Condenser
$c1, c2, c3$	Superheated, Two-Phase, Subcooled zones in condenser
ci	Inner Tube of Condenser
co	Condenser Tubing to Ambient Air
$cr1, cr3$	Average Values in Superheated, Subcooled Zones of Condenser
$cw1$	Average Wall Temperature in Superheated Zone of Condenser

E	Evaporator
$e1, e2$	Two-Phase, Superheated zones in evaporator
ea	Temperature of Conditioned Space
$er2$	Average Values in Superheated Zone of Evaporator
ei	Inner Tube of Evaporator
H	Total Heat Rejected at the Condenser
in	Input
k	Compressor
L	Total Heat Absorbed at the Evaporator
o	Outer Diameter of Heat Exchanger Tubing
oa	Outside Air Temperature
s	Entropy
sat	Two-Phase Region of Evaporator and Condenser
SH	Superheat Region of Evaporator and Condenser
sub	Subcool Region of Condenser
V	Flow Restrictor

CHAPTER ONE: INTRODUCTION

Energy Efficiency

The worldwide demand for electricity has driven a growing interest in conservation, renewable generation and energy storage. When it comes to appetites for energy usage, Americans are the most voracious in the world. As a nation, we only represent 5% of the total world's population yet we consume 20% of the total energy produced (World Population Balance 2001 - 2014). This suggests that if anyone has the ability to improve efficiency it is the American population. As a nation, Americans have become accustomed to luxuries that not everyone enjoys. For example, in 2015 it was found that approximately 87% of American homes and residences utilize an air-conditioning system of some sort and that percentage continues to increase (Sivak 2015). The main purpose of air conditioning is simply to make the occupant of a building more comfortable during warm weather cycles. It is interesting to note that such a system is highly used and yet the average owner of the system knows nothing about it other than the settings on the thermostat. There are ways in which one can conserve overall energy consumption and improve efficient use of energy while maintaining a level of comfort. However, his is not achievable unless there is an understanding of the system and its functionality.

VCC Cycle

Nearly all cooling systems, and an increasing proportion of heating systems, utilize the vapor compression cycle (VCC) to provide and remove heat from conditioned space by use of refrigerant filled tubing (Cengel and Boles 2008). In cooling systems, this refrigerant is used to transfer heat from the air inside a conditioned space to the ambient outside air. Since the first applications in the nineteenth century, great strides have been made in the design and operation of these mechanical refrigeration systems, yet, in order to increase the design efficiency of these units the ability to model their performance must also evolve (Refrigerator 2016). Traditionally, the most practical way of studying the system cycle performance is through mathematical modeling. Standard science and engineering formulas are applied to mathematically describe processes occurring within a given cycle. This is a key first step in simulation and optimization modeling.

Originally, the process for modeling VCC units required reviewing the performance curves of the various components involved in the system. As conditions changed, the ideal operating point was found by locating the intersection of the appropriate component performance curves. This was a very graphical process requiring a large amount of empirical data for each model and design iteration. Unfortunately, using this approach, there was no ability to get real time results on how the machinery was operating. More recently, with the assistance of computer modeling software, research has been done on the best way to model the thermal performance of a VCC unit.

One approach to modeling VCC systems is the “moving boundary method”. This approach is popular because it provides a computationally efficient and effective way of capturing the complexities of the heat exchangers within the overall system. A key element of this approach is that the evaporator and condenser are modeled as two and three lumped elements, respectively, the lengths of which can change in response to changing conditions. Work by (X.-D. He 1996) is some of the first and most detailed applications of this method. More recently (McKinley and Alleyne 2008) have expanded upon this research to include a more common heat exchanger design with the inclusion of fins to the condenser and evaporator tubing. This method incorporates the heat transferred between the refrigerant to the tubing, through the tubing, the tubing to the fins, and the fins to the outside ambient air as well.

Past models have been developed to determine the output of the dynamic system for one specific air conditioner that could be tested in an engineering lab. While this is a step in the right direction, there are shortcomings to these models, specifically their applicability to a variety of air conditioning systems. All of the provided research is only applicable for a specific model of air conditioning unit and requires a rigorous testing program for each new model.

This thesis describes a physics-based model that uses steady state conditions, as can be found in manufacturers' test data, to determine model parameters that can be used in a variety of dynamic and steady-state models for energy saving estimations. This approach is adaptable to a wide variety of air conditioner specifications and sizes. The model is not dependent on a specific heat exchanger design as it utilizes an effective value that will accommodate current designs as well as future innovations. The model uses empirically driven values measured by the air conditioner manufacturer during testing. This provides values that are not theoretically derived yet do not require extensive testing for the user to facilitate to obtain the parameters required to run the model.

While this model is based on steady state conditions the parameters determined within the analysis can be applicable to a dynamic model as well. A model, dynamic or not, is only as good as the parameters for which it is based upon. Furthermore, because this tool is adaptable for a variety of VCC units, all that is needed to run the analysis are the air conditioner specifications and test conditions provided from the manufacturer, typically located on their website for the product.

CHAPTER TWO: REFRIGERATION CYCLE

The very first space conditioning system was invented in 1851 by Dr. John Gorrie to reduce diseases, like malaria. His thought was that by keeping patients cool and comfortable their recovery time would be sped up and the spread of the contagion would be greatly reduced (Lester 2015). The research and development in cooling systems from there was very slow to gain traction. In 1902 Willis Haviland Carrier began the initial design for the modern air conditioner. By the 1920's air conditioning in public buildings became increasingly popular. This increase in popularity was due to American attendance at the local movie theaters to see their favorite Hollywood stars on the big screen. Again, after many years of scientific development, installment of central air conditioning in an individual's household substantially increased in the 1970's (Green 2015). The rapid increase in market penetration of air conditioning systems served to exacerbate the energy crisis of the 1970's. To assist in the resolution of this crisis, laws were passed to set equipment standards for air conditioners and reduce overall energy consumption. These regulations and design condition requirements have been the basis of the standards that are still in effect today.

Improvements in the design of air conditioning units have become a point of interest for many mechanical engineers. These systems, ranging in sizes from a typical residential window unit to a large cooling system for a data center, can initially seem like a simple one; take the hot air and replace it with cool air. However, as a person starts to uncover the layers involved and the design improvements made they will soon realize that this system is in fact extremely complicated and underappreciated.

Ideal VCC System

In an ideal system, the refrigerant in a VCC leaves the evaporator as a saturated vapor and immediately enters the compressor, shown as the first state in Figure 1. The saturated vapor is then compressed causing both the temperature and the pressure of the refrigerant to increase. Since the temperature is increased during compression the refrigerant will then be forced into a superheated state at the exit of the compressor and the entrance of the condenser, shown as state two. The ideal cycle assumes isentropic compression.

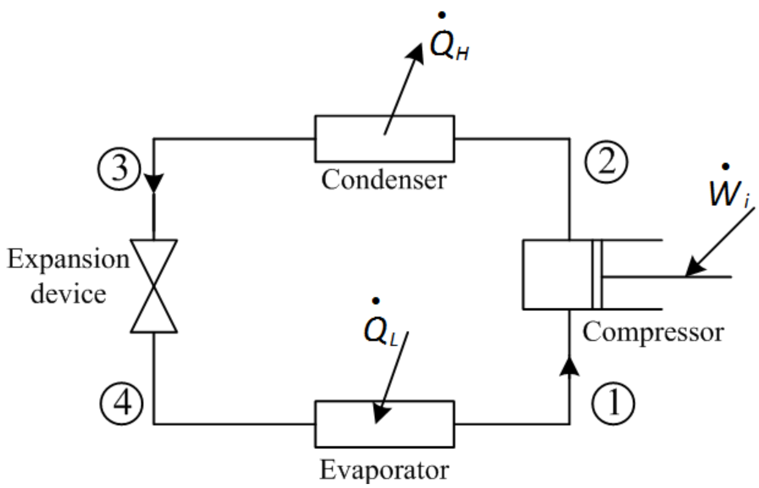

Figure 1: Ideal Vapor Compression Cycle. Reprinted from Kissock, Kelly. "Energy Efficient Buildings: Chillers." Dayton, OH: Unitversity of Dayton, January 2012.

From state two to state three the refrigerant goes through a condenser which is typically located outside the conditioned building. During this passage the excess heat removed from the conditioned space and added by the compressor is rejected to the outdoor environment. The temperature entering the condenser must be high enough to allow the heat to spontaneously flow to the environment. It would be incorrect to assume that the temperature entering the condenser is constant and the same year around because heat rejection rates will change depending on atmospheric temperatures, which change from day to day.

From states three to four, the refrigerant goes through a flow restricting, or throttling, valve, assumed isenthalpic, which will reduce both the temperature and the pressure. The temperature is reduced low enough that it will absorb the excess heat from the conditioned space while in the evaporator where it transitions from state four to state one, thus completing the cycle. As a reversal to the condenser operation, the refrigerant in the evaporator must be low enough to enable heat transfer from the conditioned space to the evaporator. This process is dependent on the set point temperature of the conditioned space and can be changed at any time during operation. A view of the components of a VCC was seen earlier and a thermodynamic graph of this ideal cycle at each refrigerant state can be seen in Figure 2.

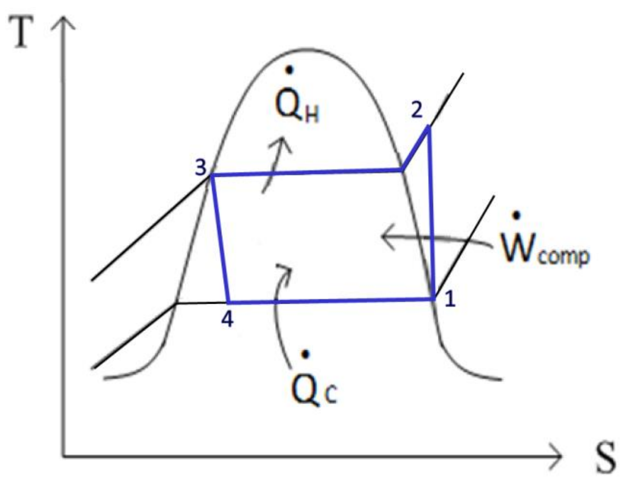

Figure 2: Temperature / Entropy Diagram for Ideal Vapor Compression Cycle

This diagram shows that the temperature and pressure remain constant throughout the evaporator and that the pressure alone remains constant throughout the condenser. There is a substantial decrease in temperature due to refrigerant superheat in the first portion of the condenser but the diagram reflects that this temperature reduction happens very quickly and then remains constant across majority of the condenser.

Actual VCC System

There are safety factors built into mechanical designs to keep the system from failing, and air conditioners are no different. It cannot be stressed enough that the views in Figure 1 and Figure 2 are for an idealized system. A thermodynamic view of a more realistic refrigeration cycle can be seen in Figure 3.

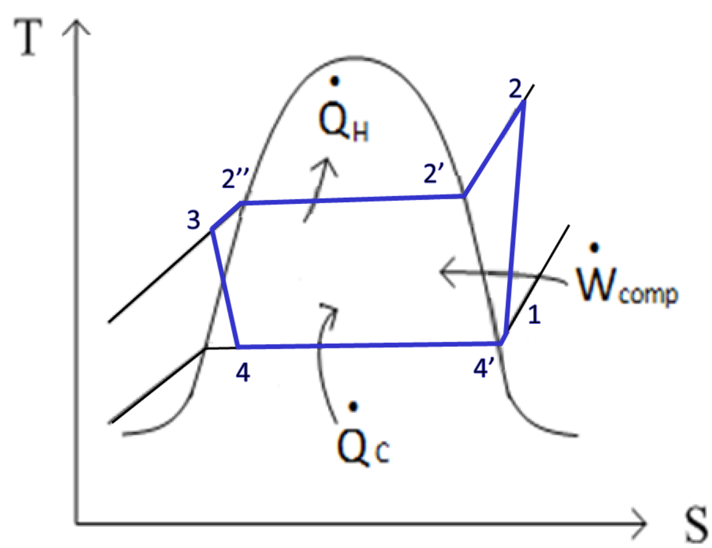

Figure 3: Temperature / Entropy Diagram for Real Vapor Compression Cycle

One can see that there are some changes when comparing Figure 2 and Figure 3. These changes are most obvious at the evaporator exit, the condenser exit and the non-isentropic behavior of the compressor between states one and two. While Figure 3 is considered the “actual VCC system” it is important to note that this system still assumes an isobaric relationship across the heat exchangers, this is an assumption that could affect the results but, due to the low flow rate of refrigerant relative to the tubing diameters that are typically seen in these systems, it is an assumption that is widely used in the research for this thermodynamic process.

For the analysis laid out within this paper the actual VCC system will be employed with reference to the states as shown in Figure 3. When comparing the ideal cycle to the actual cycle there are many additional states within the system that appear. These states and their refrigerant properties are shown below in Table 1.

Table 1: Refrigerant Phases

State 1	Refrigerant leaves evaporator as a superheated vapor and enters the compressor
State 2	Refrigerant leaves compressor as a superheated vapor with increased pressure and enters the condenser
State cr1	Average state values in first condenser region; between State 2 and State 2' (to be used in later equations)
State 2'	Refrigerant within condenser phase changes from superheated vapor to saturated vapor then a liquid / vapor two-phase combination
State 2''	Refrigerant within condenser phase changes from a liquid / vapor two-phase combination to a saturated liquid
State cr3	Average state values in third condenser region; between State 2' and State 2'' (to be used in later equations)
State 3	Refrigerant leaves the condenser as a subcooled liquid and enters the flow restrictor
State 4	Refrigerant leaves the flow restrictor as a liquid / vapor two-phase combination and enters the evaporator with a reduced pressure
State 4'	Refrigerant within the evaporator phase changes from a liquid / vapor two-phase combination to a saturated vapor
State er2	Average state values in second evaporator region; between State 4' and State 1 (to be used in later equations)

Evaporator

The main purpose of the VCC is to remove heat from a conditioned space. The evaporator assembly of the system cools the air within the room by absorbing the excess heat. Refrigerant is used in air conditioning systems because of its ability to easily go through phase changes and absorb heat quickly and efficiently. Air conditioners are often designed with a specific refrigerant type in mind. The refrigerant chosen is dependent on environmental considerations, cost, and the ability to optimize the ease of phase change. It is during this phase change within the evaporator that allows for the highest heat absorption rate allowable for the design of the unit.

In the evaporator, the refrigerant enters the evaporator coil immediately after leaving the thermal expansion valve at state four. At this point the refrigerant is in the two-phase region including both vapor and liquid properties. The quality of the mixture defines what portion of the refrigerant is in the liquid state and what portion is in the vapor state. As the refrigerant within the evaporator begins to absorb the heat from the conditioned space more of the liquid evaporates. This process continues and eventually the refrigerant leaves the evaporator as a superheated vapor.

Compressor

In the actual VCC measures are taken to ensure the system is working correctly with no failure. For example, at state one, the temperature is actually pushed into the superheat region by metering the flow into the evaporator. This ensures that the refrigerant entering the compressor contains no liquid particles. If there is liquid entering the compressor the compressor will not work properly and there will be substantial capital costs to fix or replace that component of the system.

As the refrigerant is compressed, both the temperature and the pressure rise. To better model a realistic compressor, an isentropic efficiency is used. Many sources confirmed that, typically, for modern air conditioning units, this efficiency ranges from 80% - 90%. Once the refrigerant leaves the compressor it has been pushed further into the superheated region at a much higher pressure and temperature as it enters the condenser so that the heat can easily be rejected into the outside atmosphere.

Condenser

In contrast to the evaporator, the main purpose for the condenser is to reject the heat absorbed by the evaporator and the compressor work, which was converted to heat, to the atmosphere. Much like the evaporator, the condenser function is highly dependent on the phase change of the refrigerant. Once the compressor discharges the superheated refrigerant to the condenser it begins to reject heat to the atmosphere. Once enough heat is rejected, the refrigerant becomes saturated vapor. Further heat loss transforms the refrigerant into a two-phase mixture of liquid and vapor. The heat is continually being rejected to the atmosphere causing the refrigerant to eventually condense into a liquid. The two-phase portion of the heat exchanger accounts for majority of the heat transfer available to this component of the system. In most cases, the additional capacity beyond this point and the refrigerant continues to reject heat until it leaves the condenser as a subcooled liquid.

Flow Restrictor

Like the refrigerant entering the compressor the refrigerant entering the flow restrictor must be monitored to ensure correct operation of the system. This is why the refrigerant leaving the condenser must be a subcooled liquid, to avoid any vapor particulates entering the flow restrictor and causing it to operate inefficiently. The purpose for the flow restrictor is to reduce the pressure of the refrigerant, thus bringing about a drop in temperature. In fact, this reduction is so severe and sudden it will change the phase of the refrigerant from a subcooled liquid to a two-phase refrigerant.

As can be expected there are various components in this system that can be modified to improve efficiency and decrease energy consumption. These components tend to come at a cost to the manufacturer so they have the option on deciding what improvements they are willing to incorporate into their model. One of the components that are the easiest to modify is the flow restrictor. Every air conditioning system has a flow restrictor but the complexity of this component may vary. While there are many types of flow restrictors the three most common are the fixed orifice, the electronic expansion valve and the thermal expansion valve.

Fixed Orifice

The fixed orifice design restricts the flow regardless of operating conditions. Due to its simplified nature this is the easiest flow restrictor to compute and model. Because the component is unchanging both the valve coefficient and the area remain constant regardless of operating conditions. This component is the most economical option and is found in most residential air conditioning systems because reliability is often more highly valued than efficiency.

Electronic Expansion Valve

The electronic expansion valve, EE valve, is a component that can be used to increase the overall performance of the unit. Often the design of this valve incorporates a needle valve that is controlled by a stepper motor and modeled using nonlinear static equations. In the case of the EE valve the valve coefficient can be considered consistent over a small range of operating conditions, yet the area remains variable as it is changing to increase or reduce the flow rate of the refrigerant (Rasmussen and Hariharan 2010).

<u>Thermal Expansion Valve</u>

It is most common for commercial air conditioners to utilize a thermal expansion valve, TX valve, to restrict refrigerant flow. A TX valve uses mechanical feedback to regulate the amount of superheat achieved for a variety of operating conditions. A sensing bulb is fastened to the refrigerant outlet of the evaporator. This bulb is filled with two-phase refrigerant and as the temperature of the system raises the saturation pressure within the sensing bulb increases as well. This pressure acts on a diaphragm inside the valve causing it to open and increase fluid flow to the evaporator and thus reducing the degree of superheat. A pictorial view of this process can be seen in Figure 4.

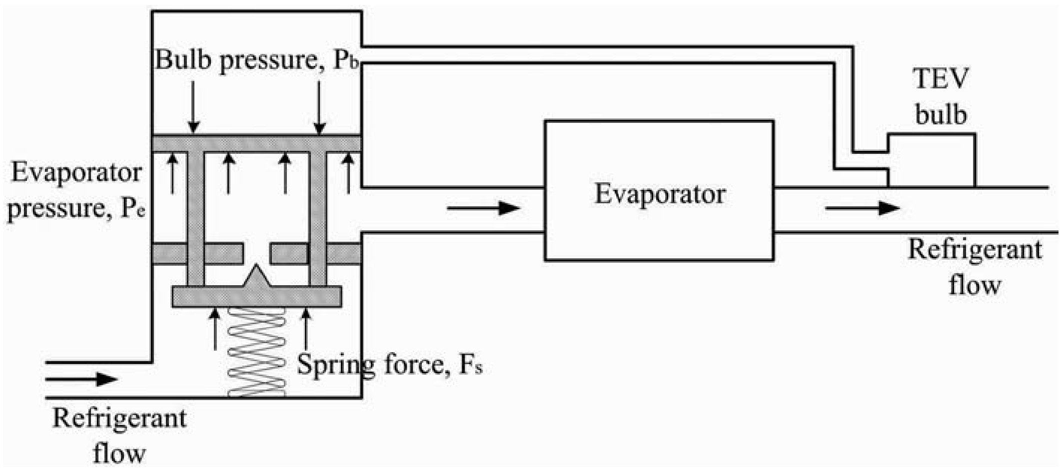

Figure 4: Diagram of Thermal Expansion Valve Operation. Reprinted from Rasmussen, Bryan Philip. Dynamic Modeling and Advanced Control of Air Conditioning and Refrigeration Systems. Urbana, Illinois: University of Illinois, 2005.

Due to the many parts involved in this flow restrictor, the TX valve is the most difficult to model. Both the valve coefficient and the area are constantly changing to accommodate a specified superheat by increasing or decreasing the mass flow rate of the refrigerant. Because this component makes the overall air conditioner operate more efficiently, there are various retrofit options available that can be installed on just about any specific model of air conditioner.

CHAPTER THREE: MOVING BOUNDARY METHOD

In an attempt to improve the ability to model system performance various methods have been established. One of the popular processes for modeling the VCC is the “moving boundary method” which is a type of lumped parameter model with a fixed number of zones that change in length. The complexity of this model was originally presented in (Wedekind and Stoeker 1966) and expanded upon by (Grald and MacArthur 1992) and many others since then. In this method the total length of the heat exchangers are divided into zones containing gas, liquid or mixed phases of the working fluid. This procedure is commonly used because it provides a computationally efficient and effective way of capturing the complexities of the heat exchangers used within the overall system.

Considering most of the air conditioner operation happens within the heat exchangers of the system, the evaporator and the condenser designs can be intricate. The refrigerant enters these heat exchangers at one thermodynamic state and exits as another. Knowing when these phase changes happen and the lengths of each division is important in understanding the overall efficiency of the system and the overall heat transfer performance is dependent on the location of these boundaries.

The moving boundary method captures salient subtleties within the entire heat exchanger while minimizing the number of differential equations required for a detailed simulation (McKinley and Alleyne 2008). Alternatively, using an approach with several volumes of fixed length throughout the heat exchanger would cause the simulation to run a factor of two to four times slower than a simulation using the moving boundary model

(Bendapudi 2004). Applying the moving boundary method provides greatly reduced computation time because the focus is on a minimum number of zones with variable lengths instead of many zones with fixed lengths.

Model Concerns

One of the concerns with using the moving boundary model is that the model may become singular and fail under certain operating conditions. For example, if a zone within the heat exchanger becomes zero in length, the governing equation set will become singular which will cause the simulation to fail. Because of this, the applicability of the initial approach of the model can often be considered as both limited and incomplete (McKinley and Alleyne 2008). Singularities most likely occur during the system start-up and shut-down as well as extreme and sudden changes to operating conditions, which does not often occur during typical operation. In order to avoid this singularity, parameter tuning must be incorporated to better constrain the model during simulation. Additional constraints must be put into place to ensure the refrigerant enters the compressor as a superheated vapor and enters the flow restrictor as a subcooled liquid while under operation. While these design constraints make the system less efficient, they are also ensuring long term usage of the system.

Method Description

The moving boundary method was created to assist in real time simulation needs because it is more computationally efficient. The faster speed makes it the method of choice for control purposes (McKinley and Alleyne 2008). This increased speed is significant, especially when controlling a system for energy efficiency. The main reason the moving boundary method is much quicker is because it lumps the refrigerant within the heat exchangers into a minimum number of divisions. There are three zones within the condenser; the superheated flow between state 2 and state 2', the two-phase zone between state 2' and 2'', and the subcooled zone between state2'' and state 3 as seen in Figure 5. For the evaporator there are two zones, the two-phase zone between state 4 and state 4' and the superheated zone between state 4' and state 1 as seen in Figure 6.

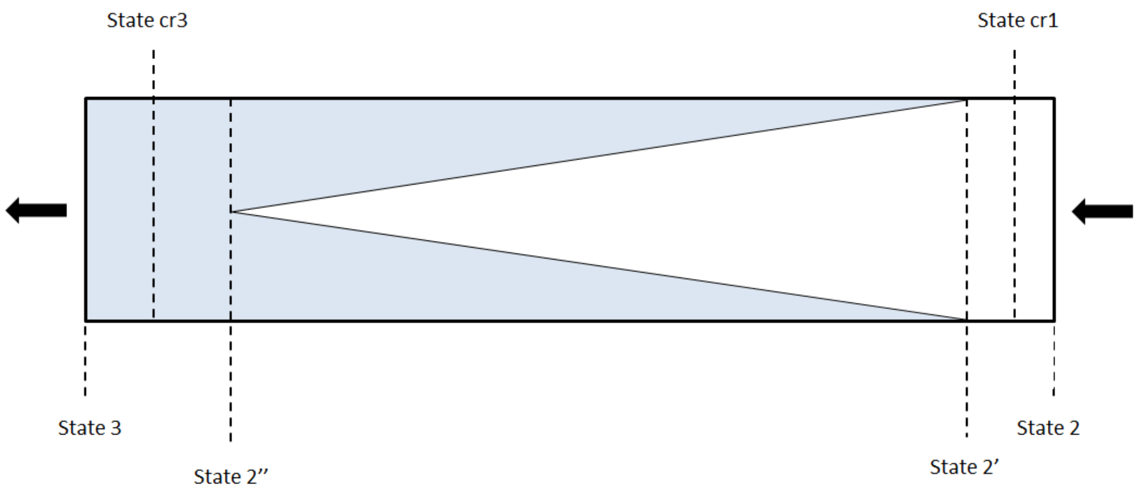

Figure 5: Lumped Parameters at the Condenser

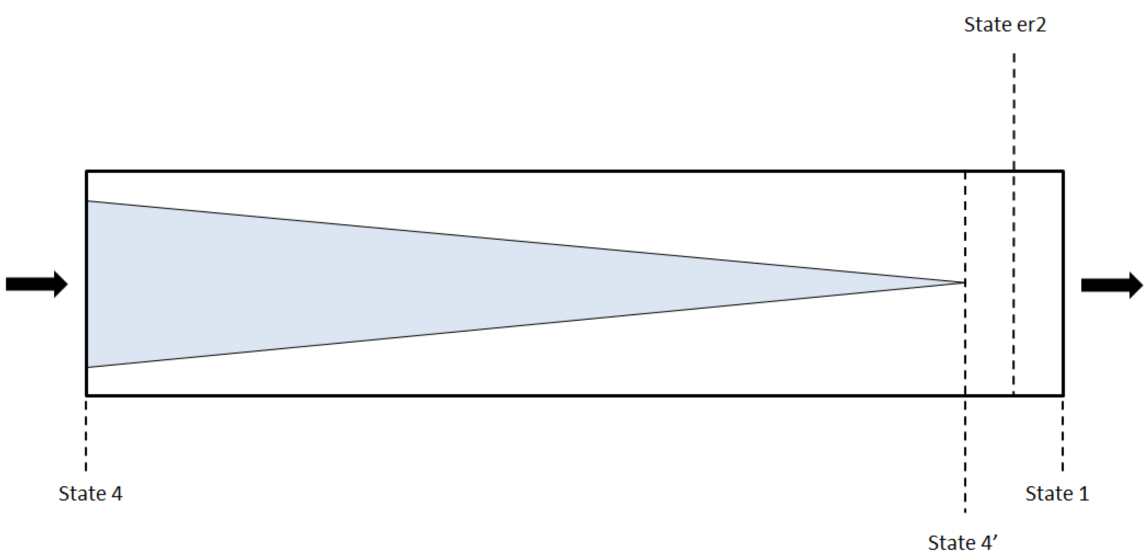

Figure 6: Lumped Parameters at the Evaporator

The lengths of these zones change in response to changes in the operating conditions. This is different than the finite model that has the heat exchangers broken up into dozens of zones with unchanging lengths (Bendapudi 2004). The use of the minimized number of zones significantly reduces the calculations required to track the refrigerant's thermodynamic state at every instant during its flow through the heat exchangers yet previous research proves it still provides an accurate overall analysis.

A key simplification used is the assumption that there is not a pressure drop across either of the heat exchangers. The assumption that the pressure drop within the heat exchangers is negligible is incorrect yet universally applied as the pressure drop is extremely minor (Qiao, Aute and Radermacher 2014). In order to understand the effect of the refrigerant properties throughout the entire heat exchanger it is important to calculate the length of each region within each different heat exchanger. The lengths of each region are crucial in determining the total heat transfer rate because both the heat transfer coefficient and the density of the refrigerant differ from zone to zone.

Heat Exchangers

One of the unique attributes of the moving boundary method is that the lengths of each zone are time dependent and integration must be done to track these time varying quantities. Strides have been made in developing lumped parameter models for VCC's including the groundbreaking work in (X.-D. He 1996) and (McKinley and Alleyne 2008).

The dynamic model represented in (X.-D. He 1996) focuses the attention to a cross-flow type heat exchanger with R-22 refrigerant filled tubing and air as the secondary fluid. He uses various partial differential equations and heat exchanger dynamics to generate the equations required for his research. The heat transferred from the refrigerant to the tube wall as well as the tube wall to the atmosphere is both considered. The matrix of equations presented in (X.-D. He 1996) for both the evaporator and the condenser encompasses both energy balance equations as well as mass balance equations due to the nature of the partial derivatives. Since then, research has been done to address concerns with outdated research regarding heat exchanger design, nonlinear air temperature distribution as well as non-circular refrigerant passages (McKinley and Alleyne 2008). The work supports the moving boundary method over a finite volume model but notes the probability of the model becoming singular and failing under atypical operation. This operation includes the possibility that the number of zones within the heat exchangers can be variable and not fixed to three and two for the condenser and evaporator respectively. Later, the research was taken a step further to understand the basis of operation when the VCC undergoes start-up and shut-down procedures (Li and Alleyne 2010). All of the progress, however, falls back on the foundation that was built in (X.-D. He 1996) including his matrix of equations for both the evaporator and the condenser which will be presented in the following sections.

<u>Evaporator</u>

This paper gets its starting point from the work done in (X.-D. He 1996). He begins with a matrix of partial derivatives and the evaporator dynamic model can be seen in Equation 1.

Equation 1: Dynamic Model for Evaporator

$$\mathbf{D}_{\mathrm{E}}\dot{\mathbf{x}}_{\mathrm{E}} = \mathbf{f}_{\mathrm{E}}(\mathbf{x}_{\mathrm{E}}, \mathbf{u}_{\mathrm{E}})$$

This model is based off of a group of functions as seen in Equation 2 where $\dot{\mathbf{x}}_{\mathbf{E}}$ is the vector of state variables given by Equation 3 and $\mathbf{u}_{\mathrm{E}}$ are input variables shown in Equation 4. Expressions of all the elements in the $\mathbf{D}_{\mathbf{E}}$ matrix can be seen in (X.-D. He 1996).

Equation 2: Dynamic Model Functions for Evaporator

$$\mathbf{f}_{\mathrm{E}} = \begin{bmatrix} \dot{m}_i h_i - \dot{m}_i h_g + \alpha_{i1}\pi D_i L_1 (T_{w1} - T_{r1}) \\ \dot{m}_o h_g - \dot{m}_o h_o + \alpha_{i2}\pi D_i L_2 (T_{w2} - T_{r2}) \\ \dot{m}_i - \dot{m}_o \\ \alpha_{i1}\pi D_i (T_{r1} - T_{w1}) + \alpha_o \pi D_o (T_a - T_{w1}) \\ \alpha_{i2}\pi D_i (T_{r2} - T_{w2}) + \alpha_o \pi D_o (T_a - T_{w2}) \end{bmatrix}$$

These functions reference inlet and outlet flow rates of the evaporator along with inlet and outlet enthalpies of the various zones. In addition, the heat transfer coefficients of the tubing are needed along with the inner diameter and lengths of the different zones. Lastly, the temperatures of the tube wall, the refrigerant and the conditioned space are included. This group of equations is key and the focus of further analysis later in the thesis.

Equation 3: Dynamic Model State Variables for Evaporator

$$\mathbf{x}_{\mathrm{E}} = [L_{e1}\ P_E\ h_{eo}\ T_{ew1}\ T_{ew2}]^T$$

The state variables of the dynamic model include the length of the two-phase flow zone, the pressure in the evaporator, the enthalpy at the exit and the average wall temperatures within the two zones.

Equation 4: Dynamic Model Input Variables for Evaporator

$$\mathbf{u}_{\mathrm{E}} = [\dot{m}_i\ h_i\ \dot{m}_o\ ]^T$$

Equation 4 shows that the dynamic model of the evaporator takes, as input, the entering and exiting mass flow rate and the entering enthalpy of the heat exchanger. These values are determined by models of the other components of the system.

Condenser

Much like the evaporator a basis of study for the condenser operation begins with the work done in (X.-D. He 1996). The matrix of partial derivatives for the condenser dynamic model can be seen in Equation 5.

Equation 5: Dynamic Model for Condenser

$$\mathbf{D}_{\mathrm{C}}\dot{\mathbf{x}}_{\mathrm{C}} = \mathbf{f}_{\mathrm{C}}(\mathbf{x}_{\mathrm{C}}, \mathbf{u}_{\mathrm{C}})$$

This model is based off of a group of functions as seen in Equation 6 where $\mathbf{x}_{\mathrm{C}}$ is the vector of state variables given by Equation 7 and $\mathbf{u}_{\mathrm{C}}$ are control variables shown in Equation 8. Expressions of all the elements in the $\mathbf{D}_{\mathrm{C}}$ matrix can be seen in (X.-D. He 1996).

Equation 6: Dynamic Model Functions for Condenser

$$\mathbf{f}_{\mathrm{C}} = \begin{bmatrix} \dot{m}_i h_i - \dot{m}_i h_g + \alpha_{i1}\pi D_i L_1 (T_{w1} - T_{r1}) \\ \dot{m}_o h_g - \dot{m}_o h_l + \alpha_{i2}\pi D_i L_2 (T_{w2} - T_{r2}) \\ \dot{m}_o h_l - \dot{m}_o h_o + \alpha_{i3}\pi D_i L_3 (T_{w3} - T_{r3}) \\ \dot{m}_i - \dot{m}_o \\ \alpha_{i1}\pi D_i (T_{r1} - T_{w1}) + \alpha_o \pi D_o (T_a - T_{w1}) \\ \alpha_{i2}\pi D_i (T_{r2} - T_{w2}) + \alpha_o \pi D_o (T_a - T_{w2}) \\ \alpha_{i3}\pi D_i (T_{r3} - T_{w3}) + \alpha_o \pi D_o (T_a - T_{w3}) \end{bmatrix}$$

Like the evaporator equations, this equation uses inlet and outlet flow rates over the entire condenser along with inlet and outlet enthalpies of the various zones. The heat transfer coefficients and the inner diameter of the tubing are used along with the temperatures of the tube wall, the refrigerant and the ambient outside air are included.

Equation 7: Dynamic Model State Variables for Condenser

$$\mathbf{x}_C = [L_{c1}\ L_{c2}\ P_C\ h_{co}\ T_{cw1}\ T_{cw2}\ T_{cw3}]^T$$

The state variables of the dynamic model include the length of the superheat zone and the two-phase flow zone, the pressure at the condenser, the enthalpy at the exit and the average wall temperatures at each of the three zones.

Equation 8: Dynamic Model Input Variables for Condenser

$$\mathbf{u}_C = [\dot{m}_i\ h_i\ \dot{m}_o\ ]^T$$

Equation 8 shows that the input for the condenser dynamic model requires the entering and exiting mass flow rate and the entering enthalpy of the heat exchanger. Once again, these values are determined by models of the other components of the system.

Compressor

The compressor design has substantially evolved making it the single most complex component in the VCC. When looking at this feature as a steady operating component, and assuming that the compressor is well insulated, the relationships between compression and flow rate can be determined utilizing the following equation (X.-D. He 1996):

Equation 9: Flow rate Through Compressor

$$\dot{m} = \omega \Upsilon_k \frac{1}{v_1}\left[1 + C_k - C_k\left(\frac{P_C}{P_E}\right)^{\frac{1}{2}}\right]$$

Equation 9 takes into account the rotating shaft speed of the compressor, ω, as well as a compressor coefficient, C_k, and the effective volume displacement Υ_k.

While all other conditions are thermodynamically determined for the compressor analysis, it is important to note that the compression process is an isentropic process and isentropic efficiencies must be taken into account in order to accurately model a system. The relationship between the enthalpies with and without the consideration of isentropic efficiency can be seen in Equation 10 and Equation 11.

Equation 10: Enthalpy without Isentropic Efficiency

$$h_{2s} = f(P_2, s_2)$$

Equation 11: Enthalpy with Isentropic Efficiency

$$h_2 = h_1 + \frac{h_{2s} - h_1}{\eta_s}$$

Flow Restrictor

The operation of the flow restrictor, and its relationship to the changing flow rate, can be determined using Equation 12 (X.-D. He 1996). This orifice equation takes into account the valve coefficient, C_v, and the area of the valve opening, A_v; all other values are determined using thermodynamic properties.

Equation 12: Flow rate through Flow Restrictor

$$\dot{m} = C_v A_v \sqrt{\frac{1}{\upsilon_3} * (P_C - P_E)}$$

The simple algebraic relationship shown above can be used for all different types of flow restrictors as discussed earlier in the thesis. The only difference in the application of the equation is which values are considered variable to the system. For example, with an EE valve, the area of the valve is a continually adjustable variable where with the fixed orifice it is a constant value.

Interaction of the Component Models

Figure 7 reflects how the information flows between the component models to generate the overall analysis of the system. The blue arrows reflect how the pressures are used within each model, the green arrows reflect how the flow rate is used within each model and the red arrows reflect how the enthalpy is used within each model.

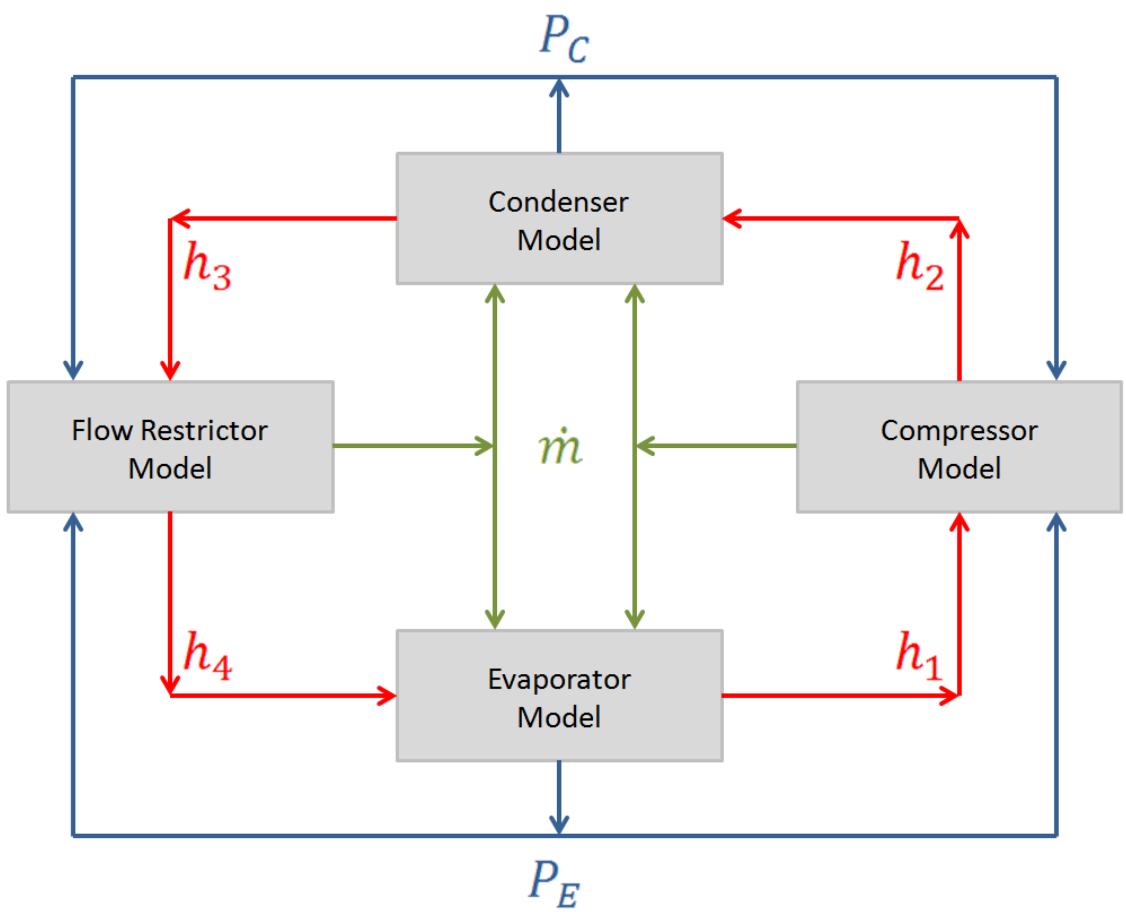

Figure 7: Information Flow between Systems

CHAPTER FOUR: STEADY-STATE COMPONENT MODELING

Heat Exchangers

Typically, the mechanical components of a VCC, the compressor and flow restrictor, are the main focus of research and often times the heat exchangers within the system are overlooked or modeled in an overly simplistic manner. In reality the evaporator and the condenser are vital components to the overall performance. Optimizing the design and operation of these pieces will greatly impact the functionality of the whole unit. Reviewing the analysis used in (X.-D. He 1996) for a lumped parameter model along with other past research, it is common to assume an older design of air conditioner that utilizes smooth circular refrigerant tubes with no fins through the heat exchangers was referenced. Heat exchanger design has significantly developed and is always continuing to make technological advances. One of the major factors in improving heat transfer capabilities and reducing material costs is to add fins to the tubing within the heat exchangers.

While some of the more recent research has utilized a fin design for a heat exchanger, much of the research done does not focus on the heat transfer from the coil/fin assembly to the ambient air. In order to optimize efficiency, all of the applicable heat transfer opportunities need to be evaluated and considered. There is the convective heat exchanged from the refrigerant to the tube wall, conducted heat transferred through the tube wall and convective heat transferred from the tube wall to the fins/ambient air and from the fins to the ambient air (Xue, et al. 2011). While some research has been done to incorporate these heat transfer capabilities, there is still a problem with making the model universally adaptable. The problem with previous research is that the information available is only applicable to a single model and design of air conditioner.

The first step in overcoming this barrier is to create a model with parameters that are simple enough to get from existing data, yet complex enough to model a wide range of systems. At this point, only steady state models are used because manufacturers provided test data was developed under steady state operation and most equipment use is under steady state operation, or very near so. In order to do this, one must take the dynamic model from the literature, as described earlier, and transition to a steady state model by setting the state derivatives to zero. This transition forced the state derivatives to go to zero turning Equation 1 and Equation 5 into Equation 13 and Equation 14 respectively.

Equation 13: Steady State Model for Evaporator

$$0 = f_E(x_E, u_E)$$

Equation 14: Steady State Model for Condenser

$$0 = f_C(x_C, u_C)$$

Changing to a steady state model allows for parameter determination using manufacturer provided test data, which was also evaluated under steady-state conditions, allowing the model to be utilized for a variety of VCC units. At this point each of the state derivatives with respect to time, in Equation 2 and Equation 6 are set equal to zero. The thermal mass of the tube walls, an essential part of the dynamic model, is not important for steady-state analyses. Therefore, the equations are manipulated further and the overall effect is combined into an effective heat transfer value per unit length. This value encompasses all heat transfer capabilities and has been adapted using a steady state assumption (X.-D. He 1996). Each of these equations utilizes effective heat transfer per unit length values adapted from He's work as well. As an example, using the nomenclature as previously stated, the effective heat transfer per unit length for the superheated phase in the condenser can be seen in Equation 15.

Equation 15: Effective Heat Transfer for Superheat Zone - Condenser

$$U_{c1} = \frac{(\alpha_{ci1}\pi D_{ci} * \alpha_{co}\pi D_o)}{(\alpha_{ci1}\pi D_{ci} + \alpha_{co}\pi D_o)}$$

This equation is derived using the steady state versions of the heat transfer equations from the refrigerant to the tube wall and from the tube wall to the outside air. T_{cw1} must be solved for in Equation 16 and then the result substituted into Equation 17 as found in (X.-D. He 1996).

Equation 16: Steady Heat Transfer between Refrigerant and Tube Wall

$$\dot{m}(h_2 - h_{2'}) + \ \alpha_{ci1}\pi D_{ci} l_{c1}(T_{cw1} - T_{cr1}) = 0$$

Equation 17: Steady Heat Transfer between Tube Wall and Air

$$\alpha_{i1}\pi D_i(T_{r1} - T_{w1}) + \alpha_o \pi D_o(T_a - T_{w1}) = 0$$

Figure 8 shows a pictorial view of the refrigerant tube wall and the associated flow rate and temperatures as seen in previous equations.

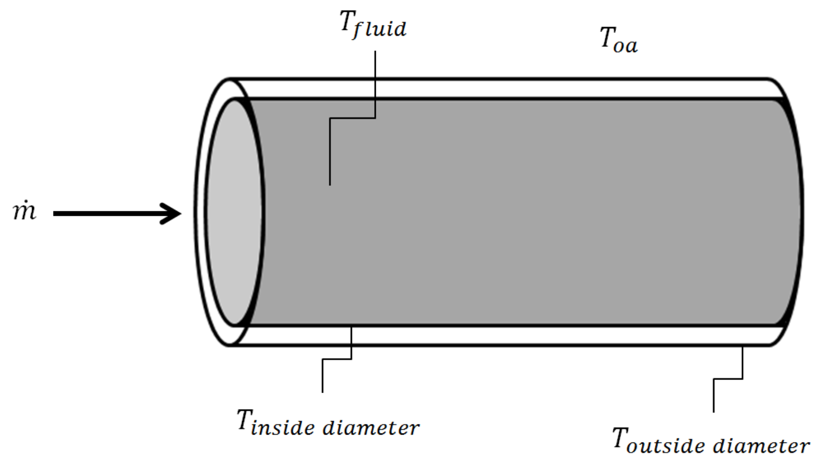

Figure 8: Temperatures Surrounding Heat Exchanger Performance

Combining all the various heat transfer capabilities into an effective value allows the user to look at the "big picture" and see how the machinery is operating within the various zones. For Equation 15 specifically, once the equations were modified from the original work the $\alpha_{co}\pi D_o$ values are considered a lumped value for all heat transfer past the tube wall which could include fin incorporation to the design. By lumping all heat transfer capabilities into one parameter, U_{c1} in this example, the complexity of the various components and design specifications can be considered one single effective component instead of separated thermal resistances within each region. Figure 9 shows this simplification.

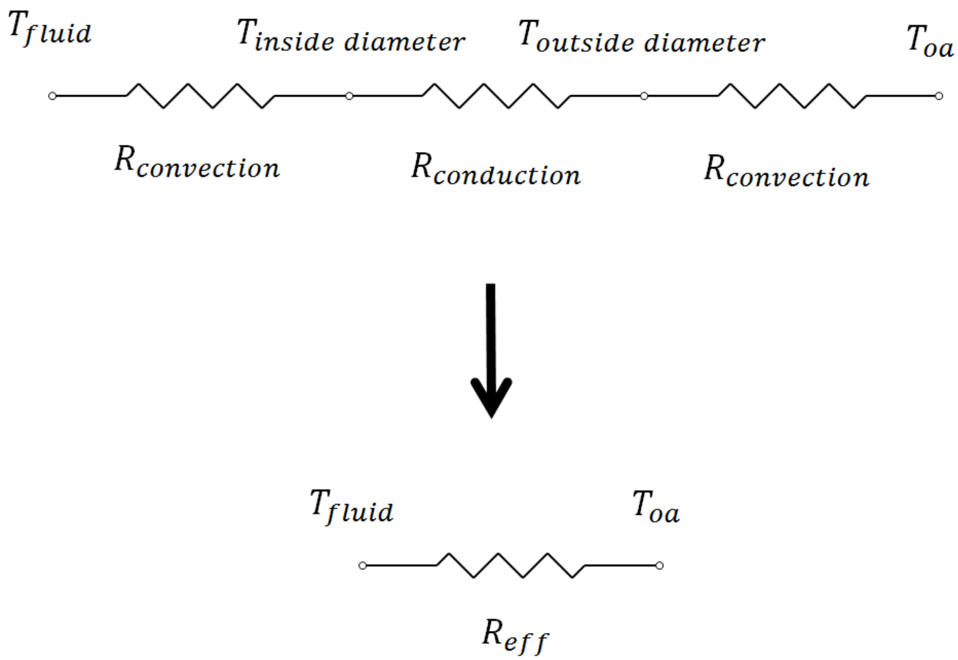

Figure 9: Simplification for Heat Transfer Components

The effective heat transfer value is a more useful parameter because by making this value a general constant for each refrigerant phase, the model can be utilized for a variety of air conditioners and heat exchanger designs allowing this model to be replicated time and time again with ease. This approach allows the user the ability to find a parameter that matches the model's performance to the actual test data. Because of fin geometry and the complexity of the heat transfer in the fins previous models could not grasp this value as simply.

After adapting each equation shown in the matrices presented earlier, the functions to be used in this model are shown in Equation 18 - Equation 22, one equation is given for each refrigerant phase within the heat exchangers. Maintaining an adapted form of He's work dictates an energy balance has been incorporated throughout the entire system and has not been violated (X.-D. He 1996).

Evaporator

Equation 18: Two-Phase Region - Evaporator

$$\dot{m}(h_{4'} - h_4) = U_{e1} * l_{e1} * (T_{ea} - T_4)$$

Equation 19: Superheat Region - Evaporator

$$\dot{m}(h_1 - h_{4'}) = U_{e2} * l_{e2} * (T_{ea} - T_{er2})$$

Condenser

Equation 20: Superheat Region - Condenser

$$\dot{m}(h_2 - h_{2'}) = U_{c1} * l_{c1} * (T_{cr1} - T_{oa})$$

Equation 21: Two-Phase Region - Condenser

$$\dot{m}(h_{2'} - h_{2''}) = U_{c2} * l_{c2} * (T_{2'} - T_{oa})$$

Equation 22: Subcool Region - Condenser

$$\dot{m}(h_{2''} - h_3) = U_{c3} * l_{c3} * (T_{cr3} - T_{oa})$$

Utilizing a constant effective heat transfer coefficient for each region that incorporates all aspects of allowable heat transfer provides a unique approach to identifying censorious parameters required for an accurate VCC model. This framework along with limited sensor information can provide a dynamic model to assist in design, control and operation of traditional VCC systems.

Mass Balance

In order for the steady state model to correctly capture performance, the mass of the refrigerant in each component must be tracked. In a dynamic model, this was done with unique dynamic states for the mass in the evaporator and the condenser, but those relationships became trivial in the steady state case. Instead, the steady state model will enforce the constraint that the total mass of refrigerant is unchanged under various operating conditions. The mass of a VCC needs to incorporate all applicable components within the system where refrigerant can be located and contribute to the overall refrigerant mass. This view has been adopted for the model as laid out in this paper, but considering the refrigerant goes through phase changes at different operating conditions the mass in these components will be dependent on time and operating conditions.

VCC Refrigeration Mass

Refrigerant mass distribution is dependent on the specific air conditioning unit. While there are four main elements to every VCC, additional mechanisms can be added or modified to increase the efficiency or production of the unit. These additional components often times include some type of refrigerant mass that needs to be accounted for in the total mass migration of the system. The applicable components for a complex VCC containing refrigerant mass are shown in Table 2.

Table 2: VCC Components Containing Refrigerant Mass

Evaporator	The mass within the evaporator is dependent on the tube's inner diameter, overall tube length and thermodynamic state of the refrigerant
Accumulator	The accumulator is attached to the evaporator outlet to ensure that only vapor is entering the compressor. If the system is running properly and going into superheat all the refrigerant entering the accumulator should be superheated vapor but there could be a small fraction of liquid refrigerant as well that would need to be calculated for in the mass
Compressor	The mass of the refrigerant at the compressor is minimal but will still be dependent on the size and specification of the compressor and may need to be considered
Condenser	The mass within the condenser is dependent on the tube's inner diameter, overall tube length and thermodynamic state of the refrigerant

Liquid Tube	The liquid tubing is the refrigerant between the condenser outlet and the flow restrictor inlet
Additional Piping	Often times in split systems the condenser is located outside and the evaporator supplying the building is located inside. In this case, there are refrigerant lines that run from the outside to the inside and vice versa. The following lengths and tube's inner diameter need to be considered when accounting for refrigerant mass throughout the migration process: ○ Compressor to condenser ○ Flow restrictor to the evaporator ○ Evaporator to the accumulator ○ Accumulator to compressor
Flow Restrictor	The mass of the refrigerant within the flow restrictor will be dependent on the type of flow restrictor used
Miscellaneous	Any additional components added to the system

When each of these pieces of equipment is taken into account the mass balance becomes what is seen in Equation 23.

Equation 23: Refrigerant Mass of Split System

$$M_{total} = M_{Evaporator} + M_{Accumulator} + M_{Piping} + M_{Compressor} + M_{Condneser} + M_{Liquid\ Line} + M_{Flow\ Restrictor}$$

In some cases, an air conditioner is designed as a packaged unit that includes all the equipment of the VCC into one cabinet assembly. A packaged unit reflects the model that was used for the analysis within this paper. While there are still the four main parts to the machine, majority of the refrigerant is within the evaporator and condenser. The liquid line and the additional piping are very small and will be neglected as it is all packaged within the same structure and the mass at these locations remains constant. Additionally, with common residential units, since the compressor is not very large, the refrigerant mass within is negligible. When a fixed orifice is used no mass is being held within the valve so that mass can be removed from the calculation as well. Understanding this, the mass balance equation is easily manipulated and simplified to meet the needs of this analysis; Equation 23 simply becomes Equation 24.

Equation 24: Refrigerant Mass of Packaged Unit

$$M_{total} = M_{Evaporator} + M_{Condneser}$$

<u>Mean Void Fraction</u>

In determining the mass migration through a refrigeration system it is common to assume that the enthalpy has a linear profile along the regions making the mass inside readily evaluated. This is why the information within the single-phase regions, superheated and subcooled, are calculated using the arithmetic average between the two states and the associated length of the region. However, when looking at the two-phase flow within the heat exchangers a more sophisticated approach is required. A mean void fraction model can be applied to calculate the mass within the two-phase flow portion of the heat exchangers (Beck and Wedekind 1981).

A mean void fraction is used to help determine the mass of the refrigerant within the two-phase mixture region of the heat exchangers. This component allows one to predict the amount of vapor refrigerant within the evaporator and the condenser throughout the two-phase flow. In turn, this will help determine the total mass of the refrigerant so as to satisfy the mass balance laws. The mean void fraction is imperative in the use of the lumped parameter method to forecast the transient responses within these heat exchangers.

As with the length of the moving boundaries, the mean void fraction will also vary depending on time and various conditions that will affect the system. Reviewing the previous work done on the mean void fraction and integrating it into this system Equation 25 - Equation 28 have been determined to reflect the mean void fraction relationships in both the evaporator and the condenser and their contribution to the total mass calculation. These equations were formed applying the Zivi void fraction correlation (G.L. Wedekind 1976). For a full derivation of this equation please see Appendix B.

Evaporator

Equation 25: Mean Void Fraction at Evaporator

$$\bar{\gamma}_E = \frac{1}{\left(1-\left(\frac{v_4}{v_{4'}}\right)^{\frac{2}{3}}\right)} + \frac{\left(\frac{v_4}{v_{4'}}\right)^{\frac{2}{3}}}{(1-x_4)\left(1-\left(\frac{v_4}{v_{4'}}\right)^{\frac{2}{3}}\right)^2} * ln\left[\left(\frac{v_4}{v_{4'}}\right)^{\frac{2}{3}} + \left(1-\left(\frac{v_4}{v_{4'}}\right)^{\frac{2}{3}}\right) * x_4\right]$$

Equation 26: Refrigerant Mass at Evaporator

$$M_E = \frac{\pi D_{ei}^2}{4}\left[l_{e1}\left(\frac{\bar{\gamma}_E}{v_{4g}} + \frac{1-\bar{\gamma}_E}{v_{4f}}\right) + \frac{l_{e2}}{v_{er2}}\right]$$

Condenser

Equation 27: Mean Void Fraction at Condenser

$$\bar{\gamma}_C = \frac{1}{\left(1-\left(\frac{v_{2''}}{v_{2'}}\right)^{\frac{2}{3}}\right)} + \frac{\left(\frac{v_{2''}}{v_{2'}}\right)^{\frac{2}{3}}}{\left(1-\left(\frac{v_{2''}}{v_{2'}}\right)^{\frac{2}{3}}\right)^2} * ln\left[\left(\frac{v_{2''}}{v_{2'}}\right)^{\frac{2}{3}}\right]$$

Equation 28: Refrigerant Mass at Condenser

$$M_C = \frac{\pi D_{ci}^2}{4}\left[\frac{l_{c1}}{v_{cr1}} + l_{c2}\left(\frac{\bar{\gamma}_C}{v_{2\prime}} + \frac{1-\bar{\gamma}_C}{v_{2\prime\prime}}\right) + \frac{l_{c3}}{v_{cr3}}\right]$$

Now that the mean void fraction is determined in values that can be inferred from the initial input they can be used to determine the overall mass of the system that is within the evaporator and the condenser.

It is important to note that although the mean void fraction calculated is helpful in acquiring an accurate model, this value is constantly changing. An important simplification to the mean void fraction study is that the time dependence is neglected. Not only is this value changing in different operating conditions it is also changing throughout the two-phase section of both the evaporator and the condenser. Using a single average value that does not incorporate the time-variance, however, does not cause major impact on the overall system (Beck and Wedekind 1981).

As mentioned earlier, there can be additional components added to the VCC to make the unit more efficient and avoid failure. One of these components is the accumulator, and its purpose is to catch any lingering liquid refrigerant as the flow leaves the evaporator. The need for this component suggests that there is the off chance that the refrigerant undergoes incomplete vaporization. If this component is added to the system a new method of the mean void fraction must be used. A method introduced by Beck presents a generalization mean void fraction method which capitalizes on this concept. This research, however, is also simplified to assume the mean void fraction is not time varying (Beck and Wedekind 1981).

CHAPTER FIVE: PARAMETER DETERMINATION

The specific model used to demonstrate this approach was the Goodman PC1436H41 as seen in Figure 10. The model was a 36,000 BTU/hr (3 ton) residential unit that, due to its size, would be applicable to many homeowners. The manufacturers website had most of the required information for the model. If any additional information or clarification was needed a manufacturer representative was contacted to obtain this information or clarity. Once all of the testing data and information on the system was acquired the analysis could proceed. The parameters below are required to be known from the manufacturer's documentation to run the analysis on the test conditions and utilize the overall model. Where a test variable corresponds to a variable in the model, the variable name is listed:

- Suction Pressure, P_E
- Discharge Pressure, P_C
- Indoor Set Point Temperature, T_{ea}
- Outdoor Ambient Temperature, T_{oa}
- Refrigerant Type
- Total Compressor Work, $\dot{W}_{in}$
- Degrees of Superheat
- Degrees of Subcool
- Compressor Speed, ω
- Flow Restrictor Area (only if fixed orifice), A_v
- Charge of System, M_{total}
- Heat Absorption Rate (Refrigerant Load), $\dot{Q}_L$

Once the above information was acquired an operating condition was selected from the test data. A thermodynamic analysis was done to get all the states as listed in Table 1 earlier in the thesis. It is at this point that the refrigerant load was used to back out the mass flow rate for that test condition. The remaining data was used to determine the remaining parameter values and minimize the error within the model.

The documentation received from the manufacturer had all the required information for four different tests using various outdoor ambient temperatures at the same set point temperature. There were three different set point temperatures for these ambient conditions allowing for twelve uniquely tested data points to be used in various examinations to solve for the required parameters. The twelve different test conditions were built in various worksheets within a single Excel document and all the required data was determined for each one. A view of these simulated spreadsheets can be seen in Appendix D.

Figure 10: Goodman GPC1436H41 Air Conditioner. Reprinted from http://www.goodmanmfg.com/ResidentialProducts/AirConditioners.aspx

Spreadsheet Methodology

Thermodynamic Add-In

In order to utilize the spreadsheet based analysis an add-in was required to be downloaded for Microsoft Excel. For this analysis a free download offered by University of Alabama was used (Excel in Mechanical Engineering n.d.). This add-in includes psychometric functions and thermodynamic properties for the following refrigerants: R407C, R410A, R22 and R134a. The VCC model used for this research requires R-410a refrigerant, which is a blended refrigerant common for residential air conditioners and supported by the Excel add-in. This downloaded feature along with Excel’s provided *Solver* add-in is all that is needed to replicate the model.

Solver

To generate and operate this model, Excel’s *Solver* function was used. First it was used in collaboration with manufacturer provided test data to determine the parameters required satisfying the needs of the full model. These parameters were extrapolated in a series of three different spreadsheets, one to solve for the parameters for the compressor, one to solve for the parameters for the flow restrictor, and one to solve for the parameters for the evaporator and condenser while maintaining an appropriate mass balance. Once these parameters were deduced, they were input into the full VCC model. This model is then used to predict system performance of any given air conditioning unit based off of minimal user inputs. It does this by minimizing the overall error within the system by comparing the results found thermodynamically and the results found using the equations modified from (X.-D. He 1996).

In general, the main purpose of *Solver* is to find a solution that minimizes or maximizes an objective cell value while satisfying a number of constraints that could be placed on the system. The kind of solution one can expect and computation time depends on three characteristics of the model (Frontline Solvers 2016):

1. The size of the model
    a. Including number of variables, constraints and formulas
2. Complexity of mathematical relationships between objective cell and constraints
    a. Nonlinear vs. linear
3. Use of integer versus variables within the model

Using Excel for this model pushes the limits of its capabilities; yet, the model is still accurate when compared against the results of other models in past research using a more sophisticated modeling software program. Once the constraints and the objective cell were determined *Solver* was ready to be run. To speed up the process the GRG Nonlinear setting in *Solver* was used. This is a generalized reduced gradient (GRG) algorithm used for optimizing a range of nonlinear problems. It employs an iterative numerical method that involves adjusting trial values for the adjustable cells and reviewing the results of the objective cell. When multiple values are entered, as with this analysis, partial derivatives and gradients assist in measuring the rate of change (Microsoft Support 2016).

A shortcoming of the GRG Nonlinear setting was that the resulting values provided a local solution and not a global one. Two things were done to overcome this fault. An appropriate initial condition or “guess value” had to be given to *Solver* as a starting point in the computation. In order to have the analysis run properly initial conditions had to be placed as initial "stand in" values for what was to be determined. The initial assumptions were determined by using relationships seen within the test conditions and can be seen in Appendix C. The second thing to ensure the accuracy of Excel's Solver was that in some cases the simulation needed to be ran more than once. At most the simulation needed to be run three times, each time providing *Solver* with more accurate initial conditions.

Assumptions

In both determining the required parameters as well as running the full analysis an array of assumptions were considered. A comprehensive list of these assumptions is shown below:

- No pressure drop across heat exchangers
- No temperature drop between State 4 and State 4'
- No temperature drop between State 2' and State 2''
- Consistent ambient air temperature across the condenser
- Constant mean void fraction calculations
- 85% Isentropic efficiency
- No refrigerant mass in the compressor, flow restrictor, and additional tubing
- Information from manufacturer was detailed and accurate
- The test conditions were measured during steady state operation

Solving for Parameters

One of the barriers in applying VCC performance models is the identification of parameter values required to make these models useful. In order to have a model that accurately depicts how the air conditioner is performing various parameters need to be solved for and utilized.

There are a total of eight nonlinear equations which can be used with Excel's *Solver* to identify the parameter values which minimize the errors in previous equations (Equation 9, Equation 12, Equation 18, Equation 19, Equation 20, Equation 21, Equation 22, and Equation 24). Using these equations there are, in total, ten model parameters to be identified. These parameters are listed below:

- Heat Transfer Coefficient for each Region ($U_{e1},\ U_{e2},\ U_{c1},\ U_{c2}, and\ U_{c3}$)
- Total Length of Tubing within Evaporator (L_E)
- Total Length of Tubing within Condenser (L_C)
- Compressor Displacement Volume (Υ_k)
- Compressor Coefficient (C_k)
- Valve Coefficient (C_v)

Accompanying those parameters are five region lengths that will change for each operating condition ($l_{e1},\ l_{e2},\ l_{c1},\ l_{c2}, and\ l_{c3}$). A minimum of four tests are required to determine the parameters because when four tests are used the number of equations outnumbers the number of unknowns. Table 3 shows the breakdown on required test conditions compared to the number of unknowns and the number of equations. To improve the robustness of the process, a total of 12 test conditions were used.

Table 3: Minimum Required Tests to Determine Parameters

1 Test Condition	15 Unknowns	8 Equations
2 Test Conditions	20 Unknowns	16 Equations
3 Test Conditions	25 Unknowns	24 Equations
4 Test Conditions	30 Unknowns	32 Equations
5 Test Conditions	35 Unknowns	40 Equations
…	…	…
12 Test Conditions	70 Unknowns	96 Equations

Mass Balance

Since the mass migration of the system are directly related to the region lengths in both the evaporator and the condenser, it is important to solve for these lengths and the mass balance simultaneously. One analysis must be done to ensure that the total length of the tubing and the specific zone lengths mesh appropriately with the mass migration throughout the system. When the analysis is run simultaneously the total tubing length of the evaporator and the condenser can be found. Along with this are the specific lengths of each region in the heat exchangers and their associated heat transfer values per unit length.

Overall Lengths and Mass

In order to solve for the mass at the evaporator and the condenser the mean void fractions from the test conditions was needed as well as the total lengths of the evaporator and the condenser tubing and the specific lengths of each zone within the heat exchangers. The mean void fraction was previously solved for and the other information required came from the manufacturer’s provided data. However, the lengths of each zone and the overall lengths within the heat exchangers were still unknown.

To determine the unknowns, *Solver* was used to run an analysis with the twelve test conditions to find the best solution for the various zone lengths and total tubing lengths to satisfy the mass balance equation under specifically determined constraints.

Evaporator

For the evaporator analysis the effective heat transfer per unit length needs to be determined for both the two-phase region and the superheat region. Along with the refrigerant load, as reported in the test conditions, a thermodynamic analysis was done at each test condition to determine the appropriate properties required to successfully compute the region energy balance equations, Equation 18 and Equation 19. An analysis was done comparing the results of the calculated heat absorption rates in both regions using the effective heat transfers against the test condition results. Excel *Solver* was directed to change the value of one common U_{e1} and U_{e2} and different l_{e1} values for each of the twelve data points. The length of the superheat zone was automatically solved for using the relationship as seen in Equation 29 which is easily done considering the total length of the evaporator has been predetermined when confirming an appropriate mass balance.

Equation 29: Region Length Relationships within Evaporator

$$l_{e2} = L_E - l_{e1}$$

The final result provided an analysis with varying zone lengths, as predicted, but a common U_{e1} and U_{e2} that can be used as parameters in the more evolved model. Whereas these parameters are considered constant in the final analysis the overall heat transfer is also affected by the set point temperatures and the lengths of each region which move to accommodate the required heat absorption. The effective heat transfer values are, essentially, an average value across each region it is safe to assume that the heat transfer characteristics will remain largely constant across each region.

There are three different error calculations within the evaporator model; the error associated with heat absorption in the two-phase region, the superheated region and the overall heat absorption in the evaporator. To calculate error throughout this model Equation 30 was employed. Each test condition had two parameters to review, the parameter directly measured from the test conditions and thermodynamic analysis and the parameter as computed using previous equations. In the case of the evaporator, both U_{e1} and U_{e2} were solved for simultaneously in order to reduce the overall error associated with the evaporator calculations. When comparing the calculated heat absorbed in the superheated region using the determined U_{e2} and Equation 19 there was a high percentage error against what the test conditions and thermodynamic analysis measured. This error was most notable when paralleled against the percent error within the two-phase region; on average the superheated region had a 20-30% higher error. However, at most the superheated region only contributed 10% of the overall heat transfer required within the evaporator. Since this accounted for such a small portion of the overall heat transfer, the overall error of heat absorption was very minimal. When comparing the values predicted using the effective heat transfer against the values measured during testing, there was a resulting error ranging from 0.06% error to 3.35% error.

Equation 30: Percent Error

$$\%\,Error = \frac{Excel\ Computed\ Value - Measured\ Test\ Condition}{Measured\ Test\ Condition}$$

<u>Condenser</u>

For the condenser analysis the effective heat transfer per unit length needs to be determined for the three different regions; superheated region, two-phase region and subcooled region. Along with the heat rejection required, as reported in the test conditions, a thermodynamic analysis was done at each test condition to determine the appropriate properties required to successfully compute the region energy balance equations, Equation 20, Equation 21, and Equation 22. An analysis was done comparing the results of the calculated heat rejection rates in all regions using the effective heat transfers against the test condition results. Excel *Solver* was directed to change the value of one common U_{c1} , U_{c2} and U_{c3} while allowing l_{c1} and l_{c2} values to be different for each of the twelve data points. The length of the subcool region was automatically solved for using the relationship as seen in Equation 31 which is easily done considering the total length has been predetermined.

Equation 31: Boundary Length Relationships within Condenser

$$l_{c3} = L_C - (l_{c1} + l_{c2})$$

There are four different error calculations within the condenser model; the error associated with heat rejected in the superheated region, the two-phase region, the subcooled region and the overall heat rejected in the condenser. In the case of the condenser, U_{c1}, U_{c2} and U_{c3} were solved for simultaneously in order to reduce the overall error associated with the condenser calculations. The outcome showed varying lengths of the heat exchanger segments at each of the test conditions but single values for the heat transfer coefficients. Again, when comparing the computed heat rejection rates at each of the regions as calculated versus measured, there resulted in various concerns. Most of these concerns fell within the subcool equation but the heat rejection required during this region only contributed 6% at most to the total heat rejection required. The overall percent error ranged from 0.96% to 16.22%.

Compressor

The compressor analysis is straight forward in solving for the unknown parameters essential to the model. The speed of the compressor was given from the manufacturer's provided information and, for testing purposes, the scroll compressor used as seen in Figure 11, operated at a speed of 1800 RPM. Using this added knowledge along with other test conditioned data all but two parameters were known.

Figure 11: Copeland ZP31K5E-PFV-830 Scroll Compressor

Using the twelve test conditions, an analysis was done to extract the compressor coefficient and the effective displacement volume. Using the error equation when comparing the flow rate found using the compressor mass flow rate relationship, Equation 9, and the flow rate under testing conditions the error ranged from 0.08% to 2.96%.

Flow Restrictor

Similar to the compressor, the flow restrictor analysis was straightforward in determining the unknown parameter. The Goodman air conditioning unit used in this analysis has a fixed orifice flow restrictor which made the results more consistent and predictable because the opening area of the valve was constant. Looking at the flow rate relationship at the flow restrictor, Equation 12, it is clear that, in this case, the only missing parameter is the valve coefficient (X.-D. He 1996).

Figure 12: 0.065 Flow Restrictor

An initial assumption was that the valve coefficient followed a sharp edge orifice design which would result in a 0.61 coefficient as derived from Bernoulli's equation for orifice operation (Munson, et al. 2009). However, looking at the orifice for this model as seen in Figure 12 one can tell that this is not a strictly sharp edge orifice so the coefficient needed to be determined. From the manufacturer's information this VCC unit required a 0.065 Flowrator making the diameter of the opening known to be 0.065 inches. From there the opening area could be determined and the remaining values to accommodate Equation 12 can be found under the test conditions.

The resulting valve coefficient for this model was 0.6719 which validates the initial assumption because it was close yet a bit more efficient, like the design reflects. Using the error equation to compare the flow rate computed at the flow restricting valve to the flow rate measured under test conditions this value resulted in an error ranging from 0.0% to 6.8%.

Parameter Result

To conclude this section, all parameters needed to run the final model were determined using information provided within the manufacturers test conditions. Figure 13 summarizes all the information needed and all the parameters that were determined.

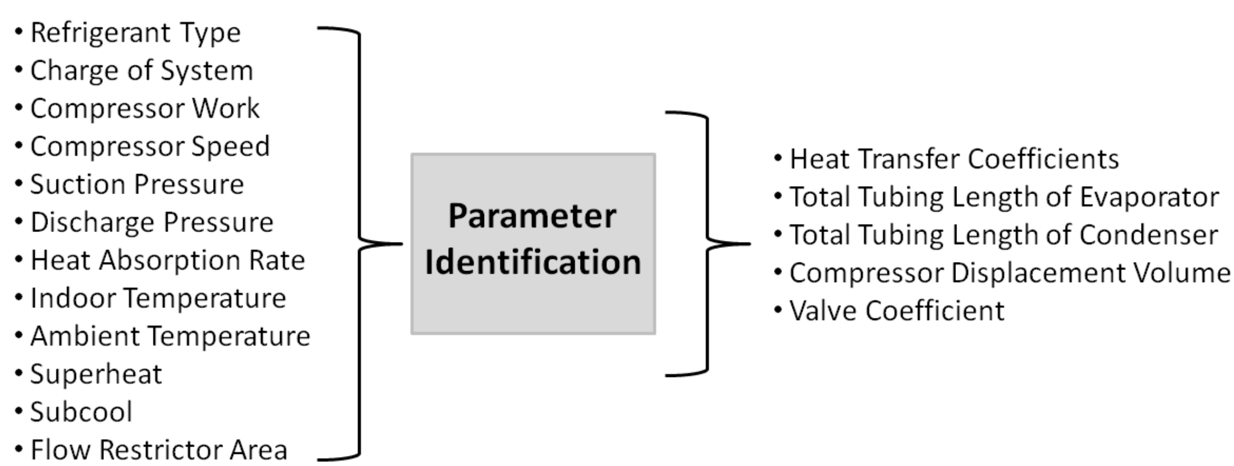

Figure 13: Block Diagram Summarizing Parameter Results

CHAPTER SIX: MODEL VALIDATION

User Input

An advantage of using this model is that the estimation can be made without expensive equipment to measure flow rate like previous research has required. Once the analysis of the manufacturer provided test conditions is done the user will be able to automatically generate the spreadsheet with known values. These values include the following:

- Values directly from manufacturer's data (or physical inspection)
    - Rated capacity
    - Refrigerant type
    - Total refrigerant charge
    - Pipe diameter
    - Speed of compressor
    - Area of valve opening
        - If fixed orifice is *not* used this will need to be a value determined from the test condition analysis
- Parameters derived from test condition analysis
    - Heat transfer coefficient for each of the five total regions
    - Total length of tubing within evaporator
    - Total length of tubing within condenser
    - Compressor displacement volume

- Compressor coefficient
- Valve coefficient

Outcome

When each of the above noted factors are entered into the final model the *Solver* function can be ran to obtain all of the appropriate remaining results. When complete the final analysis will provide the following information so that each component within the system is defined and known:

- Lengths of each heat exchanger region
- Flow rate of refrigerant
- Compressor input work required
- Heat rejection rate at the condenser
- Heat absorption rate at the evaporator
- Degrees of superheat at the evaporator exit
- Degrees of subcool at the condenser exit

One potential drawback to using Excel is that the *Solver* function is highly sensitive to initial conditions. To mitigate this problem, the model utilizes predetermined initial conditions to be used on the first *Solver* run. These conditions are based off of the user input to get a close “guess” to speed up the run time and increase the efficiency of the model. Because this model is so sensitive to initial conditions the *Solver* function may need to be run multiple times. The most number of times it needed to be run during this study was three times to get the most efficient and accurate outcome of how the system should be working.

Results

Once the model was complete validation was required. There were a total of eight equations to be used and six unknown values to be determined. In the analysis the mass flow rate had the most dramatic impact to the percent error when the entire system was being reviewed. Model performance is particularly sensitive to mass flow rate and by altering this value the percent error at each component throughout the system was dramatically affected. The next item that caused a significant change was the lengths of each region. By increasing and decreasing these lengths the heat exchanger percent errors were affected along with the mass error. Lastly, the change that caused the least impact was changing the amount of superheat and subcool. When these two parameters were altered the only thing that was slightly affected was the heat exchanger and the orifice equation, only when subcool was altered.

When looking into the sensitivity of the effective heat transfer values it was found that there was direct relationship from this value to the percent error. When the effective heat transfer values were the only thing that changed the percent error was changed by the same magnitude. This relationship was less direct and obvious as the heat transfer required within the various heat exchanger regions was proportionally smaller than the heat transfer required over the entire heat exchanger.

Utilized Test Conditions

The model was used in an attempt to predict the results as provided in the test conditions. The required input information was entered and the resulting values were compared against the provided measured data. When this was done it showed an average 3.5% error when predicting flow rate against what was thermodynamically computed with the test data, 2.9% error when predicting heat absorption rate compared to measured test data, 2.8% error when predicting heat rejection rate when compared to measured test data and a 1.6% error when predicting the input energy required compared to measured test data. The values that the model was least likely to predict correctly were the subcool zone length of the condenser, the superheat zone length of the evaporator, and the degrees of superheat.

New Test Conditions

Considering that twelve test conditions were used to solve for the parameters it is obvious that the results reflect this in low percent errors. Error values reported to this point are indicative of the “goodness of fit” for the parameter values. In order to accurately test the model, one must test it under different conditions than those used to solve for the parameter values. From the manufacture’s data twelve additional tests were analyzed. These new conditions provided the required inputs from the user to run the model (suction pressure, discharge pressure, set point temperature and ambient air temperature) as well as the input work required and the measured rate of heat absorption. These conditions use temperature ranges that are more uncommon to traditional air conditioner use. The error was calculated using Equation 30, the heat absorption rate as predicted by the model and the heat absorption rate as measured under test conditions. When the model was used the measured value was predicted within 0.4% to 7.3% of the actual measured data. The 7.3% error was an outlier of the results, however, and the average error over the twelve new conditions was 3.7% which is more reasonable and expected. As an additional check, the compressor efficiency was calculated between

71.3% and 89.6% which correlated with what was found using all the initial test conditions.

Replication of Model

To further test the validity of this approach, the process was replicated for two more air conditioner models; a five ton Goodman air conditioner model, PC1460H41, and a 3 ton Bard air conditioner model, PA13362A. The Goodman brand was used again to maintain an established relationship with manufacturer's representative and to verify the size difference was not going to be a problem with the model. Again, this model can be duplicated for any brand of VCC following the steps laid out as shown in Figure 14.

In order to verify the model's applicability for various VCC brands it was used against a different 3 ton unit, the Bard unit. The only difference in this model was that the full analysis needed to be slightly modified to accommodate a TX valve. This means that the valve area in the flow restrictor spreadsheet varied for each test condition to modify flow rate to meet specified superheat conditions. At first this alteration seems to add an additional unknown to the full model but once the manufacturer representative was contacted it was confirmed that the TX valve was operated to maintain 10° Fahrenheit superheat at the evaporator exit and 10° Fahrenheit subcool at the condenser exit under their test conditions. So, in reality, the analysis for the Bard model added one unknown to the full model, the valve area, but it eliminated two, the superheat and the subcool.

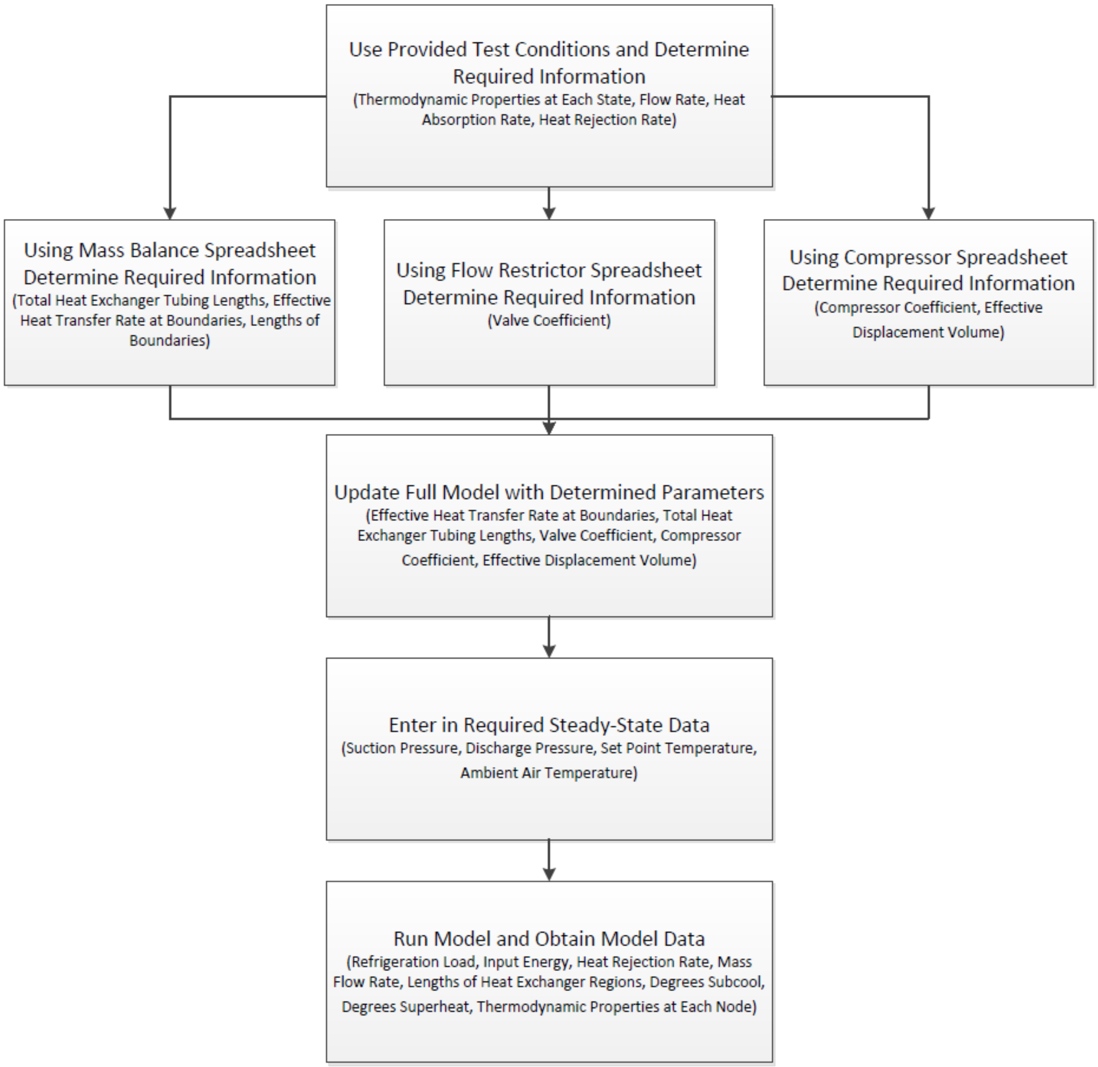

Figure 14: Process Flow Chart of Model Generation

Like the initial Goodman model, the remaining two VCC units were put to the test to ensure the model was providing an accurate simulation. The parameters were solved for using twelve different test conditions and an analysis of the results of each air conditioner model can be seen in Table 4. Once these parameters were determined and populated into the full model the original test conditions were put into the model to see if they provided the same results as to what was measured during the lab tests for the manufacturer.

Table 4: Parameters at Each VCC Unit

Measured Value & Units (where applicable)	Goodman 3 Ton Unit	Goodman 5 Ton Unit	Bard 3 Ton Unit
L_E $[ft]$	102.8	176.1	149.2
U_{e1} $\left[\frac{BTU}{hr * ft * F}\right]$	8.1	7.7	7.5
U_{e2} $\left[\frac{BTU}{hr * ft * F}\right]$	3.2	2.9	1.8
L_C $[ft]$	92.6	150.9	158.9
U_{c1} $\left[\frac{BTU}{hr * ft * F}\right]$	19.7	31.8	3.3
U_{c2} $\left[\frac{BTU}{hr * ft * F}\right]$	40.3	38.0	29.0
U_{c3} $\left[\frac{BTU}{hr * ft * F}\right]$	23.3	19.1	4.4
V_k $[ft^3]$	0.0004	0.00074	0.0004
C_k	0.3	0.4	0.4
C_v	0.7	0.6	0.6

To mimic the process used for the initial Goodman 3 Ton unit, twelve additional test conditions were input into the model to see how close it was to predicting the measured values. Table 5 shows the results of the simulation against the three different air conditioning units. The values show how close the model came to predicting what was found in the manufacturer's test conditions. Test conditions with the most information were used to determine the initial parameters required to run the model and that is why there is more of a comparison with the "identification" test data. However, for all seventy-two conditions, there was enough information to compare the cooling load from the simulation to the measured data. As one can see, the percent difference in predicting this value ranged from 1.5% to 3.7%.

Table 5: Final Results of Simulated VCC Units

Model	Test Data	$\dot{Q}_L$ Average % Difference	$\dot{Q}_H$ Average % Difference	$\dot{m}$ Average % Difference	A_v Average % Difference
Goodman 3 Ton Unit	Identification	2.9%	2.8%	3.5%	N/A
	Validation	3.7%	-	-	N/A
Goodman 5 Ton Unit	Identification	1.5%	1.5%	1.2%	N/A
	Validation	2.0%	-	-	N/A
Bard 3 Ton Unit	Identification	2.4%	2.4%	2.4%	3.2%
	Validation	2.3%	-	-	-

Distribution of Test Conditions

The determination of the parameters is ultimately driven by the temperatures and the pressures of the system. Initially, when reviewing what test condition values would be most beneficial to the analysis, the values with the most manufacturer provided information were used. After the first model was completed, a review of these values was done to see if they would be good starting points for this model to be run with other air conditioner specifications. After review of the manufacturer's published data, the justification for the expanded information with these temperatures, was because that is where majority of the operation of the unit takes place. The bulk of an air conditioner's average use is at set point temperatures between 75° and 85° while the ambient temperatures are between 65° and 95°. Because this is where the air conditioner typically gets the most use, these value ranges are justified for use in the determination of parameters.

When the model was being validated various test conditions were reviewed to see if the model would predict the measured data. The temperature ranges for this additional testing commonly was the more extreme operating conditions as well as a few conditions which were dispersed within the average use. Using these values to validate proved that the model could predict the outcome regardless of operating temperature extremes. Figure 15 shows the temperature distribution used when determining the parameters needed for the model compared to the temperature distribution used when validating the model.

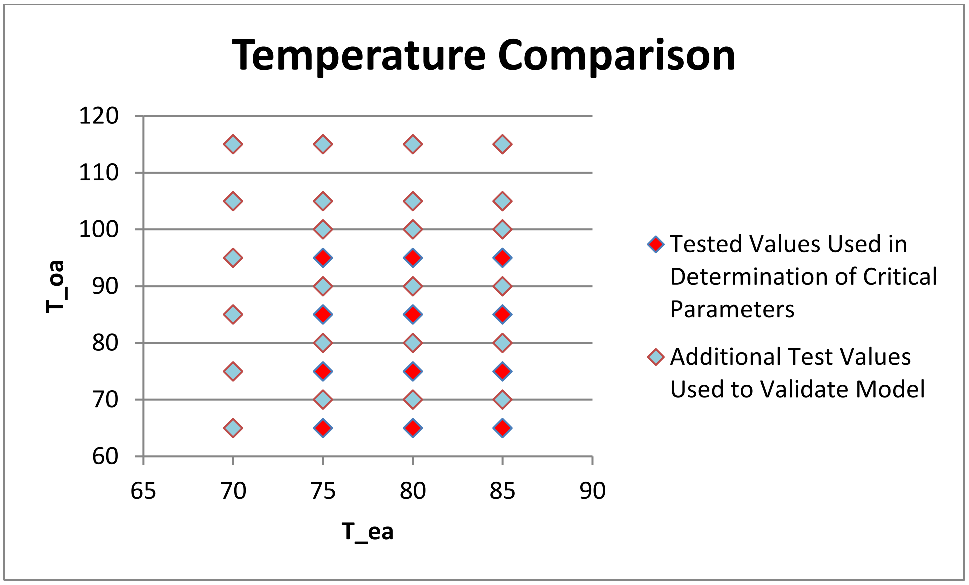

Figure 15: Temperature Selection for Model Generation and Validation

CHAPTER SEVEN: CONCLUSION

This paper addresses one of the barriers in applying VCC performance models, the identification of parameter values required to make these models useful. Using data found from manufacturer performance tests to operate a VCC model will allow this process to be replicated for a wide variety of air conditioning units ranging in sizes and complexity. More specifically, to determine the heat transfer characteristics of a given heat exchanger is a highly important parameter used both for performance optimization and prediction. Using effective heat transfer values allows for the spreadsheet-based model to represent a broad spectrum of air conditioner units despite their potential differences in heat exchanger designs that is not dependent on the number and spacing of fins or other optimization design criteria. Most importantly, these data allowed for the determination of the effective heat transfer characteristics, as opposed to values computed strictly from the geometry.

As proof of concept, the approach was used to identify parameter values for three different air conditioner models; one five ton model and two three ton models of different brands. All that was required to run the analysis of each of these VCC units was the readily available manufacturer's test data pulled from their websites. On average, the analysis predicted a heat absorption in the evaporator within 1.5% to 3.7% of what the test conditions provided for seventy-two different test conditions.

Research Contributions

This model has the potential to become an important tool for VCC designers, building designers, building energy managers and utility companies as well as those who are attempting to optimize building energy performance through the use of model-based control systems.

VCC System Designers

Applying the moving boundary method is not only computationally efficient; it also provides insight to see the effects of heat transfer and their locations. All research pertaining to the boundary method application demonstrates that majority of the available heat transfer occurs during the two-phase flow. This knowledge can assist VCC designers because they can increase efficiency by optimizing heat transfer capabilities at these locations. There is also an opportunity to review lengths of superheat and subcool and potentially reduce additional unnecessary tubing.

In addition to reviewing the zone lengths of each heat exchanger, a designer can look at optimizing the design by increasing the effective heat transfer per unit length. If they were to add fins and yet the heat transfer remained the same the fin addition was irrelevant and added cost.

Building Designers

A major part of designing a building is looking at the overall energy consumption potential. Using this tool a building designer can predict how an air conditioning unit will work and how much energy it will require to operate. It could also potentially help with the overall design of how the airflow should flow through a building and where specifically the unit should be located. This tool can help optimize their design to improve function for the building owner.

Building Energy Managers

There are a few resources to understand how much energy a VCC unit will require under various operating conditions and using manufacture specifications doesn't quite paint the whole picture. For example, the same air conditioning unit will operate differently in a cooler climate than it would a warmer one. Using this tool will allow the building energy manager to get a much clearer view of how any given unit will operate in a given building in a particular climate.

Utility Companies

Utility companies are constantly analyzing energy consumption data to determine how much energy to have on demand for distribution. Being able to predict how much energy consumption will be required on any given day is huge in minimizing wasted energy and therefore reducing the overall cost. Considering space conditioning is one of the biggest contributors to energy consumption it would be of great use to understand how much energy will be required throughout any given day to cool the building.

Future Research

While it is believed that this model provides an in depth look at VCC systems that can be applied to a wide variety of specific models, there are still shortcomings that could be eliminated in future research. There are many things mentioned in this paper that are simplified for purposes of speed, user capability, and cost. In some cases the simplification is used as it proved to be an acceptable assumption because the changes were minimal. However, future research could consider the following:

- Pressure sensors are required to do the analysis as it is currently laid out. It would be interesting to do this study checking the effectiveness of temperature sensors in lieu of pressure sensors. This would eliminate the need to calculate the degrees of superheat entering the compressor because there could be temperature sensors at the inlet of the evaporator and the inlet of the compressor. Same would apply for calculating the degrees of subcool entering the flow restrictor. A temperature sensor could be added at the inlet of the condenser and the inlet of the flow restrictor to determine actual refrigerant temperatures at these locations. As noticed earlier, the degrees of superheat and subcool don't make a huge impact to the overall analysis and four temperature sensors would be required to run the analysis this way versus the two pressure sensors in the original model layout.

  - Another way of obtaining the more precise information to run the model would be to purchase equipment to determine the flow rate. However, this is expensive equipment that is difficult to repair if needed.

- This model assumes there is no pressure drop in the heat exchangers. While the pressure drops are minor they could still be applicable. It would be interesting to see the overall effects when comparing the assumption from the original model to the calculations when a pressure drop is considered. This would, however, require additional pressure sensors to be added to the system and to the model.
- While a fixed orifice flow restrictor may be the most inexpensive option and therefore the most common for residential units it is not the most efficient. If the purpose of this model is to increase energy awareness it is the hope that a more sophisticated flow restricting system would be utilized, even if it is a retrofit condition. Future research could consider the same model with a more efficient system, like a TX valve or an EE valve.

- This model utilizes an average value for the mean void fraction. Considering this is not the most accurate assumption it would be interesting to look into Beck and Wedekind’s work a bit further and see the effects on utilizing a time-varying mean void fraction value (Beck and Wedekind 1981).
- Considering the VCC models used for this study are all packaged residential units there are not too many complex components to the system. As mentioned in previous sections of this paper, these systems can get very complex and incorporate additional components like accumulators and additional piping if it was a split system. Future work could look into the effects of adding some of these components. Not only would this require the system to be more complex and difficult to model it would also have to utilize a different mean void fraction equation for flows that may or may not fully evaporate or condense. This design would be very difficult and would really only be applicable in a commercial application but still a worthwhile study.
- Whereas the model used to develop this method was steady-state, the parameter values can be used in a dynamic model which can be investigated for advanced control schemes as well as real-time performance monitoring utilizing state observers.

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APPENDIX A

Effective Heat Transfer Parameter Derivation

The work on developing and modeling heat exchangers using lumped parameters provided in (X.-D. He 1996) is highly sophisticated and a basis of research for this paper. The purpose of the moving boundary method is to divide the heat exchangers into various control volumes based off of the particular refrigerant phase. In the case of the condenser these flow characteristics that make up the regions include the superheated vapor, the two-phase flow and the subcooled liquid. In the case of the evaporator the flow characteristics include the two-phase flow and the superheated vapor.

The main difference between the original work and the development described in this paper is the use of heat transfer values. This paper reflects an effective heat transfer value as opposed values computed strictly from the geometry. Below is the derivation process used for each of the equations for the different zones within the heat exchangers. The specific one derived below is for the superheated zone within the condenser. All of the "original" equations are directly from (X.-D. He 1996) and the "modified" equations are the modified equations that utilize the nomenclature used throughout this paper.

Equation 32: Original: Heat Transfer between Tube Wall and Air

$$\alpha_{i1}\pi D_i(T_{r1} - T_{w1}) + \alpha_o \pi D_o(T_a - T_{w1}) = 0$$

Equation 33: Modified: Heat Transfer between Tube Wall and Air

$$\alpha_{ci1}\pi D_{ci}(T_{cr1} - T_{cw1}) + \alpha_{co}\pi D_o(T_{oa} - T_{cw1}) = 0$$

Equation 34: Original: Heat Transfer between Refrigerant and Tube Wall

$$\dot{m}_i h_i - \dot{m}_i h_g + \alpha_{i1}\pi D_i L_1(T_{w1} - T_{r1}) = 0$$

Equation 35: Modified: Heat Transfer between Refrigerant and Tube Wall

$$\dot{m}(h_2 - h_{2\prime}) + \alpha_{ci1}\pi D_{ci} l_{c1}(T_{cw1} - T_{cr1}) = 0$$

In order to eliminate the wall temperature to gain an effective heat transfer T_{cw1} from Equation 35 must be solved for, thus turning into Equation 36

Equation 36: Modified: Wall Temperature

$$T_{cw1} = T_{cr1} - \frac{\dot{m}(h_2 - h_{2'})}{\alpha_{ci1}\pi D_{ci} l_{c1}}$$

At this point there is the matter of substituting Equation 36 into Equation 33. Before simplification this becomes

Equation 37: Superheated Flow within the Condenser

$$\alpha_{ci1}\pi D_{ci}\left(T_{cr1} - \left[T_{cr1} - \frac{\dot{m}(h_2 - h_{2'})}{\alpha_{ci1}\pi D_{ci} l_{c1}}\right]\right) + \alpha_{co}\pi D_o\left(T_{oa} - \left[T_{cr1} - \frac{\dot{m}(h_2 - h_{2'})}{\alpha_{ci1}\pi D_{ci} l_{c1}}\right]\right) = 0$$

$$\rightarrow \frac{\dot{m}(h_2 - h_{2'})}{l_{c1}} + \alpha_{co}\pi D_o\left[(T_{oa} - T_{cr1}) + \frac{\dot{m}(h_2 - h_{2'})}{\alpha_{ci1}\pi D_{ci} l_{c1}}\right] = 0$$

$$\rightarrow \left\{\frac{\dot{m}(h_2 - h_{2'})}{l_{c1}} + \alpha_{co}\pi D_o\left[(T_{oa} - T_{cr1}) + \frac{\dot{m}(h_2 - h_{2'})}{\alpha_{ci1}\pi D_{ci} l_{c1}}\right]\right\} * \alpha_{ci1}\pi D_{ci} l_{c1} = 0$$

$$\rightarrow [\dot{m}(h_2 - h_{2'}) * \alpha_{ci1}\pi D_{ci}] + [\alpha_{co}\pi D_o * \dot{m}(h_2 - h_{2'})] + [\alpha_{co}\pi D_o * \alpha_{ci1}\pi D_{ci} l_{c1} * (T_{oa} - T_{cr1})] = 0$$

$$\rightarrow \dot{m}(h_2 - h_{2'}) * [\alpha_{ci1}\pi D_{ci} + \alpha_{co}\pi D_o] + [\alpha_{co}\pi D_o * \alpha_{ci1}\pi D_{ci} * l_{c1} * (T_{oa} - T_{cr1})] = 0$$

$$\rightarrow \dot{m}(h_2 - h_{2'}) + \frac{\alpha_{ci1}\pi D_{ci} * \alpha_{co}\pi D_o}{\alpha_{ci1}\pi D_{ci} + \alpha_{co}\pi D_o} * l_{c1} * (T_{oa} - T_{cr1}) = 0$$

$$Let\ U_{c1} = \frac{(\alpha_{ci1}\pi D_{ci} * \alpha_{co}\pi D_o)}{(\alpha_{ci1}\pi D_{ci} + \alpha_{co}\pi D_o)}$$

$$\rightarrow\ \dot{m}(h_2 - h_{2\prime}) + U_{c1} * l_{c1} * (T_{oa} - T_{cr1}) = 0$$

$$\rightarrow\ \dot{m}(h_2 - h_{2\prime}) = l_{c1} * U_{c1} * (T_{cr1} - T_{oa})$$

Once simplified the equation becomes Equation 20 knowing that U_{c1} is to be considered the effective heat transfer per unit length. The remaining derived equations for the boundary lengths at the condenser and the evaporator follow this form, but for the sake of brevity are not shown.

APPENDIX B

Mean Void Fraction Derivation

Much of Wedekind’s research surrounds the use of the mean void fraction within the moving boundary method. Using a mean void fraction model can be applied to calculate the mass within the two-phase flow portion of the heat exchanger. This component allows us to predict the amount of vapor refrigerant within the evaporator and the condenser throughout the two-phase flow. The mean void fraction is imperative in the use of the lumped parameter method to forecast the transient responses within these heat exchangers.

Zivi’s model as laid out by Wedekind was used in this analysis because it is a simple closed form and when compared to other models there wasn’t much difference (G.L. Wedekind 1976). All of the “original” equations are directly from Wedekind, Bhatt and Beck’s article and the “modified” equations are the adapted equations that utilize the nomenclature used throughout this paper (G.L. Wedekind 1976). Once the modified equations are simplified they become Equation 25 and Equation 27 as used in this study.

Equation 38: Original: Mean Void Fraction for Evaporator

$$\bar{\alpha}_s = \frac{1}{(1-c)} + \frac{c}{(1-\bar{x}_0)(1-c)^2} * ln[c + (1-c)\bar{x}_0]$$

$$c = \left(\frac{\rho'}{\rho}\right)^{\frac{2}{3}}$$

$$\rho' = density\ at\ outlet$$

$$\rho = density\ at\ inlet$$

$$\bar{x}_0 = quality\ at\ inlet$$

Equation 39: Modified: Mean Void Fraction for Evaporator

$$\bar{\gamma}_E = \frac{1}{\left(1-\left(\frac{\frac{1}{v_{4\prime}}}{\frac{1}{v_4}}\right)^{\frac{2}{3}}\right)} + \frac{\left(\frac{\frac{1}{v_{4\prime}}}{\frac{1}{v_4}}\right)^{\frac{2}{3}}}{(1-x_4)\left(1-\left(\frac{\frac{1}{v_{4\prime}}}{\frac{1}{v_4}}\right)^{\frac{2}{3}}\right)^2}$$

$$* \, ln\left[\left(\frac{\frac{1}{v_{4\prime}}}{\frac{1}{v_4}}\right)^{\frac{2}{3}} + \left(1-\left(\frac{\frac{1}{v_{4\prime}}}{\frac{1}{v_4}}\right)^{\frac{2}{3}}\right) * x_4\right]$$

$$\rightarrow \bar{\gamma}_E = \frac{1}{\left(1-\left(\frac{v_4}{v_{4\prime}}\right)^{\frac{2}{3}}\right)} + \frac{\left(\frac{v_4}{v_{4\prime}}\right)^{\frac{2}{3}}}{(1-x_4)\left(1-\left(\frac{v_4}{v_{4\prime}}\right)^{\frac{2}{3}}\right)^2} * ln\left[\left(\frac{v_4}{v_{4\prime}}\right)^{\frac{2}{3}} + \left(1-\left(\frac{v_4}{v_{4\prime}}\right)^{\frac{2}{3}}\right) * x_4\right]$$

Equation 40: Original: Mean Void Fraction for Condenser

$$\bar{\alpha}_s = \frac{1}{(1-c)} + \frac{c}{(\bar{x}_0)(1-c)^2} * ln\left[\frac{c}{(1-c)\bar{x}_0 + c}\right]$$

$$c = \left(\frac{\rho'}{\rho}\right)^{\frac{2}{3}}$$

$$\rho' = density\ at\ inlet$$

$$\rho = density\ at\ outlet$$

$$\bar{x}_0 = quality\ at\ inlet = 1\ (saturated\ vapor)$$

Equation 41: Modified: Mean Void Fraction for Condenser

$$\bar{\gamma}_C = \frac{1}{(1-c)} + \frac{c}{(1-c)^2} * ln[c]$$

$$\rightarrow \bar{\gamma}_C = \frac{1}{\left(1-\left(\frac{\frac{1}{\upsilon_{2\prime}}}{\frac{1}{\upsilon_{2\prime\prime}}}\right)^{\frac{2}{3}}\right)} + \frac{\left(\frac{\frac{1}{\upsilon_{2\prime}}}{\frac{1}{\upsilon_{2\prime\prime}}}\right)^{\frac{2}{3}}}{\left(1-\left(\frac{\frac{1}{\upsilon_{2\prime}}}{\frac{1}{\upsilon_{2\prime\prime}}}\right)^{\frac{2}{3}}\right)^2} * ln\left[\left(\frac{\frac{1}{\upsilon_{2\prime}}}{\frac{1}{\upsilon_{2\prime\prime}}}\right)^{\frac{2}{3}}\right]$$

$$\rightarrow \bar{\gamma}_C = \frac{1}{\left(1-\left(\frac{\upsilon_{2\prime\prime}}{\upsilon_{2\prime}}\right)^{\frac{2}{3}}\right)} + \frac{\left(\frac{\upsilon_{2\prime\prime}}{\upsilon_{2\prime}}\right)^{\frac{2}{3}}}{\left(1-\left(\frac{\upsilon_{2\prime\prime}}{\upsilon_{2\prime}}\right)^{\frac{2}{3}}\right)^2} * ln\left[\left(\frac{\upsilon_{2\prime\prime}}{\upsilon_{2\prime}}\right)^{\frac{2}{3}}\right]$$

APPENDIX C

Parameter Tuning

Parameter Tuning

In order to run the model accurately, constraints were required to be placed on the changing parameters. This allowed Excel to run at the fastest speed possible while considering all possible results.

Test Conditions

The *Solver* function in Excel was not required to gain all of the traditional thermodynamic information required from the twelve test conditions. The input information from test conditions and data from the air conditioning unit was all that was needed to run the analysis with the thermodynamic add-in previously noted.

Mass Balance

While completing the analysis for the mass balance throughout the system the analysis at the condenser and the evaporator was done simultaneously. Once *Solver* was complete the overall tubing length at the evaporator and the condenser was determined, the region lengths within the heat exchangers were determined and the effective heat transfer values per unit length were determined. Based off of the geometry of the VCC and the findings from the test conditions the following constraints were placed on the model:

- $l_{e2} \geq 0.01 * L_E$
- $50 \leq L_E \leq 200$
- $l_{c3} \geq 0.01 * L_C$
- $50 \leq L_C \leq 200$

Compressor

The *Solver* function in Excel required no additional constraints to gain all of the required information on the compressor. The input information from test conditions and data from the air conditioning unit was all that was needed to solve for the unknowns.

Flow Restrictor

The *Solver* function in Excel required no additional constraints to gain all of the required information on the flow restrictor. The input information from test conditions and data from the air conditioning unit was all that was needed to solve for the single unknown.

Full Model

The final VCC model was complex for any modeling software. Considering, this type of work is not traditionally done in a spreadsheet-based analysis there was some component tuning required to reduce overall run time and increase accuracy. In order to have the analysis run properly initial conditions had to be placed as "stand in" values for what was to be determined. Below is a list of all the components being solved for in this analysis and their initial guess for each parameter. The initial assumptions were determined by using relationships seen within the test conditions. In some cases there wasn't a clear relationship so averages were used.

- Superheat boundary length for the condenser
  - Initial Assumption: $Average\left(\frac{Test\ Condition\ l_{c1}}{Test\ Condition\ T_{oa}}\right) * T_{oa}$
- Two-Phase boundary length for the condenser
  - Initial Assumption: $Average\left(\frac{Test\ Condition\ l_{c2}}{Test\ Condition\ P_{C}}\right) * P_{C}$
- Two-Phase boundary length for evaporator

- Initial Assumption: $Average(Test\ Condition\ l_{e1})$
- Flow Rate
  - Initial Assumption: $Average(Test\ Condition\ \dot{m})$
- Superheat
  - Initial Assumption: $Average(Test\ Condition\ Superheat)$
- Subcool
  - Initial Assumption: $Average(Test\ Condition\ Subcool)$

Once the input information is in and the initial assumptions have been populated the analysis is ready to be run. The model calculates percent error at the condenser, evaporator, mass balance, compressor and flow restrictor. The main function of *Solver* is to reduce the overall error by changing the properties listed above. To get results that are more accurate and at a reasonable time lapse, the following constraints were placed on the model. These constraints were determined based off of results of test conditions.

- $0.05 * L_C \leq l_{c1} \leq 0.4 * L_C$
- $0.3 * L_C \leq l_{c2} \leq 0.9 * L_C$
- $l_{c3} \geq 0.01 * L_C$
- $0.6 * L_E \leq l_{e1} \leq 0.99 * L_E$
- $l_{e2} \geq 0.01 * L_E$
- $7 \leq superheat \leq 50$
- $8 \leq subcool \leq 12$
- $\dot{Q}_L \leq 1.5 * Rated\ Capacity$
- $\dot{Q}_H \leq 0.8 * Rated\ Capacity$

APPENDIX D

Spreadsheet Modeling Examples

Many spreadsheets were built in order for this model to run correctly. A view of the main spreadsheet models and what they were required to calculate can be seen below. A key to navigate the cell colors can be seen in Table 6.

Table 6: Excel Highlight Key

Green Cells	Objective Cell
Blue Cells	Parameters to be Solved
Yellow Cells	Input Information from User
Pink Cells	Previously Calculated Information from Other Spreadsheet
Clear Cells	Automatically Calculated

Test Conditions

Below is an example of one of the spreadsheets used to create this model. As noted, there are twelve test conditions that were utilized to find the parameters required. The spreadsheet used to calculate this information is shown and there was not a need for Excel *Solver* to computer any parameter on this spreadsheet, it all came from calculations utilizing thermodynamic properties and the input information regarding parameters specific to the air conditioning unit and then parameters given from the test that had been previously done on this unit.

Air Conditioner Parameters:

Refrigerant	=	R-410A	
Rated Capacity	=	36,000	BTU/h
Isentropic Efficiency	=	85%	
Evap Fan Work	=	0.5	hp
Cond Fan Work	=	0.25	hp
Compressor Speed	=	1800	RPM
Charge of System	=	65	oz
Diameter of Flow Restrictor	=	0.065	in
Inner Diameter of Evaporator Tubing:	=	0.45	in
Inner Diameter of Condenser Tubing:	=	0.45	in

Test Condition Values:

Tea	=	75	F
Toa	=	95	F
Pe=P1=P4	=	125	psia
Pc=P2=P3	=	350	psia
QdotL	=	33000	BTU/h
Ptotal	=	2.84	kW
Superheat	=	10	F
Subcool	=	10	F

Compressor Work:

Compressor Work:
(from test conditioned data)

WdotIn	=	**7782.160**	BTU/h

Compressor Work:
(calculated with thermo properties)

WdotIn	=	**6433.981**	BTU/h

Compressor Efficiency:

nc	=	**83%**	BTU/h

Heat Transfer:

Condenser

QdotH	=	**39434.0**	BTU/h	#
QdotSH	=	**6997.5**	BTU/h	
QdotSAT	=	**29927.7**	BTU/h	
QdotSUB	=	**2508.8**	BTU/h	

Evaporator

QdotSH	=	**1198.9**	BTU/h
QdotSAT	=	**31801.1**	BTU/h

Flow Rate:

mdot	=	**437.3758**	lbs/hr	=QdotL/(h1-h4)
mdot	=	**437.3758**	lbs/hr	=QdotH/(h2-h3)
mdot	=	**437.3758**	lbs/hr	=Wdot/(h2-h1)

Thermodynamic Properties:

State 1 (Superheat):

T1	=	**46.37941**	F	=T4'+Superheat
P1	=	**125**	psia	
h1	=	**184.1438**	BTU/lbm	
s1	=	**0.43665**	BTU/lbm*R	
v1	=	**0.504031**	ft^3/lbm	

State 2s (Superheat): *** Without isentropic efficiency

P2	=	**350**	psia
h2s	=	**196.64772**	BTU/lbm
s2	=	**0.4366592**	BTU/lbm*R

State 2 (Superheat): *** With isentropic efficiency

T2	=	**149.98435**	F	
P2	=	**350**	psia	
h2	=	**198.85428**	BTU/lbm	=h1+((h2s-h1)/ns)
s2	=	**0.4366592**	BTU/lbm*R	
v2	=	**0.1994366**	ft^3/lbm	

State 2' (Saturated Vapor):

T2'	=	**103.8374**	F
P2'	=	**350**	psia
h2'	=	**182.85548**	BTU/lbm
s2'	=	**0.4130457**	BTU/lbm*R
v2'	=	**0.1555845**	ft^3/lbm

State cr1 (Average Between 2 & 2'):

Tcr1	=	**126.91088**	F
Pcr1	=	**350**	psia
hcr1	=	**190.85488**	BTU/lbm
scr1	=	**0.4248525**	BTU/lbm*R
vcr1	=	**0.1775106**	ft^3/lbm

State 2'' (Saturated Liquid):

T2''	=	**103.62372**	F
P2''	=	**350**	psia
h2''	=	**114.42986**	BTU/lbm
s2''	=	**0.291498**	BTU/lbm*R
v2''	=	**0.0164098**	ft^3/lbm

State 3 (Subcool):

T3	=	**93.623723**	F	=T2''-subcool
P3	=	**350**	psia	
h3	=	**108.69389**	BTU/lbm	
s3	=	**0.2772927**	BTU/lbm*R	
v3	=	**0.0150199**	ft^3/lbm	

State cr3 (Average between 2'' & 3):

Tcr3	=	**98.623723**	F
Pcr3	=	**350**	psia
hcr3	=	**111.56187**	BTU/lbm
scr3	=	**0.2843954**	BTU/lbm*R
vcr3	=	**0.0157148**	ft^3/lbm

State 4 (Vapor/Liquid Mixture):

T4	=	**36.282161**	F	
P4	=	**125**	psia	
h4	=	**108.69389**	BTU/lbm	=h3
s4	=	**0.2846022**	BTU/lbm*R	
v4	=	**0.119489**	ft^3/lbm	
x4	=	**0.2250504**		

v4g	=	**0.4834**	ft^3/lbm
v4f	=	**0.013807**	ft^3/lbm

State 4' (Saturated Vapor):

T4'	=	**36.379417**	F
P4'	=	**125**	psia
h4'	=	**181.40283**	BTU/lbm
s4'	=	**0.4312344**	BTU/lbm*R
v4'	=	**0.4833999**	ft^3/lbm

State er2 (Average between 4' & 1):

Ter2	=	**41.379417**	F
Per2	=	**125**	psia
her2	=	**182.77335**	BTU/lbm
ser2	=	**0.4339468**	BTU/lbm*R
ver2	=	**0.4937157**	ft^3/lbm

Parameter Determination

Once the information from each test condition was acquired, three additional spreadsheets were built to determine the parameters required for this model. These spreadsheets included the compressor analysis, the flow restrictor analysis and finally the mass balance analysis which included an analysis of total mass distribution as well as the effects on the evaporator and condenser. Below is an example of each of these spreadsheets.

<u>Compressor</u>

Unknown

V_k =	**0.00043**	ft^3
C_k =	**0.34098**	

Known

Omega =	**678584**	rad/hr

Compressor

	m_dot *lbs/hr*	v_1 *ft^3/lbm*	P_e *psia*	P_c *psia*	Flow Rate Difference	Total Error
Test 1a	**403.88**	**0.61024**	**109**	**241**	**-6.41**	**1.59%**
Test 2a	**414.10**	**0.56179**	**115**	**270**	**9.65**	**2.33%**
Test 3a	**427.36**	**0.53539**	**119**	**307**	**3.59**	**0.84%**
Test 4a	**437.38**	**0.50403**	**125**	**350**	**7.18**	**1.64%**
Test 1b	**404.53**	**0.61616**	**110**	**243**	**-10.78**	**2.66%**
Test 2b	**413.68**	**0.57204**	**116**	**273**	**2.16**	**0.52%**
Test 3b	**430.79**	**0.53847**	**120**	**310**	**-2.51**	**0.58%**
Test 4b	**452.27**	**0.50000**	**126**	**353**	**-4.23**	**0.94%**
Test 1c	**405.79**	**0.61631**	**111**	**245**	**-12.03**	**2.96%**
Test 2c	**416.78**	**0.57451**	**117**	**275**	**-2.56**	**0.61%**
Test 3c	**435.34**	**0.53517**	**122**	**313**	**-3.40**	**0.78%**
Test 4c	**456.35**	**0.49213**	**128**	**357**	**-0.38**	**0.08%**

Flow Restrictor

Unknown

C_v = **0.672**

Known

A_v = **0.000023** ft^2

Thermal Expansion Valve

	m_dot *lbs/hr*	v_3 *ft^3/lbm*	P_e *psia*	P_c *psia*	Flow Rate Difference	Total Error
Test 1a	**403.88**	**0.01340**	**109**	**241**	**-27.37**	**6.78%**
Test 2a	**414.10**	**0.01387**	**115**	**270**	**-13.02**	**3.14%**
Test 3a	**427.36**	**0.01442**	**119**	**307**	**5.86**	**1.37%**
Test 4a	**437.38**	**0.01502**	**125**	**350**	**26.96**	**6.16%**
Test 1b	**404.53**	**0.01343**	**110**	**243**	**-27.06**	**6.69%**
Test 2b	**413.68**	**0.01391**	**116**	**273**	**-10.64**	**2.57%**
Test 3b	**430.79**	**0.01446**	**120**	**310**	**4.11**	**0.95%**
Test 4b	**452.27**	**0.01506**	**126**	**353**	**13.47**	**2.98%**
Test 1c	**405.79**	**0.01347**	**111**	**245**	**-27.35**	**6.74%**
Test 2c	**416.78**	**0.01394**	**117**	**275**	**-12.95**	**3.11%**
Test 3c	**435.34**	**0.01450**	**122**	**313**	**0.00**	**0.00%**
Test 4c	**456.35**	**0.01511**	**128**	**357**	**10.63**	**2.33%**

Mass Balance

Unknown

Evaporator

L_E =	**102.812**	ft
C_e1 =	**8.075**	BTU/(hr*ft*F)
C_e2 =	**3.232**	BTU/(hr*ft*F)

Condenser

L_C =	**92.587**	ft
C_c1 =	**19.704**	BTU/(hr*ft*F)
C_c2 =	**40.303**	BTU/(hr*ft*F)
C_c3 =	**23.316**	BTU/(hr*ft*F)

Known

M_total =	**4.0625**	lbs
D_ei =	**0.0375**	ft
D_ci =	**0.0375**	ft

Mass Analysis

	l_e1 *ft*	l_e2 *ft*	l_c1 *ft*	l_c2 *ft*	l_c3 *ft*	Gamma_e	Gamma_c	M_E *lbs*	M_C *lbs*	M_total *lbs*	Total Error
Test 1a	**87.479**	**15.33**	**8.306**	**74.072**	**10.21**	**0.774**	**0.777**	**1.77**	**2.29**	**4.07**	**0%**
Test 2a	**93.446**	**9.37**	**9.110**	**76.599**	**6.88**	**0.772**	**0.764**	**1.89**	**2.15**	**4.04**	**0%**
Test 3a	**96.052**	**6.76**	**10.216**	**77.244**	**5.13**	**0.772**	**0.749**	**1.94**	**2.12**	**4.06**	**0%**
Test 4a	**101.783**	**1.03**	**11.127**	**80.535**	**0.93**	**0.772**	**0.733**	**2.04**	**2.00**	**4.04**	**1%**
Test 1b	**79.801**	**23.01**	**8.440**	**71.579**	**12.57**	**0.774**	**0.776**	**1.63**	**2.43**	**4.06**	**0%**
Test 2b	**84.143**	**18.67**	**9.278**	**71.709**	**11.60**	**0.772**	**0.763**	**1.72**	**2.41**	**4.13**	**2%**
Test 3b	**87.360**	**15.45**	**10.160**	**74.644**	**7.78**	**0.772**	**0.748**	**1.78**	**2.27**	**4.05**	**0%**

Test 4b	**92.667**	**10.14**	**11.328**	**80.333**	**0.93**	**0.773**	**0.731**	**1.87**	**2.01**	**3.88**	**5%**
Test 1c	**73.922**	**28.89**	**8.713**	**69.431**	**14.44**	**0.774**	**0.775**	**1.53**	**2.54**	**4.06**	**0%**
Test 2c	**77.237**	**25.58**	**9.298**	**70.179**	**13.11**	**0.772**	**0.762**	**1.60**	**2.50**	**4.10**	**1%**
Test 3c	**80.764**	**22.05**	**10.279**	**71.957**	**10.35**	**0.772**	**0.747**	**1.66**	**2.40**	**4.06**	**0%**
Test 4c	**86.366**	**16.45**	**11.097**	**74.373**	**7.12**	**0.772**	**0.730**	**1.76**	**2.31**	**4.07**	**0%**

Condenser

	Boundary Lengths			Heat Transfer Difference			
	l_c1	l_c2	l_c3	Superheat	Two-Phase	Subcool	Total Error
	ft	*ft*	*ft*	*BTU/h*	*BTU/h*	*BTU/h*	
Test 1a	**8.31**	**74.07**	**10.21**	**16.97**	**4672.51**	**201.67**	**12%**
Test 2a	**9.11**	**76.60**	**6.88**	**4.95**	**-380.04**	**-1159.06**	**4%**
Test 3a	**10.22**	**77.24**	**5.13**	**61.61**	**-2322.28**	**-1510.24**	**10%**
Test 4a	**11.13**	**80.53**	**0.93**	**-1.24**	**-1243.14**	**-2430.55**	**9%**
Test 1b	**8.44**	**71.58**	**12.57**	**40.32**	**5101.45**	**719.73**	**14%**
Test 2b	**9.28**	**71.71**	**11.60**	**52.44**	**0.01**	**-338.05**	**1%**
Test 3b	**10.16**	**74.64**	**7.78**	**-10.53**	**-1288.89**	**-1151.54**	**6%**
Test 4b	**11.33**	**80.33**	**0.93**	**-4.20**	**-158.66**	**-2081.16**	**6%**
Test 1c	**8.71**	**69.43**	**14.44**	**266.09**	**5523.59**	**970.34**	**16%**
Test 2c	**9.30**	**70.18**	**13.11**	**1.92**	**626.99**	**7.21**	**2%**
Test 3c	**10.28**	**71.96**	**10.35**	**44.20**	**-549.32**	**-723.71**	**3%**
Test 4c	**11.10**	**74.37**	**7.12**	**-70.76**	**-17.33**	**-1789.34**	**5%**

Evaporator

	Boundary Lengths		Heat Transfer Difference		
	l_e1	l_e2	Two-Phase	Superheat	Total Error
	ft	*ft*	*BTU/h*	*BTU/h*	
Test 1a	**87.48**	**15.33**	**27.43**	**-1002.48**	**3%**
Test 2a	**93.45**	**9.37**	**1.08**	**-787.73**	**2%**
Test 3a	**96.05**	**6.76**	**-24.75**	**-722.88**	**2%**
Test 4a	**101.78**	**1.03**	**21.36**	**-1087.16**	**3%**
Test 1b	**79.80**	**23.01**	**-7.72**	**-618.64**	**2%**
Test 2b	**84.14**	**18.67**	**-23.09**	**-489.86**	**1%**
Test 3b	**87.36**	**15.45**	**51.20**	**-30.88**	**0%**
Test 4b	**92.67**	**10.14**	**-22.41**	**43.89**	**0%**
Test 1c	**73.92**	**28.89**	**-7.42**	**-15.02**	**0%**
Test 2c	**77.24**	**25.58**	**-22.80**	**44.83**	**0%**
Test 3c	**80.76**	**22.05**	**-3.03**	**593.73**	**2%**
Test 4c	**86.37**	**16.45**	**46.09**	**1022.94**	**3%**

Full Model

Once all of the parameters have been established the user is ready to utilize the full VCC model with dynamic data. Below is what the spreadsheet for this model looks like.

User Input Values:

Tea	=	**85**	F
Toa	=	**95**	F
Pe=P1=P4	=	**128**	psia
Pc=P2=P3	=	**357**	psia

Predetermined Air Conditioner Parameters:

Refrigerant	=	**R-410A**	
Rated Capacity	=	**36000**	BTU/h
Isentropic Efficiency	=	**85%**	
Compressor Speed	=	**678584**	rad/hr
Charge of System	=	**4.0625**	lbs
Area of Flow Restrictor	=	**0.000021**	ft^2
D_ei	=	**0.0375**	ft
D_ci	=	**0.0375**	ft

Predetermined Parameters:

Evaporator

C_e1	=	**8.0751**	BTU/(hr*ft*F)
C_e2	=	**3.2317**	BTU/(hr*ft*F)
L_E	=	**102.8116**	ft

Condenser

C_c1	=	**19.7044**	BTU/(hr*ft*F)
C_c2	=	**40.3034**	BTU/(hr*ft*F)
C_c3	=	**23.3156**	BTU/(hr*ft*F)
L_C	=	**92.5870**	ft

Compressor

V_k	=	**0.0004**	ft^3
C_k	=	**0.3410**	

Fixed Orifice

C_v	=	**0.6719**	

Unknown Values:

W_in	=	**6691.981**	BTU/h
Q_H	=	**39988.576**	BTU/h
Q_L	=	**33296.595**	BTU/h
m_dot	=	**437.149**	lbm/hr
l_c1	=	**11.124**	ft
l_c2	=	**71.284**	ft
l_c3	=	**10.179**	ft
subcool	=	**8.0**	F
l_e1	=	**80.562**	ft
l_e2	=	**22.250**	ft
superheat	=	**21.877**	F

Mass Parameters:

Gamma_E	=	**0.774**
M_E	=	**1.650**
Gamma_C	=	**0.730**
M_C	=	**2.445**

Thermodynamic Properties:

State 1 (Superheat):

T1	=	**59.62376927**	F	=T4'+Superheat
P1	=	**128**	psia	
h1	=	**187.2854528**	BTU/lbm	
s1	=	**0.442236528**	BTU/lbm*R	
v1	=	**0.513414541**	ft^3/lbm	

State 2 (Superheat): *** Without isentropic efficiency

P2	=	**357**	psia
h2s	=	**200.2974984**	BTU/lbm
s2	=	**0.442236528**	BTU/lbm*R

State 2 (Superheat): *** With isentropic efficiency

T2	=	**163.8700268**	F	
P2	=	**357**	psia	
h2	=	**202.5936834**	BTU/lbm	=h1+((h2s-h1)/ns)
s2	=	**0.442236528**	BTU/lbm*R	
v2	=	**0.204532567**	ft^3/lbm	

State 2' (Saturated Vapor):

T2'	=	**105.3010914**	F	
P2'	=	**357**	psia	
h2'	=	**182.7796676**	BTU/lbm	
s2'	=	**0.412560455**	BTU/lbm*R	
v2'	=	**0.151781881**	ft^3/lbm	

State cr1 (Average Between 2 & 2'):

Tcr1	=	**134.5855591**	F	
Pcr1	=	**357**	psia	
hcr1	=	**192.6866755**	BTU/lbm	
scr1	=	**0.427398491**	BTU/lbm*R	
vcr1	=	**0.178157224**	ft^3/lbm	

State 2'' (Saturated Liquid):

T2''	=	**105.0874139**	F	
P2''	=	**357**	psia	
h2''	=	**115.0796747**	BTU/lbm	
s2''	=	**0.292607901**	BTU/lbm*R	
v2''	=	**0.016493974**	ft^3/lbm	

State 3 (Subcool):

T3	=	**97.0874139**	F	=T2''-subcool
P3	=	**357**	psia	
h3	=	**111.1178821**	BTU/lbm	
s3	=	**0.280574173**	BTU/lbm*R	
v3	=	**0.01525038**	ft^3/lbm	

State cr3 (Average between 2'' & 3):

Tcr3	=	**101.0874139**	F	
Pcr3	=	**357**	psia	
hcr3	=	**113.0987784**	BTU/lbm	
scr3	=	**0.286591037**	BTU/lbm*R	
vcr3	=	**0.015872177**	ft^3/lbm	

State 4 (Vapor/Liquid Mixture):

T4	=	**37.64851136**	F	
P4	=	**128**	psia	
h4	=	**111.1178821**	BTU/lbm	=h3
s4	=	**0.28938929**	BTU/lbm*R	

v4	=	**0.1276513**	ft^3/lbm
x4	=	**0.246418482**	

v4g	=	**0.475687**	ft^3/lbm
v4f	=	**0.013845**	ft^3/lbm

State 4' (Saturated Vapor):

T4'	=	**37.746952**	F
P4'	=	**128**	psia
h4'	=	**181.572704**	BTU/lbm
s4'	=	**0.43118651**	BTU/lbm*R
v4'	=	**0.4756870**	ft^3/lbm

State er2 (Average between 4' & 1):

Ter2	=	**48.685361**	F
Per2	=	**128**	psia
her2	=	**184.42907**	BTU/lbm
ser2	=	**0.4367115**	BTU/lbm*R
ver2	=	**0.4945508**	ft^3/lbm

Equations:

Sum of % Total Errors: **3.6%**

Evaporator:

Evaporator Error: **0.4%**

Two-Phase Flow

$$0 = \dot{m}(h_{4'} - h_4) - U_{e1} * l_{e1} * (T_{ea} - T_4)$$

-4.784783

Superheated Flow

$$0 = \dot{m}(h_1 - h_{4'}) - U_{e2} * (L_E - l_{e1}) * (T_{ea} - T_{er2})$$

-113.8592

Condenser:

Condenser Error: **0.8%**

Superheated Flow

$$0 = \dot{m}(h_2 - h_{2'}) - U_{c1} * l_{c1} * (T_{cr1} - T_{oa})$$

-15.00936

Two-Phase Flow

$$0 = \dot{m}(h_{2'} - h_{2''}) - U_{c2} * l_{c2} * (T_{2'} - T_{oa})$$

-0.009162

Subcooled Flow

$$0 = \dot{m}(h_{2''} - h_3) - U_{c3} * (L_C - l_{c1} - l_{c2}) * (T_{cr3} - T_{oa})$$

287.17644

Mass Balance: **Mass Error:** **-0.8%**

$$0 = M_{total} - [M_E + M_C]$$

-0.032228

Compressor: **Compressor Error:** **0.0%**

$$0 = \dot{m} - \omega \Upsilon_k \frac{1}{v_1}\left[1 + C_k - C_k \left(\frac{P_C}{P_E}\right)^{\frac{1}{2}}\right]$$

0.0899478

Flow Restrictor: **Orifice Error:** **1.7%**

$$0 = \dot{m} - 3{,}600 * C_v A_v \sqrt{\frac{1}{v_3} * [(P_C - P_E) * 32.174 * 144]}$$

7.3317752