

ABSTRACT

Title of Dissertation: AN OBJECT ORIENTED SIMULATION
FRAMEWORK FOR STEADY-STATE ANALYSIS
OF VAPOR COMPRESSION REFRIGERATION
SYSTEMS AND COMPONENTS

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Thermo fluid energy system simulation has shown to be a useful tool for engineers; encompassing component design, system design and with increasing interest, system optimization. Thermo fluid energy systems, be they for comfort cooling, comfort heating, power generation, or any other purpose typically possess a unique composition and function. This has resulted in simulations for individual

rather narrowly defined energy systems, each customized for the particular system of interest. However, it is impossible to ignore that the majority of thermo fluid energy systems share, among others, the common characteristics of fluid flow, mechanical work input/output and energy input/output via heat transfer. This dissertation exploits this similarity, and develops an object oriented methodology for modeling components and solving systems created from such components, operating in steady-state. The technique is novel in that it discriminates between systems, and their sub-systems, referred to as components. This methodology serves as a functional starting point which will appeal to the objectives of individual research groups, such as industrial sponsors, academic professionals, and students. The dissertation then presents several examples highlighting the major points in the analysis, and a complex example that demonstrates where such a tool may be usefulness in a product design environment. Lastly, the dissertation presents a component based, user-friendly interface specifically for vapor compression refrigeration systems. Several examples are used to validate the component models, reproducing experimental data reasonably well within a range of 5% for most performance variables.

AN OBJECT ORIENTED SIMULATION FRAMEWORK FOR STEADY-
STATE ANALYSIS OF VAPOR COMPRESSION REFRIGERATION SYSTEMS
AND COMPONENTS

By

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Dissertation submitted to the Faculty of the Graduate School of the
University of Maryland, College Park, in partial fulfillment
Of the requirements for the degree of
Doctor of Philosophy
2006

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Dedication

This work is dedicated to my very special friends Glenn Campopiano and Adam Carter. Rest peacefully; knowing that regardless of my state of mind, you are always there to assure me that I *can* do this. Understand that even though you could not stick around to see it done, you have been here every step of the way.

Acknowledgements

I would like to acknowledge Dr. Reinhard Radermacher, and the entire Mechanical Engineering Department at the University of Maryland College Park.

I first thank Dr. Radermacher first and foremost, for affording me the opportunity to pursue my studies at the University of Maryland, and for his infinite patience as I found my way through the process.

I also thank the other members of my dissertation committee, Dr. Azarm, Dr. Hwang, Dr. Jackson and Dr. Rabin.

I would like to thank Dr. Ugo Piomelli, for listening to me above and beyond the call of Fluid Mechanics, and for lending advice of a personal nature.

Lastly, I would like to thank Ronnie Brown, who helped me find my way through a very dark place.

Nomenclature

Symbols

A	Cross-Sectional / Surface Area	m ²
C	Heat Capacity Rate	W/K
c*	Heat Capacity Rate Ratio	-
CF	Correction Factor	-
COP	Coefficient of Performance	-
C _p	Specific Heat	J/kg-K
DP _{Factor}	Pressure Drop Correction Factor	-
d	Diameter	m
\dot{E}	Energy Flow	W
E _v	Viscous Dissipation	W
e	Electrical Charge	C
\vec{F}	Force	N
FR	Fin Ratio	-
f	Friction Factor	-
g	Gravitational Acceleration	m/s ²
G	Gibbs Free Energy	W
h	Specific Enthalpy	J/kg
h	Heat Transfer Coefficient	W/m ² -K
k	Thermal Conductivity	W/m-K
L	Length	m
m	Mass	Kg
\dot{m}	Mass Flow Rate	kg/s
μ	Viscosity	Pa-s
μ	Chemical Potential	J
NTU	Number of Transfer Units	-
Nu	Nusselt Number	-
P	Property	-
p,P	Pressure	Pa, Psia
P0	Environmental Pressure	Pa, Psia
PLF	Pseudo Length Factor	-
\bar{Q}	Characteristic Energy Flow Rate	W
Q, \dot{Q}	Heat Transfer/ Heat Transfer Rate	J/W
r	residual	W, kg/s
R	Thermal Resistance	m ² -K/W
Re	Reynold's Number	-
s	Specific Entropy	J/kg-K
S	Surface Area	m ²

SEER	Seasonal Energy Efficiency Ratio	Btu/W-hr
SH	Superheat	K
T	Temperature	K, °C, °F
T0	Environmental Temperature	K
u	Specific Internal Energy	J/kg
U	Overall Heat Transfer Coefficient	W/m ² -K
V	Electrical Potential	V
V	Volume	m ³
\bar{v}	Velocity	m/s
W, \dot{W}	Work/Power	J/W
x	Thermodynamic Quality	-
z	Elevation	m

Greek Letters

δ	Deviation	W, kg/s
Δ	Change	-
ε	Relative Roughness	m
ε	Convergence Criterion	W, kg/s
ε	Effectiveness	-
η _s	Isentropic Efficiency	-
η _v	Volumetric Efficiency	-
μ	Viscosity	Pa-s
μ	Chemical Potential	J
Φ	Gravitational Potential Energy	J
ω	Humidity Ratio	g/kg

Subscripts/Superscripts

1,2	Weak/Strong Fluid
Air	Air
Cond	Condenser
Des	Desuperheater
i/o	In/Out
J	Junction
LHT	Liquid Heat Transfer
PD	Pressure Drop
TPHT	Two-Phase Heat Transfer
p	Port
ref	Refrigerant
Sub	Subcooler
t	Transported
*	Characteristic Value

Table of Contents

Dedication	ii
Acknowledgements.....	iii
Nomenclature	iv
Table of Contents.....	v
List of Tables	xii
List of Figures	xiv
1. Introduction.....	1
1.1 Vapor Compression Refrigeration and Simulation and Research Motivation	1
1.2 Literature Review.....	2
1.2.1 Generalized Simulation.....	2
1.2.2 Air-Conditioning/Refrigeration Simulation.....	4
1.2.3 Existing Software Packages.....	10
1.2.4 Summary	11
1.3 Research Objective	11
1.4 References.....	13
2. General Thermo-Fluid Simulation Structure	18
2.1 Introduction.....	18
2.2 A Note on Thermo Fluid Properties.....	19
2.3 Conservation Equations	19
2.4 Simulation Data Structures	20
2.4.1 Selection of Independent Variables	20
2.4.2 Components	22

2.4.3	Ports	23
2.4.4	Junctions	23
2.4.5	Junction Evaluation.....	24
2.5	Constructing a Thermo Fluid System	25
2.6	Controlling the System	29
2.7	Simulation Hierarchy	31
2.8	Simulation Framework Design Rules	31
2.9	Summary	32
2.10	References.....	33
3.	The Junction Solver	34
3.1	The Junction Component Port Matrix (JCP)	35
3.2	Component Specification.....	37
3.3	Independent Variables and Residual Equations.....	37
3.4	Solution of the Model	39
3.4.1	Simultaneous Approach	39
3.4.2	Fractional Step	40
3.4.3	Solving Hydraulic Relations	44
3.4.4	Solving Thermal Relations	44
3.5	Conditioning of the Model.....	44
3.6	The Solution Seed.....	46
3.7	Summary	46
3.8	References.....	47
4.	Simulation of a Heat Exchanger	49

4.1	Model Components.....	49
4.1.1	Tubes.....	49
4.1.1.1	Hydraulic Relations	51
4.1.1.2	Thermal Relations.....	52
4.1.1.3	Refrigerant Side Heat Transfer Coefficients.....	53
4.1.1.4	Air Side Heat Transfer Coefficients	54
4.1.1.5	Air Surface Area Calculations	55
4.1.1.6	Distributed Models.....	55
4.1.2	A Refrigerant Circuit	56
4.1.3	An Air Circuit	56
4.2	The Actual Heat Exchanger	58
4.3	The Heat Exchanger Model	60
4.3.1	Wrapping the Junction Solver.....	63
4.3.2	Seeding the Junction Solver for the Heat Exchanger.....	64
4.3.3	Scaling Parameters for the Model.....	64
4.4	Validation.....	67
4.4.1	Thermophysical Property Distributions.....	67
4.4.2	Mass Balances and Energy Balances	69
4.5	Verification	71
4.6	Summary	75
4.7	References.....	76
5.	VapCyc – Vapor Compression Refrigeration System	77
5.1	VapCyc Component Specifications	78

5.2	The Test System.....	80
5.3	Simulation of Vapor Compression Refrigeration	81
5.4	Premise.....	82
5.5	VapCyc’s Component Library	83
5.5.1	A Generic Compressor Model	83
5.5.2	The Alpha Compressor	84
5.5.3	The Beta and Gamma Compressor	86
5.5.4	The Delta Compressor	86
5.5.5	ARI 10 COEFFICIENT MODEL.....	87
5.5.6	The Generic Orifice	87
5.5.7	Generic Heat Exchanger	88
5.5.8	The Alpha Heat Exchangers	90
5.5.9	The Eta and Epsilon Condensers	91
5.5.10	Heat Exchanger Summary	93
5.6	VapCyc Utility	93
5.6.1	Charge Management	94
5.6.2	Component Inter-Changeability.....	94
5.6.3	Multiple Refrigerants	94
5.6.4	Multiple Unit Systems	95
5.6.5	Physical Variables.....	95
5.6.6	Economic Variables.....	95
5.6.7	Seasonal Performance.....	96
5.6.8	Interactive Mode	96

5.6.9	Automated Mode: Parametric Analysis.....	98
5.7	VapCyc System Model	98
5.7.1	Scaling the Independent Variables and Balance Equations.....	98
5.7.1.1	The System Context.....	100
5.7.1.2	The VapCyc Air-Conditioning Context.....	105
5.7.1.3	The Multiple VapCyc Contexts	106
5.7.1.4	Scaling With the Context.....	108
5.7.2	Control of VapCyc.....	108
5.8	VapCyc Examples of Capabilities and Verification.....	109
5.8.1	Verification of the Model.....	110
5.8.2	Verification of the Scaling.....	112
5.8.3	Examining the VapCyc Context	114
5.8.4	Comparison with Test Data	117
5.8.5	Optimization of Charge.....	122
5.8.6	SEER Example.....	124
5.8.7	Application Examples: Expandability of the Simple System.....	125
5.8.7.1	Evaporators in Parallel.....	125
5.8.7.2	Compressors in Series.....	127
5.8.8	Parametric Study of COP With Respect to Charge and Air Velocity.....	129
5.9	Summary.....	130
5.10	References.....	133
6.	Conclusions.....	135
6.1	Thermal Energy System Simulation Framework.....	135

6.2	Basic Components	136
6.3	Component Models.....	136
6.4	Compound Energy Systems.....	137
6.5	VapCyc	138
6.6	Future Work.....	141
6.7	Major Accomplishments.....	141
6.8	Major Engineering Accomplishments	143
A1.	TEST System	147
A1.1	The TEST Compressor Model.....	147
A1.2	The TEST Orifice Model.....	148
A1.3	The TEST Evaporator Model.....	149
A1.4	The TEST Condenser Model	151
A1.5	Raw Data.....	152
A1.6	References.....	156
A2.	VapCyc – The Program	157
A2.1	Basic Description.....	157
A2.2	The Interface	158
A2.2.1	Choosing Components.....	160
A2.2.2	Editing Component Independent Variables	161
A2.2.3	The Parametric Interface.....	162
A2.3	The Solution Routine	163
A2.3.4	The Pre Processor Phase.....	164
A2.3.5	The Solution Phase	164

A2.3.6 The Post Processor Phase..... 164

List of Tables

Table 2-1: Equations used to Model Generalized Thermo Fluid Energy Systems.....	28
Table 2-2: Equations used to Model Generalized Thermo Fluid Energy Systems with Negligible Kinetic and Potential Energies.....	29
Table 3-1: JCP for a Vapor Compression Refrigeration System.....	36
Table 3-2: A List of Necessary Properties for all Components Utilized in the Junction Solver.....	37
Table 4-1: Test Data for R-404a Condenser.....	59
Table 4-2: Test Data for R-290 Condenser.....	60
Table 4-3: Correction Factors for Modeled Heat Exchanger (R-404a).....	62
Table 4-4: Correction Factors for Modeled Heat Exchanger (R-290).....	62
Table 5-1: VapCyc Component Properties.....	79
Table 5-2: Alpha Compressors Rated For Air-Conditioning.....	85
Table 5-3: Alpha Compressors Rated For Low-Temperature Refrigeration.....	85
Table 5-4: Eta Condenser Nominal Operating Point for Air-Conditioning.....	91
Table 5-5: A Summary of VapCyc Heat Exchangers.....	93
Table 5-6: VapCyc Evaporating and Condensing Temperatures.....	107
Table 5-7: Scaled Independent Variables and Balance Equations.....	109
Table 5-8: Mass and Energy Balances for Test Configurations.....	110
Table 5-9: Air Temperatures of A/C and Refrigeration Context.....	115
Table 5-10: Measured vs. Simulated Data for R-290 Test System.....	121
Table 5-11: Measured vs. Simulated Data for R-404a Test System.....	122
Table 5-12: Alpha 3600 System Performance at ASHRAE "A" and "B" Conditions...	124

Table 5-13: Results for Nominal 20 kW Alpha Split Air Conditioning System 127

List of Figures

Figure 2-1: Energy System Components	22
Figure 2-2: A Junction	25
Figure 2-3: Open (a) and Closed (b) Energy Systems	26
Figure 3-1: A Modified Vapor Compression Refrigeration System.....	36
Figure 3-2: Flow Chart for Simultaneous Equations	41
Figure 3-3: Fractional Step Method of Jiang (2003)	42
Figure 3-4: A Modified Fractional Step for the Junction Solver	43
Figure 3-5: Successive Substitution Energy Routine.....	44
Figure 4-1: A Refrigerant Circuit	56
Figure 4-2: An Air Stream: Air Flows from Left to Right, Through Three Air Segments, Simulating Air Flow Over Three Tube Segments	57
Figure 4-3: Cross Section of Heat Exchanger Configuration. Air Flows Across Banks of Tubes from Left to Right	58
Figure 4-4: Heat Exchanger Physical Parameters.....	59
Figure 4-5: Geometry Modification For Heat Exchanger Model. The Real Condenser on the Left is Approximated with Circuitry on Right.....	61
Figure 4-6: Heat Exchanger Solution Flow Chart	63
Figure 4-7: Algorithm for a Linear Pressure Distribution	64
Figure 4-8: Condition Number of Jacobian Matrix versus Hydraulic (Inner) Iteration Number. The Iterative Hydraulic Routine is Wrapped in a Successively Substituted Outer Thermal Routine. Scaling of the Independent Variables and Residual	

Equations Reduces the Condition Number of the Jacobian Matrix of the Hydraulic Routine.....	66
Figure 4-9: Refrigerant (a) Pressure and (b) Quality Distribution. Air Flow, Not Shown In the Figure, Flows from Left To Right Over the Tube Banks, Cooling Them.	69
Figure 4-10: Heat Exchanger Temperature Distribution –	70
Figure 4-11: Heat Exchanger (a) Mass Balance and (b) Energy Balance	70
Figure 4-12: Mass Flow Rate Comparison for R-404a Condenser (See Table 4-1).....	72
Figure 4-13: Heat of Rejection Comparison for R-404a Condenser (See Table 4-1)	73
Figure 4-14: Subcooling Comparison for R-404a Condenser (See Table 4-1)	73
Figure 4-15: Mass Flow Rate Comparison for R-290 Condenser (See Table 4-2)	74
Figure 4-16: Heat of Rejection Comparison for R-290 Condenser (See Table 4-2)	74
Figure 4-17: Subcooling Comparison for R-290 Condenser (See Table 4-2)	75
Figure 5-1: η_s versus Compression Ratio for Delta Compressor.....	86
Figure 5-2: The Alpha 3600 System in Air-Conditioning Mode.....	97
Figure 5-3: Changing of the Compressor.....	97
Figure 5-4: Simple Vapor-Compression Refrigeration System.....	99
Figure 5-5: A Compressor Model Number (Tecumseh, 2004).....	100
Figure 5-6: ARI 540-99 Rating Points for Compressors	101
Figure 5-7: ARI 540-99 Heating and Cooling Compressor Operating Points.....	102
Figure 5-8: Performance Summary of a Copeland CS10K6E-PFV Compressor (Copeland Web Site)	103
Figure 5-9: Performance Summary of a Copeland ARJ11K1E-PFV Compressor.....	104
Figure 5-10: A Nominal Air-Conditioning Operating Point for VapCyc.....	106

Figure 5-11: A Nominal Air-Conditioning Operating Point for VapCyc.....	107
Figure 5-12: P-h Diagrams for Various Alpha 3600 Configurations	111
Figure 5-13: P-h Diagrams of Numerical Experiment Systems	113
Figure 5-14: Jacobian Condition Number vs. Iteration Number	113
Figure 5-15: Jacobian Condition Number vs. Iteration Number	114
Figure 5-16: Performance of Alpha3600 Components:.....	115
Figure 5-17: Low Temperature Solution to an Alpha Hybrid System.....	116
Figure 5-18: Simulated vs. Measured Evaporator Heat Load.....	118
Figure 5-19: Simulated vs. Measured COP	118
Figure 5-20: Simulated vs. Measured Mass Flow Rate	119
Figure 5-21: Simulated vs. Measured Refrigerant Charge	120
Figure 5-22: Suction and Discharge Pressure for R290 Test System.....	120
Figure 5-23: Measured and Simulated COP vs. Condenser Sub-cooling	123
Figure 5-24: ASHRAE "A" and "B" Rating Conditions for Air-Conditioning Systems	124
Figure 5-25: Alpha 3600 System SEER vs. System Charge	125
Figure 5-26: A Nominal 20 kW Alpha Split Air Conditioning System	126
Figure 5-27: System COP versus Ambient Temperature	129
Figure 5-28: COP and Capacity versus Charge and Air Velocity	130

1. Introduction

1.1 Vapor Compression Refrigeration and Simulation and Research

Motivation

In the global economy of the twenty first century, energy production, distribution and conversion are of paramount importance. Environmental acts, such as the Montreal Protocol and the Kyoto Protocol have demonstrated a global initiative toward protecting the environment from man made products which may cause damage. A large part of the world's energy usage relates to heating, ventilation, air-conditioning and refrigeration (HVAC/R) systems. Consequently, research is motivated to meeting simulation needs for component manufacturers, system manufacturers and researchers.

A simulation framework, catered toward vapor compression refrigeration and its components is a useful tool for research. Such a Framework that is general in nature would be very helpful. Ideally, the framework facilitates the development and use of libraries of individual component models that integrate into systems seamlessly, and without a priori knowledge of the system to which they are being inserted.

The generality of the vapor compression refrigeration framework allows future generations to utilize it without specific knowledge of the inner workings of the framework. This prevents generation after generation of re-invention of the simulation structure, which hampers the development of the research of individual components and/or system configurations.

1.2 Literature Review

1.2.1 Generalized Simulation

A variety of engineering problems encountered in practice exhibit behavior that is predicated upon the interaction of complex physical phenomena, i.e. their response is determined not by a single phenomena, but by the multidisciplinary interaction (Gupta and Meek, 2000). Considering individual components as a ‘discipline’, combines different components and estimating their performance as a system is multidisciplinary in nature.

Object oriented simulation allows for increased programming productivity in code maintenance and re-usability. A single generalized simulation technique can be used to simulate many different individual energy systems, which can be assembled into coordinating multidisciplinary energy systems. Applying object oriented simulation to the standard vapor compression refrigeration system, each individual system component; compressor, condenser, expansion device and evaporator can be modeled independently from the coordinating system simulation tool which will integrate them to a complete system.

This approach is beneficial, as modular component models which can utilize a majority of the same code. A secondary benefit is realized when a generalized solution routine is used within a coordinating system model.

Along with these object oriented techniques comes the potential benefit of components conforming to a single standard for their use in the generalized simulation; namely an industry wide component library. This industry wide component library

coupled with a simulation tool to utilize can then estimate the performance of systems comprised of these individual components.

Multidisciplinary optimization, or MDO, (Alexandrov and Hussaini, 1997) may be defined as a methodology together with a set of tools for assisting in the design of complex coupled systems, that is systems whose behavior is governed by many distinct but interacting physical phenomena. An object oriented, simulation tool for thermo fluid energy systems is a logical foundation for an industry wide analysis and optimization tool which may be used for MDO.

Although moderate work has been performed in generalized energy system simulation, little has focused in on vapor compression refrigeration. In 1994, Paulus et al. (1994) describes a technique using vectors and matrices to specify a system. The technique is lacking because it fails to discriminate between the component equations, and system property balances. This in effect creates a single system of equations which must be solved simultaneously. Performance relations are hard coded into the program, and there is no flexibility to change components.

A generalized approach (Majumdar et.al, 1995), initially developed to simulate the fluid flow systems found in rocket propulsion became a very comprehensive program for solving generalized flow networks. The interface for the program was somewhat cumbersome, in that the user interacts with the program through many input and output files, as well as through an interactive mode. As with Paulus, the program is configured for a single system. A change in make-up of the system is achieved through the input of a different set of input files, or an entirely new run in the interactive mode. The use of

input/output files, coupled with component models which are integrated into the simulation itself are limiting factors in achieving a flexible modeling environment.

In 2000, Lindsay (2000) offers “Network Builder”, which is a foundation for the generalized energy system simulation technique given here. The formulation is based on a series of components connected by junctions, and serves as a starting point for this work.

Jiang (2003) creates an object oriented heat exchanger simulation, based upon Lindsay’s techniques. The simulation is comprehensive and flexible, allowing many parameters to be specified via a user interface. In this simulation, tubes act as individual components, which can be combined into different configurations by the user. Although general in nature, the only components which could be changed were tubes.

1.2.2 Air-Conditioning/Refrigeration Simulation

The simulation of the vapor compression refrigeration system is clearly a benefit to the engineer, and it is therefore not surprising that great deal of work has been done toward creating good simulations. Meaningful simulations of refrigeration systems can be traced back to as early as 1976, when Hiller and Glicksman (1976) as well as Davis and Scott (1976) offered the first modern looks at simulation, which were limited in scope to air-to-air heat pump simulations. Ellison and Creswick in 1978 then used the Hiller and Glicksman model to examine the change in system performance due to changes in components. Also appearing around the same time was a simulation for optimizing the heating function of heat pumps by Carrington (1978). In 1979, Ellison and Rice gave a report on the Oak Ridge heat pump model, a model that continued to evolve over the years, and is indeed still existing and evolving today.

The early 1980's gave birth to many refrigeration simulations. Rice et al. (1981) used computer simulation to explore optimization. This was achieved by combining the Oak Ridge model with an optimization program to optimize system level performance as a function of specific component level variables. Also in the early 1980's were McMullan et al. (1981) used simulation to compare field performance with (laboratory) test performance. Domanski and Didion in 1983 created a model (HPSIM) for the United States National Bureau of Standards (NBS-Now known as the National Institute for Testing and Standards, NIST). The HPSIM model is still in existence today, and has undergone many enhancements over the years which allow it to be very flexible.

Refrigerant mixtures were introduced to vapor-compression models in the mid 1980's. Connon (1984) presents a simple simulation model in 1984, utilizing refrigerant mixtures.

Parise (1986) offers a simulation model that is based upon the governing equations for its component models, rather than performance maps and empirical equations. Parise claims at this point that empirical performance maps are responsible for deviations of simulations from actual equipment behavior.

Welsby et al. (1988) creates a simulation model for water-to-water heat pump systems, and gives a literature review as the starting point, detailing the growth of the field. In 1987, simulation was put to use comparing the performance of pure refrigerants versus mixtures, as given in McLinden and Radermacher (1987).

Simulation literature becomes more prevalent in the 1990's, as the personal computer became affordable enough to be used by more researchers

In 1990, Jolly et al. used simulation to present work on heat pump assisted drying. Also in 1990 the Oak Ridge model had evolved into CYCLEZ, or the Mark III system. Rice and Sand (1990) also present a modified version of CYCLEZ for a Lorentz-Meutzner cycle refrigerator-freezer model. In 1992 Stafanuk et al. present a model specifically addressing the need for a so called charge inventory, or and accounting of the refrigerant mass contained within the system That work addresses the affects of the charge inventory, specifically the effect it has on prediction of off-design performance. Prior to Stafanuk only the Oak Ridge model and the HPSIM (NIST) simulation model accounted for system charge. Bare (1993) presents a model designed to address alternative refrigerants and their performance potential. It is noted that the paper presents a very simple model, as the focus is on refrigerant effect on the system, not accurate simulation of the system itself. The topic of alternate refrigerants is also the focus of a study by Ouazia and Snelson (1994). This is also a simple model, but is typical of system simulations involving alternative refrigerants at this time, which is co-incident with the phase out of CFCs. Around this time the Oak Ridge model had grown to the Mark V, also named PUREZ (Rice et al. 1994).

A “toolkit” for chiller simulation is presented by Bourdouxh et al. (1994), which represents an ASHRAE initiative for simple simulation of HVAC components for building simulation. Levins et al. (1996) used simulation to explore the effects of oversizing of residential air conditioning units. The Oak Ridge Model (Mark II) was modified to reflect performance data from an experimental apparatus, and the results used to estimate energy savings by reducing compressor size in existing units. The DOE/Oak

Ridge model history and overview, as well as some applications to R-22 alternatives were presented in 1997 at a symposium on heat pumps in cold climates (Rice et. al. 1993)

Water chillers are a small part of the vapor compression simulation family, and oft ignored, but in 1998 Richardson as well as Browne and Bansal (1998) address them. The study by Brown and Bansal addresses many sources of simulation literature to date, and concludes that although there is (to that date) a large amount of simulation research in existence, chillers are different in operation and therefore the field of chiller simulation was immature. Browne and Bansal conclude that the two main differences between water chillers and typical vapor compression heat pumps are the heat exchanger configurations and the compressors. It is concluded that simulations that accurately accounted for these differences would be very similar in nature to existing simulations.

Anand and Tandon (1999) discuss two separate works, both of which focus on simulation of refrigeration systems. The works involve using modeling techniques separating the component models from the system models.

LeRoy (2000) uses PUREZ to predict the performance of unitary air conditioners. The concept of “tuning” the PUREZ model is introduced to help the simulation match actual data.

Browne and Bansal (2001) continue their modeling of chillers with an improved heat exchanger model. Updates to the Oak Ridge model (now available for on-line simulation over the internet) can be found from the Oak Ridge web site (Rice 2001).

An indicator of the direction is the move to component based modeling, Grossman et al. (2001) present ABSIM, a simulation of absorption systems, presents a modular format

to simulations. Here Grossman discusses the advantages of using modular simulation components for the sharing of work as well as for development of the tool.

Several key issues emerge from the study of the literature, each of which needs to be included into a comprehensive simulation, namely:

- Charge Inventory
- Architecture and solution technique of the system model
- (Lack of) Optimization

Few simulations to date account for “charge inventory”. Stefanuk et al. clearly presents (1994) that for off-design simulation the charge inventory is mandatory. Consequently, the majority of the existing models are only representative of a single operating point, not a uniquely charged system at different operating points.

The majority of the system models in existence today integrate the components and the system into a single system of equations to be solved. Nearly every cited reference implements this technique.

Many solution techniques exist for the system of equations developed in simulations of vapor compression refrigeration systems. The large majority of the existing simulations use a mixture of iteration and successive substitution. The overwhelmingly popular solution technique involves solving systems of equations in successive substitution technique created by combining the component models with the system conservation laws. Typically satisfying some of the equations solves a “high side” pressure, and the results of the “high side” outputs are then fed to the “low side”, which must in turn satisfy the remaining equations. Results from the “low side” are fed to the “high side”. The process continues until neither the high nor low sides change. A second successive substitution technique (Annand, Tandon 1999) involves separate models for

the components. Utilizing the charge inventory, a single component can be solved for its operating point and inlet and outlet flows (given an initial guess for its independent variables and charge). Any discrepancy between inlet and outlet flow results in a net change in charge for the component. This charge difference is passed along to the next component, along with the thermodynamic state of the fluid entering the component. This process is repeated until there is no change in the system, i.e. there are no discrepancies in charge or between mass flows into or out of any components. This process is difficult to implement in complex systems (components in parallel for example) and requires fundamental modeling differences between components that store significant amounts of refrigerant and those that do not. Another popular technique is to formulate the equations and solve them using an iterative solver such as Newton-Raphson (Jolly et al. 1990, Paulus et al. 1994, Bourdoux et al. 1994, Richardson 1998, Brown and Bansal 1998, LeRoy et al. 2000).

Some attempts to optimize refrigeration systems as a whole, have been made (Rice et al. 1981, Fisher and Rice 1986, Levins et al. 1996). However, in all of these cases the system simulation model is typically customized to begin with and then is either (a) integrated with the optimization routine, making the program even more customized, or (b) the optimization is performed manually with simulation program results.

Object oriented simulation allows more flexibility to customize individual components, which are separated from the solution routines, which themselves are separated from optimization routines.

1.2.3 Existing Software Packages

Many software packages currently exist, which may be used for energy system simulation. Refrigeration systems definitely fit within the scope of these simulation packages. The current software comes in two groups and hybridizations of the groups. One group is general equation solvers, which allow the user to specify the system model in terms of governing equations, and the software solves the set of equations. A second group is advanced energy system software.

Clearly general equation solvers are valid for any conceivable energy system and therefore refrigeration systems. However, the equation solvers are extremely dependent upon the user to formulate the model and properly specify it, as the software has no intelligence with regard to the specific problem. Examples of general equation solvers (or programming environments conducive to general equation solving) are: *The Engineering Equation Solver* (E-Chart), *MATLAB* (Mathworks), and *Mathematica* (Wolfram).

Advanced energy system software on the other hand is often very specific to the task at hand (determining the operation of a refrigeration system, steam power plant, gas turbine power plant, etc). Often the user is “locked” into the specific system, and modification or generalization of the system is difficult or impossible. Examples of advanced energy system software are: *Aspen Plus* (Aspen Technologies) and *Gate Cycle* (Stork).

Hybridizations of the two groups offer fixed systems, but user defined component models. An example of hybridization is *Sinda/Fluint* (C&R).

Each of the three classes of software has advantages and drawbacks, often leading a user to create their own custom software to specifically suit their individual needs.

1.2.4 Summary

Many simulations for refrigeration systems exist, and there are many software packages that exist to facilitate custom system models. It is clear from the literature, that most often, simulations are created to fill a specific need, rather than to serve as a general tool. The main output of each simulation is the performance of the specific system being simulated. A drawback of previous simulations is the rigid definition of the system. Often existing simulations offer generality through a wide range of component independent variables. For the most part, component models themselves are integrated into the simulation itself and therefore not subject to any change not built into the simulation at its creation.

Two main challenges are inherent in current modeling of vapor compression in regard to creating generalized modeling tools, namely (1) a fixed systems configuration and (2) specific, usually fixed, component configurations.

1.3 Research Objective

The findings from the literature demonstrate that although the fields of energy system simulation in general, and vapor compression refrigeration specifically, are mature, many opportunities for research and progress exist today.

A significant engineering challenge is the creation of a general vapor compression refrigeration system simulation tool, which can serve many individual constituent objectives. This tool best serves the individual user by allowing all the detail required by

an individual constituent, but does not force a complete understanding of the inner workings of disciplines beyond the scope of this individual constituent group.

A true general-purpose vapor compression simulation program includes all decided features in a set defined by components, refrigerants, refrigerant charge, system boundary conditions and controls.

The primary objective of the dissertation is the development of a general-purpose framework for simulating fluid flow systems, such as those found in refrigeration and air-conditioning applications. The cornerstone of the general-purpose nature is the prediction of the performance of the complete set of components, working fluids, working fluid charge, system boundary conditions and controls, while each component of the system has its own independent, stand-alone simulation model.

This thesis accomplishes the following objectives:

1. Develop and create a simulation framework, and subsequent simulation tool in an object oriented manner, which provides the ability to simulate experiments in a laboratory. The framework is developed with the following fundamental objectives:
 - a. Serve as a functional starting point which will appeal to the individual objectives of each individual user (industrial sponsors (development of models and/or real equipment), academic professionals (utility of the simulation tool without an intimate understanding of its inner workings), and students (re-using of existing software to facilitate the growth of the project(s)))

- b. Serve as a simulation tool such that each component within the system is a stand-alone simulation model, enabling the analysis and if desired optimization of individual simulation models and/or the system as a whole
2. Verify the simulation tool with experimental data, specifically a condenser, and air conditioning system utilizing this condenser.

1.4 References

Alexandrov, N., Hussaini, M. Y., 1997, *Multidisciplinary Design Optimization State of the Art*, Society for Industrial and Applied Mathematics, 1997

Anand, G., 1999, "Transient and Steady State Model of a Household Refrigerator", Masters Thesis, Department of Mechanical Engineering, University of Maryland, 1999

Bare, J.C., 1993. "Simulation Of Performance Of Chlorine-Free Fluorinated Ethers And Flourinated Hydrocarbons To Replace CFC-11 And CFC-114 In Chillers", *ASHRAE Transactions* 99 (1): pp 397-407

Bourdouxh, J.H., Grodent, M, Lebrun, J.J, Saavedra, C, Silva, K, 1994, "A Toolkit For Primary HVAC System Energy Calculation-Part 2: Reciprocating Chiller Models". *ASHRAE Transactions* 100 (2): pp. 774-786

Browne, M., Bansal, P., 1998, "Steady-State Model Of Centrifugal Liquid Chillers", *International Journal of Refrigeration*, 1998, Vol. 21, pp. 343-358

Browne, M, Bansal, P., 1998, "Challenges in Modeling Vapor-Compression Liquid Chillers", *ASHRAE Transactions*, 1998, V. 104, Pt. 1

Browne, M., Bansal, P., 2001, "An elemental NTU- ϵ model for vapour-compression liquid chillers", *International Journal of Refrigeration*, 2001, Vol. 24, pp. 612-627

Carrington, C.G., 1978, "Optimizing a heat pump for heating purposes", *International Journal of Energy Research*, Vol. 2, pp. 153-170

CEEE web site, http://www.eng.umd.edu/Department_gateways/CEEE.html

Clark, D. R. and May, W. R, Jr., 1985, HVACSIM+ Building Systems and Equipment Simulation Program Users Guide. NBSIR 85-3243. National Bureau of Standards, September, 1985.

Davis, G.L., Scott, T.C., 1976, "Component Modeling Requirements For Refrigeration Systems Simulation", *Proceedings of the 1976 Purdue Compressor Technology Conference*, West Lafayette, USA, July 1976, pp. 401-408

Domanski, P., Didion, D., 1983. "Computer Modeling of the Vapor Compression Cycle with Constant Flow Area Expansion Device", *NBS Building Science Series 155*, May 1983

Ellison, R.D. and Creswick, F.A. 1978. "A Computer Simulation Of Steady-State Performance Of Air-To-Air Heat Pumps", *Report ORNL/CON-16*, Department of Energy, Oak Ridge National Laboratory, Tennessee, USA

Ellison, R.D. and Rice, C.K. 1979. "ORNL Heat Pump Model Update – May 1979", Department of Energy, Oak Ridge National Laboratory, Tennessee, USA

Glicksman, L.R., 1976, "Improving Heat-Pump Performance Via Compressor Capacity Control – Analysis And Test". Volumes I and II, *Energy Laboratory Reports MIT-EL 76-0001 and MIT-EL 76-0002*, Department of Mechanical Engineering, Massachusetts Institute of Technology, Massachusetts, USA (January 1976)

Gupta, K.K., Meek, J.L., 2000, *Finite Element Multidisciplinary Analysis*, AIAA Education Series, American Institute of Aeronautics and Astronautics, Inc, 2000

Grossman, G., Zaltash, A., 2001, "ABSIM – Modular Simulation Of Advanced Absorption Systems", *International Journal of Refrigeration*, 2001, Vol. 24, pp 531-543

Jiang, H., 2003, *Development Of A Simulation And Optimization Tool For Heat Exchanger Design*, Ph.D. Thesis, Department of Mechanical Engineering, University of Maryland, 2003

Jolly, P., Jia, X. and Clements, S., 1990 "Heat Pump Assisted Continuous Drying Part I: Simulation Model", *International Journal of Energy Research*, 1990, Vol 14, pp. 757-770

LeRoy, J, Groll, E., Braun, J, 2000, "Evaluating The Accuracy Of PUREZ In Predicting Unitary Equipment Performance", *ASHRAE Transactions*, 106 (1)

Levins, W., Rice, C.K., Baxter, V.D. 1996, "Modeled And Measured Effects Of Compressor Downsizing In An Existing Air Conditioner/Heat Pump In The Cooling Mode", *ASHRAE Transactions*, V 102, Pt 2

Lindsay, D., 2000, "Network Builder Notes, University of Maryland – CEEE, Internal Document, July, 2000

- Paulus, D. Jr., Gaggioli, R., Park, H. and Dunbar, W., 1994, "Development of Personal Computer Software for Energy System Simulation", *Thermodynamics and the Design, Analysis and Improvement of Energy Systems*, Advanced Energy Systems Division (Publication) AES-Vol. 33, Proceedings of the 1994 International Mechanical Engineering Congress and Exposition, p. 177-184
- Majumdar, A. K., Bailey, J. W., Schallhorn, P. A. and Steadman, T., 1999, "Generalized Fluid System Simulation Program (GFSSP) Version 3.0" Sverdrup Technology MSFC Group Report No. MG-99-290, Nov, 1999, MSFC Science and Engineering Contract NAS 8-40836.
- McLinden, M.O. and Radermacher, R., 1987. "Methods for Comparing the Performance of Pure and Mixed Refrigerants in the Vapor Compression Cycle", *International Journal of Refrigeration*, 10, pp. 318-325
- McMullan, J.T., Morgan, R., Hughes, D.W., 1981. "The Discrepancy Between Heat-Pump Field And Test Performance: A Simulation Study", *International Journal of Energy Research*, Vol. 5, pp. 83-94
- Metcalf, R.R., 1989, Development of a Generalized Fan System Simulation for the Building Loads Analysis and System Thermodynamics Program (BLAST), Masters Thesis, University of Illinois, 1989
- Ouazia, B., Snelson, P., 1994. "Predicting System Performance Of Alternative Refrigerants Using A Water-Water Heat Pump". *ASHRAE Transactions*, 100(2): pp. 140-147
- Parise, J., 1986. "Simulation of Vapour-Compression Heat Pumps" *Simulation* 46(2): pp. 71-76
- Rice, C.K., 2001, *ORNL-HPDM-MarkV-VI-Update.pps*, Power Point Presentation on Oak Ridge Heat Pump Design Model for the web.
<http://www.ornl.gov/~wlj/hpdm/doehpdm.html>, January, 2001
- Rice, C.K. and Jackson, W.L., 1994. "PUREZ—The Mark V ORNL Heat Pump Design Model for Chlorine-Free Pure and Near-Azeotropic Refrigerant Alternatives", Document Package, Oak Ridge National Laboratory
- Rice, C.K., Sand, J.R., 1990. "Initial Parametric Results Using CYCLEZ—An LMTD Specified Lorentz-Meutzner Cycle Refrigerator-Freezer Model", West Lafayette, PA: *Proc. International Refrigeration Conference at Purdue University, July 17-20*. pp. 448-458.
- Richardson, D. H., 1998. "Performance Models For Use in Computer Simulation of Energy Systems", Masters Thesis, Department of Mechanical Engineering, Marquette University, 1998

Resource Dynamics Corporation, 2002, "Integrated Energy Systems (IES) for Buildings: A Market Assessment", Prepared for: Energy Efficiency and Renewable Energy U.S. Department of Energy, Washington, DC, and Oak Ridge National Laboratory Oak Ridge, TN, August 2002

Rice, C.K., Fischer, S. K., Ellison, R.D., Jackson, W.L. 1981. "Design Optimization of Conventional Heat Pumps: Application to Steady-State Heating Efficiency," *ASHRAE Transactions*. Vol. 87, pp. 1037-1054

Rice, C.K., Wright, L. and Bansal, P.K., 1993, "Thermodynamic Cycle Evaluation Model For R-22 Alternatives In Heat Pumps-Initial Results And Comparisons". Moneton, Canada: Proc. 2nd Int. Conference on Heat Pumps in Cold Climates, Aug 16-17

Stefanuk, N.B.M., Aplevich, J.D., Renksizbulut, 1992. "Modeling And Simulation Of A Superheat-Controlled Water-To-Water Heat Pump", *ASHRAE Transactions*, 98(2)

Taylor, R. D., Pedersen, C. O., Fisher, D., Liesen, R. and Lawrie, L., 1991. "Impact Of Simultaneous Simulation Of Buildings And Mechanical Systems In Heat Balance Based Energy Analysis Programs On System Response And Control", *Proc. of the Building Simulation '91 Conference*, August 20-22, 1991, Sophia-Antipolis, Nice, France, pp. 227-234, International Building Performance Simulation Association.

Tandon, A., 1999, *Object Oriented Modeling of Vapor Compression Systems and Components*, Masters Thesis, Department of Mechanical Engineering, University of Maryland, 1999

University of Wisconsin, Solar Lab, TRNSYS.

Welsby, P., Devotta, S., Diggory, P.J., 1988. "Steady and Dynamic-State Simulations of Heat-Pumps. Part I: Literature Review", *Applied Energy*, Vol. 31, pp. 189-203

Welsby, P., Devotta, S., Diggory, P.J., 1988. "Steady and Dynamic-State Simulations of Heat-Pumps. Part II: Modeling of a Motor Driven Water-to-Water Heat-Pump", *Applied Energy*, Vol. 31, pp. 239-262

Commercial Software

Aspen Plus, Aspen Technologies

Engineering Equation Solver, F-Chart Software

Gate Cycle, Stork Engineers & Contractors, Radarweg 60, 1043 NT Amsterdam, The Netherlands

Mathematica, Wolfram Software

MATLAB, Mathworks

SINDA/FLUINT, C&R Technologies, 9 Red Fox Lane, Littleton CO 80127, USA

2. General Thermo-Fluid Simulation Structure

2.1 Introduction

The primary objective of the dissertation is the development and creation of a thermo fluid simulation framework primarily for use in refrigeration applications. A secondary goal is to build a simulation tool utilizing this framework which appeals to a diverse user group. The framework is to be general in nature, such that the tool built upon it will provide the ability to simulate experiments as they may be performed in a laboratory. This simulation tool is designed, such that each component within the system functions independently, enabling the analysis and if desired optimization of component models and/or the system as a whole. Creation of an object oriented tool is crucial for allowing users to utilize a single tool, despite having their own simulation goals, objectives and resources.

By virtue of the component independence, the simulation tool is to be multidisciplinary in nature. Although the components of a thermo fluid system, such as a vapor-compression refrigeration system are joined together, each represents a distinct discipline. While a physical system is combined using pipes and a working fluid, the virtual system, or simulation, is joined by governing equations. This chapter develops data structures and governing equations for a hierarchic thermo fluid system simulation. This undertaking results in a novel presentation for a generalized, multidisciplinary, thermo fluid system simulation.

2.2 A Note on Thermo Fluid Properties

The scope of this thesis is limited to the modeling of steady state thermo fluid energy systems. A further simplification is that of single substance mixtures. That is to say although multi-component mixtures, such as R404a, can be used as a working fluid, the composition remains constant.

In this context, a thermo fluid energy system is a system which may allow fluid flow, heat transfer and/or work.

2.3 Conservation Equations

For a non-isothermal system, there are three conservation equations that describe the relations between the inlet and outlet conditions of the stream(s) (Bird et al. 1960)¹:

- a. The *mass* balance (given in equation (2.1)) obtained by integrating the equation of continuity over the volume of the flow system.
- b. The *momentum* balance (given in equation(2.2)) obtained by integrating the equation of motion over the volume of the flow system.
- c. The *energy* balance (given in equation (2.3)) obtained by integrating the energy equation of continuity over the volume of the flow system.

The steady state continuity equation:

$$0 = \sum \dot{m}_{in} - \sum \dot{m}_{out} \quad (2.1)$$

describes the (lack of) mass accumulation within the system.

The steady state momentum equation:

¹ All equations are given here in their steady state form

$$0 = -\Delta(\mathbf{v}\dot{m} + p\mathbf{S}) - \mathbf{F} + m_{tot}\mathbf{g} \quad (2.2)$$

governs the change between fluid kinetic energy, $\mathbf{v}\dot{m}$, force on the fluid flow boundaries (product of pressure and flow surface areas), force of the control volume upon the fluid and gravity.

The steady state energy equation:

$$0 = -\Delta \left[\left(u + \frac{p}{\rho} + \frac{v^2}{2} + \hat{\Phi} \right) \dot{m} \right] + Q - W \quad (2.3)$$

governs the change between fluid internal energy, $\dot{m}u$, flow work, $\dot{m}\frac{p}{\rho}$, kinetic energy, $\dot{m}\frac{v^2}{2}$, gravitational potential energy, $\dot{m}\hat{\Phi}$, heat transfer out of the control volume, Q , and work done by the fluid, W .

2.4 Simulation Data Structures

The generalized simulation framework created in this thesis is premised upon individual component models communicating via a single coordinating simulation mechanism. The simulation is constructed via several object oriented data structures which facilitate communication of information by the coordinating, or parent system. The data structures presented here are *components*, *ports* and *junctions*.

2.4.1 Selection of Independent Variables

The framework created in this thesis is based upon the premise that components satisfy their own mass, momentum and energy balances, the system, comprised of these components is resolved by satisfying the balances at the points in between them, later termed “junctions” in this thesis.

Typical simulation approaches often simulate components differently from one another. For example, heat exchangers and pipes are often provided independent variables of mass flow rate and the thermodynamic state of the inlet fluid. In these cases, the component dependent variables are the thermodynamic state of the outlet fluid, such that the mass, momentum and energy balances are satisfied. Contrastingly, compressors and valves are often provided independent variables of thermodynamic state of the inlet fluid and outlet pressure, while the component dependent variables are the mass flow rate through the component and the thermodynamic state of the exit flows, which satisfy the component mass, momentum and energy balances.

Allowing components to possess different independent and dependent variables places a burden on an algorithm attempting to resolve a system comprised of these components. Often the algorithm requires additional information about which components require which specific variables. Additionally, often the evaluation order of the components has an affect on the solution algorithm.

In this thesis, the generality of the framework requires that all components have the same set of independent variables, and calculate the same set of dependent variables.

Choosing mass flow rate for component independent variables is problematic for a general routine, where a priori knowledge of system size and flow directions (which may be operating point dependent, rather than fixed) does not exist.

Therefore, this thesis utilizes the working fluid pressure at all flow points, and the fluid enthalpy at the fluid inlet points as component independent variables. For the single component mixture, the fluid pressure and enthalpy are adequate to define the fluid thermodynamic state. For multi-component mixtures, a composition is also required.

It is noted here, that a challenge exists here, due to the mandate. As mentioned previously, relations for many components typically exist in the form $\Delta P = \Phi(\text{mass flow})$. This makes the evaluation of the component mass flow rate implicit.

2.4.2 Components

Components are represented by appropriate engineering models which satisfy the macroscopic balances given in (2.1) - (2.3). A fluid flow enters or leaves a component through a port, which in turn communicates with a junction.

Figure 2-1 illustrates a typical “black box” component as an example, and the detailed component, which it represents. In this case, the component to be simulated is a combination of two compressors and an inter-cooler. This component communicates with its environment pressure and temperature (P_0, T_0) through work (\dot{W}) and heat transfer (\dot{Q}). The component communicates with other components in the system through their respective junctions.

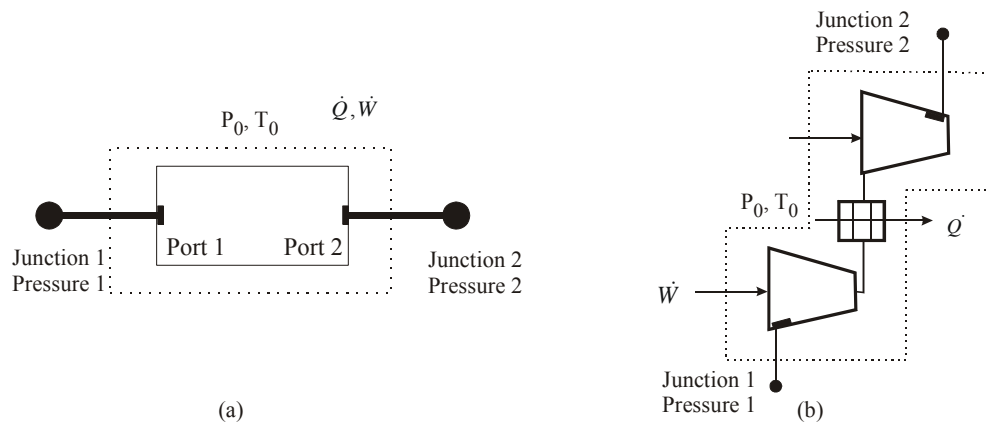


Figure 2-1: Energy System Components

Physical components are subject to certain conditions in which they operate.

Conditions imposed upon the component at its system boundaries are referred to in this

thesis as boundary conditions. Components combined into a system, are subject to both internal and external boundary conditions.

In this context, external boundary conditions are boundary conditions imposed on the component from outside the thermo fluid energy system which surrounds the component. External boundary conditions could be, but are not limited to: heat exchanger secondary fluid flow rates and temperature, an environmental temperature or a compressor's speed.

In this context, internal boundary conditions refer to the boundary conditions imposed on a component by the coordinating thermo fluid energy system of which the component is a part. Internal boundary conditions are limited to the fluid properties from fluid flow interactions with other components (via the component ports and system junctions). Internal boundary conditions include, but are not limited to: pressure, temperature, enthalpy and/or a fluid mass flow.

2.4.3 Ports

Ports are a component's interface to junctions in the system where fluid flow is the mechanism of energy transport. The fluid flow through a port has distinct values for flow rate, velocity, position (in a gravitational field) and thermodynamic state.

2.4.4 Junctions

Junctions are points, which allow for interaction between ports via fluid flow. Within the scope of this thesis, a junction is considered to be adiabatic and does not allow for any work interaction with its environment. Unlike components, junctions do not possess a unique geometry, and therefore are characterized by a single pressure.

Each fluid flow into or out of a junction has a unique mass flow rate, thermodynamic state, velocity and height.

The steady state solution of the simulation thus requires the mass and energy balances of equations (2.1) and (2.3) to be satisfied at each junction

For a special case, where kinetic and potential energies of the individual flows are ignored, junctions are assumed to have perfect mixing and thus the thermodynamic state of all fluid flows leaving a junction is identical. In this case, the energy balance can be used to resolve the appropriate value for the fluid enthalpy for each exiting fluid flow. In this instance, the single value is characteristic of the junction itself and can be referred to as the junction's mixing enthalpy.

2.4.5 Junction Evaluation

As components implicitly satisfy the conservation equations (2.1) - (2.3), the junctions can be used to simulate the thermo fluid system. For example, Figure 2-2, illustrates a general thermo fluid energy system junction. This particular junction has fluid flowing into and out of it from four different components. The individual fluid flow rates ($\dot{m}_1, \dot{m}_2, \dot{m}_3, \dot{m}_4$), fluid flow velocity (v_1, v_2, v_3, v_4) and fluid flow height (z_1, z_2, z_3, z_4) are determined from ports connected to the junction, and are determined by the solution of the component's models connected to the junction. Both \dot{m}_1 and \dot{m}_2 flow into the junction, and thus their enthalpy values, h_1 and h_2 are determined by the exit ports of the components to which they are connected. Both \dot{m}_3 and \dot{m}_4 are flowing to components. The corresponding enthalpy values, h_3 and h_4 are independent and must be determined to satisfy the energy equation applied at the junction.

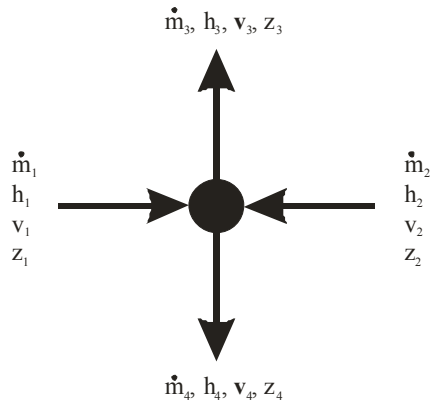


Figure 2-2: A Junction

The independent variables of a junction are therefore its pressure and enthalpy values for each exiting flow. Other junction variables, namely: fluid flow rates, velocities, elevation values and values of fluid enthalpy for entering streams are provided by the components connected to the junction, are then used to evaluate the junction mass and energy balances.

As determined previously, in the absence of kinetic and potential energies, the values for junction exit enthalpies are identical, thus this mixed enthalpy is a property of the junction. The independent variables of a junction are then its pressure and single mixed enthalpy value.

2.5 Constructing a Thermo Fluid System

Figure 2-3 shows two general energy systems; one is an open system, while the other is closed. Each system contains six components, connected via junctions. All component ports not communicating across a system boundary must be connected to an internal junction.

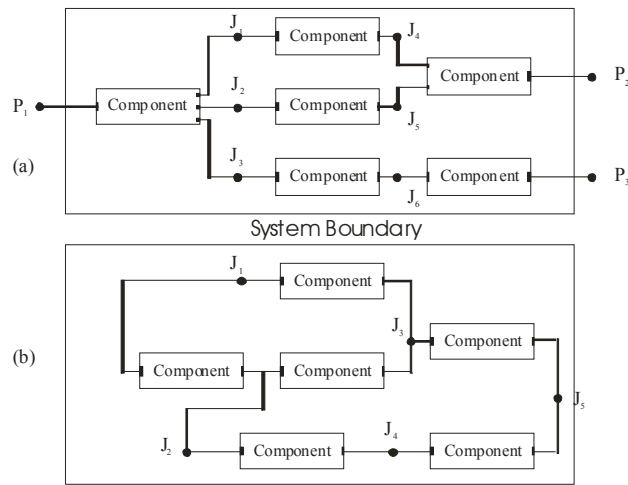


Figure 2-3: Open (a) and Closed (b) Energy Systems

Assuming thermo fluid energy flow, i.e. internal, kinetic and potential energy only, an “energy of the flow” at each port \dot{E}_p is expressed as $\dot{m}_p(h_p + \frac{v_p^2}{2} + z_p g)$, where \dot{m}_p is the mass flow rate at a given port, and h_p is the enthalpy of the fluid flowing through the port at a velocity v , and z is the port elevation in a gravitational field \mathbf{g} . (Bird, Stewart and Lightfoot, 1960)

Solution of this energy system simulation requires knowing the energy flow rates at each component port. The pressure difference of the fluid between ports is the driving potential for the flow, and will thus determine the flow rate of fluid through the component.

The pressure of each port connected to a junction is identical; therefore there are N_J (number of junctions) unknown pressures. Each port may have a unique value for enthalpy, mass flow, position (elevation relative to a reference) and velocity, adding an additional $4 N_p$ (number of ports) unknown variables. This results in a total of $4 N_p + N_J$ unknown variables which must be solved for.

The governing equations of *conservation of mass and conservation of energy* at the junction level provide $2N_J$ equations. The remaining equations are provided by the components, which provide their port position (as a component property) and determine their port mass flow and velocity as well as the energy state for outlet ports based upon boundary conditions. These boundary conditions are the thermodynamic state at the ports (internal boundary conditions) and component specific independent variables.

As mentioned previously, a special circumstance exists when internal energy plus flow energy represent the only energy accounted for by a flowing fluid, i.e. kinetic energy and potential energies are considered negligible. In this case, the unknown variables for the simulation are then N_J unknown pressures, as well as a unique value for enthalpy and mass flow at each port. This results in a total of $2 N_p + N_J$ unknown variables which must be solved for (Lindsay 2000).

Table 2-1 summarizes the system level equations for generalized thermo fluid energy system simulation for a single component fluid.

The important differences between open systems, where mass is allowed to cross the system boundary, and closed systems, where mass is conserved within the system boundary, are the boundary conditions imposed on the system, and the effect they have on the system charge.

In the open system, the system's boundaries are specified using thermodynamic properties, and allow mass to flow freely through them. A change in system boundary conditions changes the flow through the system and consequently, the mass within. In a closed system, mass is conserved not only at junctions, but for the system as a whole under all conditions.

Consequently, for the closed system, mass must be accounted for in each of the components, at all operating points. A change in system boundary conditions (via the components) changes the operating point of the system, but not its system charge. For the closed system the junction mass flow rates are no longer independent. A mass balance over each junction results in a redundant equation, so one junction mass balance must be removed. The number of system variables in the simulation remains the same however. A mass balance over the entire system provides the equation to close the system of equations.

Property Balance	Number of Equations	Equation
Junction Mass Balance ²	N_J or N_J-1	$\sum_{p=1}^{N_{ports}} \dot{m}_p = 0$
Junction Energy Balance	N_J	$\sum_{p=1}^{N_{ports}} \dot{m}_p (h_p + \frac{v_p^2}{2} + g z_p) = 0$
Mass flow through port p of component i	N_p	$\dot{m}_p = \Phi_1(P_{in}, h_{in}, P_{out})$
Thermodynamic state of each exit port in component	N_p Out of Components	$h_3 = \Phi_2(P_{in}, h_{in}, P_{out})$
Velocity of fluid through port p of component i	N_p	$v_p = \Phi_3(P_{in}, h_{in}, P_{out})$
Position (height) of port p of component i	N_p	$z_p = \text{Constant}$
System Mass Balance ³	1 or 0	$Mass_{System} = \sum_{i=1}^{N_{Components}} Mass_i$

Table 2-1: Equations used to Model Generalized Thermo Fluid Energy Systems

Table 2-2 summarizes the changes in Table 2-1 for a simplified thermo fluid energy system simulation, where kinetic and potential energies are neglected.

² The number of these equation depends upon the system boundary conditions: An open system requires N_J mass balances, while a closed system requires only $N_J - 1$ mass balances

³ A system mass balance is only required for closed systems.

Property Balance	Number of Equations	Equation
Junction Mass Balance ⁴	N_J (Open systems) or N_J-1 (Closed Systems)	$\sum_{p=1}^{N_{ports}} \dot{m}_p = 0$
Junction Energy Balance	N_J	$\sum_{p=1}^{N_{ports}} \dot{m}_p (h_p + \frac{v_p^2}{2} + gz_p) = 0$
System Mass Balance ⁵	1 or 0	$Mass_{System} = \sum_{i=1}^{N_{Components}} Mass_i$

Table 2-2: Equations used to Model Generalized Thermo Fluid Energy Systems with Negligible Kinetic and Potential Energies

2.6 Controlling the System

The general energy system simulation technique detailed here will result in a solution of the components that make up the energy system and for the internal boundary conditions supplied to the components. However, energy systems can have more than a single operating point. When a component has adjustable parameters, a change in one may affect the operation of the system. The changing of system parameters, including component parameters, to achieve results is controlling of the system. The system can be controlled by altering system level parameters, or component level parameters.

For an open system, one method of control, which does not involve manipulation of any component parameters, is achieved by modifying a system boundary condition. For example: changing the throttle pressure of a steam turbine, changing the back pressure on a compressor and changing the inlet temperature on a heat exchanger. For a closed system, the only non-component, system boundary condition that can be utilized to

4 The number of these equation depends upon the system boundary conditions: An open system requires N_J mass balances, while a closed system requires only N_J-1 mass balances

5 A system mass balance is only required for closed systems.

control the system is the working fluid charge contained within the system. Increasing or decreasing the system charge alters the system operating point.

System control through component manipulation can be, but is not limited to, an opening of a valve orifice to change its flow characteristic, the increase of a condenser fan speed, to provide additional cooling of the condenser, or the bleeding of hot gas from the compressor discharge to the evaporator inlet to accommodate low load conditions.

It is desirable for systems to have certain system performance variables set to particular values. This is done via active control of the components independent variables. This is a challenge, as components do not have access to system level information that is not communicated through a junction. This can be addressed by giving the system a control. A control is aware of the following: a current value for the objective dependent variable, the desired value of that objective variable, the component being controlled, the independent variable to control in the component and the value of the independent variable.

An example of an effective control scheme for a vapor compression refrigeration system using the above described technique is that of the thermostatic expansion valve (TXV). A system simulation that has a TXV and uses the value of evaporator superheat to solve the system cannot be modeled with the generalized technique, as there is no physical flow between the evaporator outlet and the TXV. Rather, a control is added to the system. For this example system, the system can be aware of several system-level variables which may have set-point values. For instance evaporator superheat, condenser sub-cooling, compressor head pressure, or compressor suction pressure. The control can evaluate these system variables after the completion of a simulation. The control can

then perturb the 'control variable' (TXV orifice opening) and have the system solve again, and update the new value of the control variable until the objective is obtained.

2.7 Simulation Hierarchy

The primary objective of the dissertation is the development of a general-purpose simulation framework, for analysis and design of the vapor-compression system and its components. The thermo fluid energy system itself is the parent, or coordinating system. The coordinating system manipulates values of independent variables at junctions, which share information with ports, and thus components, which are sub-systems.

2.8 Simulation Framework Design Rules

A generalized framework is developed here which allows for the construction of thermo fluid energy systems with character similar in nature to refrigeration systems. Within the context of this thesis, the framework is simplified down to a steady state model, with negligible kinetic and potential energy. A set of design rules defines the framework, and is presented as:

1. The system independent variables consist of the value for the fluid pressure at each junction, as well as the value for fluid enthalpy for all flows leaving junctions, flowing to components.
2. All components are individual units, and are subject to identical inputs and outputs. In this case, all components must evaluate their dependent variables when provided values for working fluid, fluid pressure at each port, and fluid enthalpy at each port where fluid flows into the device.

3. All components must provide identical outputs. In this case, all components provide the value for fluid mass flow rate at each outlet port, and the value for fluid enthalpy at each outlet port.
4. The system solution must satisfy an energy balance at each junction and mass balance at $N-1$ junctions for systems with no open boundaries and N junctions for systems with open boundaries, where N is the number of junctions in the system.
5. Components have complete control of their independent variables, other than port pressures and enthalpies. Note: A component can, upon request provide a list of independent variables, which may be of use to a program implementing the framework, and serve as a system control. This is demonstrated in chapter 6, where VapCyc is explained.

2.9 Summary

This chapter details the creation of a thermo fluid simulation framework. This framework is such that each component within the system is an independent, stand alone program.

The framework is created upon a system of data structures and property balances. The structures consist of components, ports and junctions which are then combined to form specific energy systems.

The components are similar to real world devices in that when boundary conditions are applied, a steady state solution results. The component has several ports, to which it communicates the flows through the component and the thermodynamic state of fluids flowing through them.

A coordinating system, which contains sub-systems, is resolved through conservation equations of mass and energy. The coordinating system has independent variables, which are intensive thermo-fluid properties: junction pressures and enthalpy. The sub-systems then provide values for their dependent variables: mass flow and exit specific enthalpy for a given set of junction properties. The coordinating system is resolved when the junction pressures and enthalpies provided to the sub-system result in mass flow and exit specific enthalpy values which satisfy the coordinating system conservation laws.

2.10 References

Rice, C.K., Wright, L. and Bansal, P.K., 1993, "Thermodynamic Cycle Evaluation Model For R-22 Alternatives In Heat Pumps-Initial Results And Comparisons". Moneton, Canada: Proc. 2nd Int. Conference on Heat Pumps in Cold Climates, Aug 16-17

Bird, Steward and Lightfoot, 1960, "Transport Phenomena", John Wiley and Sons, 1960

Hatsopolous, G., Keenan, J, 1965, "Principles of General Thermodynamics", John Wiley and Sons, 1965

Obert, E., Gaggioli, R., 1963, "Thermodynamics", McGraw Hill, 1963

Lindsay, D., 2000, "Network Builder Notes, University of Maryland – CEEE, Internal Document, July, 2000

3. The Junction Solver

In this chapter, a general purpose thermo-fluid simulation solver is created to serve as the coordinating system in an object oriented multidisciplinary tool. This general purpose solver, referred to in this thesis as the *Junction Solver*, provides specific functionality to VapCyc, a vapor compression refrigeration simulation tool. The Junction Solver feature list is as follows:

1. Single solver many thermo fluid energy systems
2. Specifically applicable to:
 - Vapor compression refrigeration (closed system)
 - Pipe network (open or closed system)
 - Heat exchanger (open system)
3. Allow user defined component connectivity.
4. The functionality to change, or swap, component models used in a fixed-component simulation at run-time thus replacing components without the need to re-compile the simulation code itself.
5. The functionality to add or subtract components at run time, without the need to re-compile the simulation code itself.
6. The ability to constrain and specify the total mass of working fluid for the system simulation.
7. Allow for the selection from several different working fluids/refrigerants.
8. Flexibility to alter component independent variables, independently from the main simulation program.

9. Arbitrary order of component evaluation, eliminating the need for a run-time specified order for component evaluation

This chapter details necessary structure the Junction Solver possesses to fulfill the objectives given in the above feature list.

3.1 The Junction Component Port Matrix (JCP)

The foundation of the Junction Solver is the connectivity, defined here as the manner in which individual component ports are connected to the coordinating system junctions. Connectivity is described to the solution algorithm at run time, via a data structure called a Junction Component Port matrix, or, JCP.

As an example, Figure 3-1 illustrates a basic vapor compression refrigeration system, with a hot gas bypass valve installed between the compressor discharge and evaporator inlet. This valve takes the compressor discharge flow and meters it into the evaporator inlet to keep the evaporator pressure up during low load situations. In the figure, components are numbered 1-5, junctions are numbered 1-4 and component ports are numbered from 1-n, with n being the maximum number of ports in an individual component.

Table 3-1 represents the JCP for the thermo fluid system depicted in Figure 3-1, a two dimensional matrix. In the JCP, rows are numbered 1- N_j , the columns are numbered 1- N_C , where N_j is the number of junctions in the coordinating system and N_C is the number of components (sub-systems) in the system. The matrix elements, $JCP[j][c]$ are the port number corresponding to component C which connects to junction J.

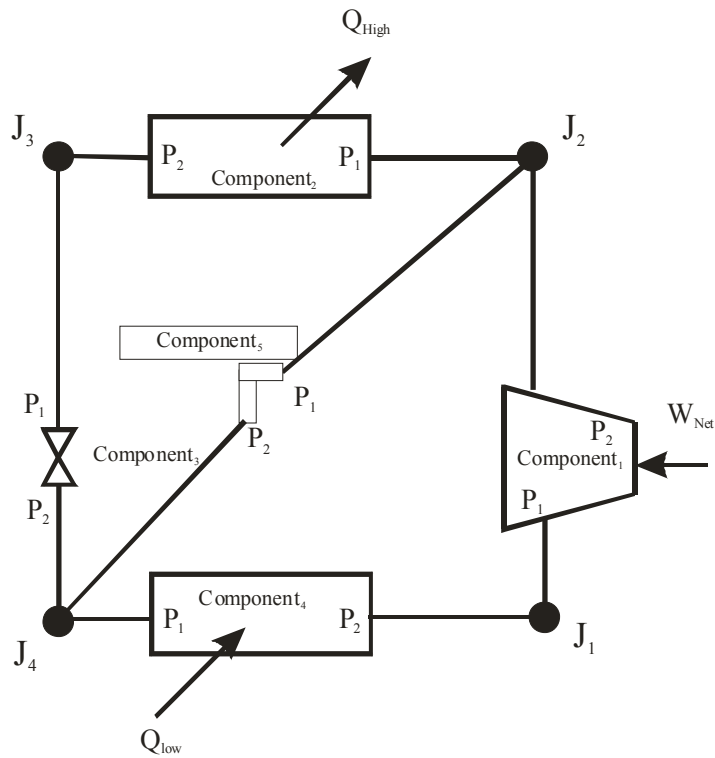


Figure 3-1: A Modified Vapor Compression Refrigeration System

The JCP is a flexible data structure which serves as the starting point for the Junction Solver solution. The JCP enables a simulation program utilizing the Junction Solver as its main solution engine, providing the utility given in 1-4 of the above feature list given in the introduction to this chapter.

Component \ Junction	1	2	3	4	5
1	1	0	0	2	0
2	2	1	0	0	1
3	0	2	1	0	0
4	0	0	2	1	2

Table 3-1: JCP for a Vapor Compression Refrigeration System

3.2 Component Specification

Features 5-7 from the above feature given at the beginning of this chapter are achieved through the object oriented multidisciplinary nature of the simulation.

Components, regardless of function, are to a degree “all the same” to the coordinating system, in this case the Junction Solver. Consequently, there exists a list of properties and methods the components must possess. Table 3-2 displays a list of properties the each component utilized by the Junction Solver must possess to resolve the simulation.

Property	Description
Charge	The mass contained within the component at the current operating point.
Port Enthalpy(p)	The enthalpy of the working fluid, at port p, at the current operating point
Port Massflow(p)	The mass flow of the working fluid, at port p, at the current operating point
Port Pressure(p)	The pressure of the working fluid, at port p, at the current operating point
Refrigerant	The working fluid
SystemUnits	The unit system which the component must comply with when receiving/giving numeric values

Table 3-2: A List of Necessary Properties for all Components Utilized in the Junction Solver

After the Junction Solver sets the values for the component thermodynamic boundary conditions, the component is then run via the Run method, and values for the component dependent variables can be assembled and used to evaluate the Junction Solver residual equations.

3.3 Independent Variables and Residual Equations

The Junction Solver, configured by an interface, formulates a system of equations to resolve, subject to a set of independent variables.

Chapter 2 of this thesis developed the necessary property balances, namely mass and energy, which when properly resolved at each junction; result in a solution sufficient to the energy system as a whole. This is predicated by the component models themselves, solving the respective mass, momentum, and energy balances. This thesis neglects the effects of kinetic and potential energies.

Recalling from chapter 2 that a component, with a number of ports numbered 1 to N_p , and a number of inlet ports numbered 1 to $N_{p,in}$, has independent variables, P_i and h_j , where i numbers from 1 to N_p , and j numbers from 1 to $N_{p,in}$. The component then calculates its dependent variables \dot{m}_i, v_i, z_i and h_k , where k refers to ports where mass flow exits, and numbers from 1 to $N_p - N_{p,in}$.

The system simulation independent variables are then: P_i , and h_j , where, $1 < i < N_{Junctions}$, and $1 < j < N_{Junction\ Exit\ Streams}$.

The Junctions Solver then uses these dependent variables to calculate the following residual equations:

$$r_i = \sum_{p=1}^{N_{ports}} \dot{m}_p \quad (3.1)$$

$$r_j = \sum_{p=1}^{N_{ports}} \dot{m}_p h_p \quad (3.2)$$

$$r_l = Mass_{System} - \sum_{C=1}^{N_{Components}} Charge_C \quad (3.3)$$

The residual equations given here are the junction mass balance, junction energy balance and the overall system mass balance.

When the junction solver operates in ‘open system mode’, a mass balance is necessary for each junction. When the junction solver operates in ‘closed system mode’,

a mass balance is only necessary $N_{\text{Junctions}}-1$ junctions, and equation (3.3), the system mass balance, is necessary.

In all cases, an energy balance is required at each junction.

3.4 Solution of the Model

Organizing the residual equations, given in equations (3.1)-(3.3) in a vector $\vec{\Phi}$, the vector of residual values \vec{r} can be evaluated through the use of the state (solution) vector \vec{S} .

The resulting system of equations,

$$\begin{aligned} (a) \quad \vec{\Phi}(\vec{S}) &= 0 \\ (b) \quad \vec{\Phi}(\vec{S}) &= \vec{r} \end{aligned} \tag{3.4}$$

is a non-linear set. Equation (3.4)(a) is the actual set of equations, and (3.4)(b) is the approximated set, where \vec{r} represents the residual, or deviation from zero. In this case, Φ_i represent the mass and energy balances to be solved. S_i are the individual state variables which make up the independent variables, namely P_j and h_j .

Solution of the system of equations given by equations (3.1) - (3.3) can be accomplished in many ways. Two methods are currently implemented in the junction solver, namely (1) a simultaneous approach, where the full set of equation is specified to a non-linear equation solver, and (2) a modified fractional step method, based upon the work of Jiang (2003).

3.4.1 Simultaneous Approach

A general approach which does not require any a priori knowledge of system configuration defines the state vector S as $S^T = [P_1, P_2, \dots, P_{N_j}, h_1, h_2, \dots, h_{N_h}]$, where N_j is the

number of system junctions, and N_h is the number of independent enthalpy values. Here S is a vector of thermodynamic properties only. The state vector also has no implications about system configuration. This approach gives a 'Junction Solver', which takes a candidate state vector, and populates the system junctions with pressure and enthalpy values. After propagating these values of pressure and enthalpy to the proper components, the components are then executed and the resulting balance equations evaluated. Figure 3-2 illustrates the calculation procedure for this method.

This simultaneous approach allows employing any non-linear equation solver, but has the drawbacks of reliance on numerical derivatives and a need for a "sufficient" initial guess.

The current version of the Junction Solver achieves the simultaneous solution of the non-linear equation set via the Newton-Raphson or Broyden's method. (Nash and Sofer, 1996)

3.4.2 Fractional Step

Jiang offers a so called fractional step technique for solving heat exchangers modeled in a component/junction manner, as illustrated in Figure 3-3. This two step technique is characterized by (1) a solution for the system junction pressures through resolution of the system mass balances at fixed system (thermal) energy, and (2) a solution for the unknown junction enthalpy values through resolution of the system energy balances at fixed (port) flow rate.

Although the system pressure field must always be solved simultaneously, Jiang exploits the fact that the energy equations can often be solved successively, reducing the

need to compute numeric partial derivatives. Figure 3-3 gives a flow chart for Jiang's method of solving heat exchangers.

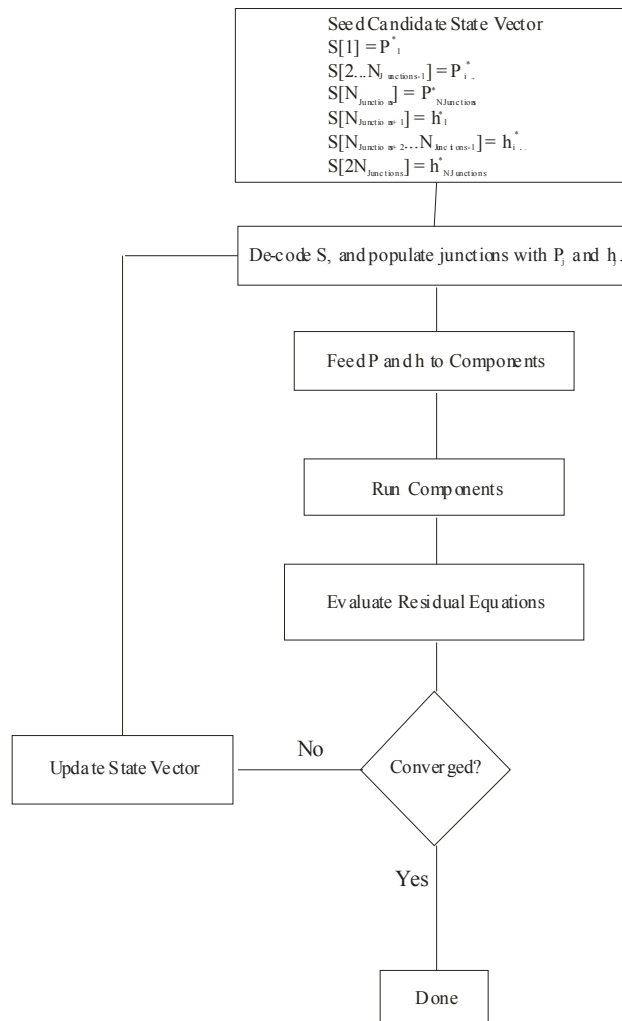


Figure 3-2: Flow Chart for Simultaneous Equations

Jiang utilizes his fractional step technique to solve heat exchanger systems, interacting with a secondary fluid, namely air. Furthermore, it is seen in Figure 3-3 that the solution of the component models is not completely separated from the solution of the refrigerant side; which is evident from the updating of the air side enthalpy values.

Recalling (Balling and Sobieszcanksi-Sobieski, 1994), the system is considered ‘non-hierarchic’ when coordinating and sub-system interactions are not forbidden. This represents a more complex case than the ‘hierarchic’ one, and is beyond the scope of the Junction Solver.

Additional methods are therefore necessary for the Junction Solver to utilize a fractional step technique, namely a (1) the ability to determine the component mass flow rate at the given internal (thermodynamic) boundary conditions without changing the component’s energy state, and (2) the ability to determine the component energy state given specified mass flow rate values at the ports.

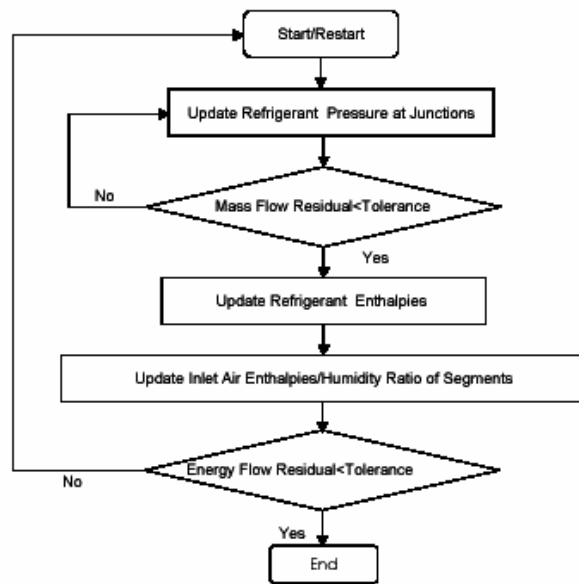


Figure 3-3: Fractional Step Method of Jiang (2003)

It is noted here, updating the “enthalpy field” for a constant component mass flow rate values, may result in a density change within one or more components. This density change may result in a deviation between the component’s current specified mass flow

rate and one calculated with the current enthalpy field. It is therefore necessary for components to have additional properties, namely values for their (deviations in) mass, and energy balances, at a specified set of port pressure and enthalpy values.

Figure 3-4 shows a modified fractional step approach which can be used in the hierarchic structure of the Junction Solver.

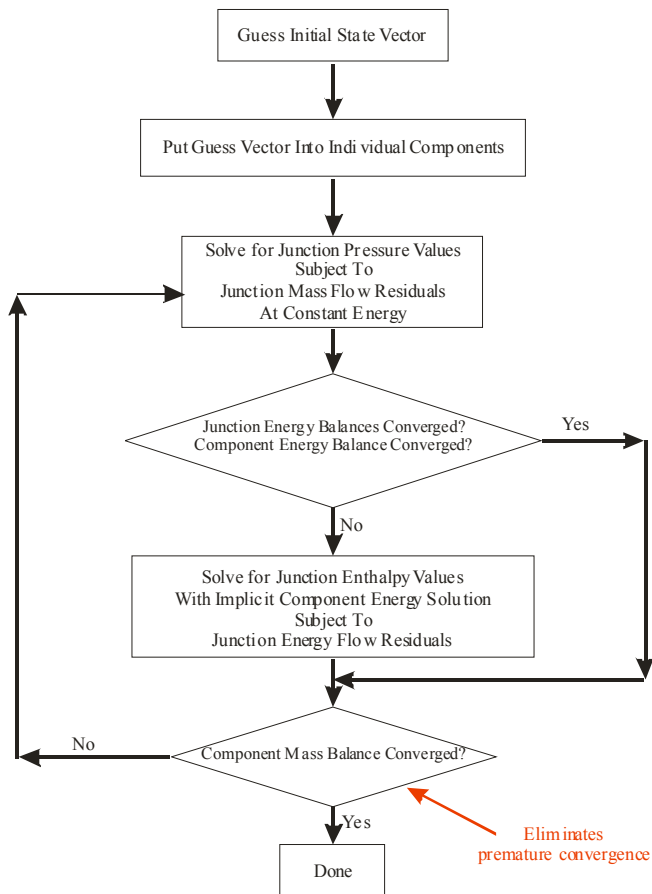


Figure 3-4: A Modified Fractional Step for the Junction Solver

The modified fractional step method will iterate until the junction mass, and energy balances are satisfied, *and* the component mass, momentum, energy balances are satisfied too.

3.4.3 Solving Hydraulic Relations

When the Junction Solver resolves the junction mass balances only, as given in equation (3.1) a non-linear equation solution technique such as Newton-Raphson or Broyden's method is employed. (Nash and Sofer, 1996).

3.4.4 Solving Thermal Relations

For the special case of the Junction Solver resolving the junction energy balances only, as given in equation (3.2), when kinetic and potential energies are ignored, a successive substitution method is employed. This technique consists of the repetition of two steps, namely (1) mixing of junctions, where the junction's mixture enthalpy is determined and fed to all tubes that receive flow from the junction and (2) running of the tubes energy routines at constant mass flow. The routine is presented graphically in Figure 3-5.

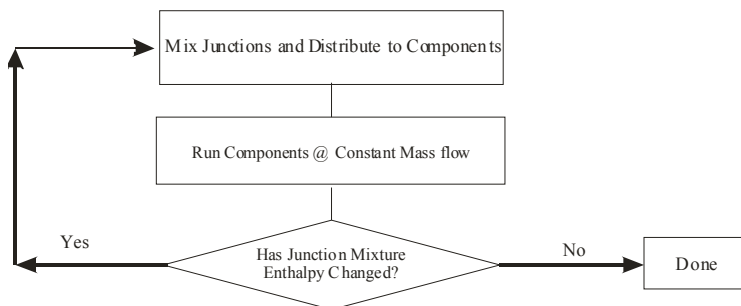


Figure 3-5: Successive Substitution Energy Routine

3.5 Conditioning of the Model

The iterative solution of nonlinear systems of equation often requires the solution of auxiliary linear systems; hence the latter has basic influence on the efficiency of the

overall method (Farago, I. and Karatson, J, 2002). The residual equations presented in (3.1) - (3.3) are nonlinear in nature due to the thermo physical properties of the fluids.

The Junction Solver developed in this thesis incorporates iterative solution techniques which require the solution of auxiliary linear systems. Rather than investigate elaborate preconditioning techniques, the junction solver incorporates the scaling of the independent variables and residual equations. The scaling is basic, scaling all independent variables between a maximum and a minimum value, and all residual equations by a characteristic value for mass flow rate or energy.

This leaves the determination of the scaling values to the specific energy system simulation which utilizes the junction solver, to determine the value for scaling parameters.

The scaling process is described here, while the specific methods for determining the value of the scaling parameters is left for future chapters, where the Junction Solver is utilized to simulate specific systems.

As the independent variables of the Junction Solver are pressure and enthalpy values, the dimensionless independent variables are defined as:

$$P_i^* = \frac{P_i - P_{low}}{P_{high} - P_{low}} \quad (3.5)$$

$$h_i^* = \frac{h_i - h_{low}}{h_{high} - h_{low}} \quad (3.6)$$

The Junction Solver must be provided values for P_{high} , P_{low} , h_{high} , h_{low} , which ideally represent the highest and lowest values for pressure and enthalpy encountered in the system.

The residual equations given in (3.1) - (3.3) represent the mass and energy balances of the system. The Junction Solver resolves scaled version of these equations, give as:

$$r_i = \frac{\sum_{p=1}^{Nports} \dot{m}_p}{\dot{m}^*} \quad (3.7)$$

$$r_j = \frac{\sum_{p=1}^{Nports} \dot{m}_p h_p}{\dot{E}^*} \quad (3.8)$$

$$r_i = \frac{Mass_{System} - \sum_{C=1}^{NComponents} Charge_c}{Mass^*} \quad (3.9)$$

The values of \dot{m}^* , \dot{E}^* and $Mass^*$, represent characteristic values for the system mass flow, energy flow and total working fluid mass.

3.6 The Solution Seed

The Junction Solver attempts to resolve a state vector, \vec{S} , subject to equations (3.7) – (3.9). The junction solver itself is completely independent of system configuration, thus the determination of the seed is given to the specific energy system simulation which utilizes the junction solver.

3.7 Summary

This chapter describes the general function of the Junction Solver, a general purpose thermo fluid simulation solver. In addition to the JCP matrix, which describes the system connectivity to the Junction Solver, a limited set of component properties and methods is introduced. The Junction Solver solution methodologies are developed and presented here as well. The solution methodologies are dependent upon the component level properties and methods presented in this chapter.

Lastly, a general method for scaling the problem for numeric stability is presented. The method is broad, introducing scaling factors which must be provided values by the thermo fluid system simulation utilizing the Junction Solver.

The simulation framework presented is able to serve many of the diverse needs of the constituents, as presented previously, namely:

1. Single solver for many thermo fluid energy systems.
2. Allow user defined component connectivity.
3. The functionality to change, or swap, component models used in a fixed-component simulation at run-time thus replacing components without the need to re-compile the simulation code itself.
4. The functionality to add or subtract components at run time, without the need to re-compile the simulation code itself.
5. The ability to constrain and specify the mass of working fluid for the simulation of system simulation. For a vapor-compression refrigeration application, this is referred to as “Charge Management”.
6. Allow for the selection from several different working fluids/refrigerants.
7. Flexibility to alter component independent variables, independently from the main simulation program.
8. Arbitrary order of component evaluation, eliminating the need for a run-time specified order for component evaluation.

3.8 References

Balling, R.J. and Sobieszcanksi-Sobieski, J., “Optimization Of Coupled Systems: A Critical Overview Of Approaches”, *Proceedings of the 5th AIAA/NASA/USAF/ISSMO*

Symposium on Multidisciplinary Analysis and Optimization, Panama City, Florida, September 7-9, 1994. Paper No. AIAA-94-4330-CP

Farago, I. and Karatson, J., 2002, *Numerical Solution of Nonlinear Elliptic Problems Via Preconditioning Operators*, Nova Science Publishers, Inc, New York, 2002

Jiang, H., 2003, *Development Of A Simulation And Optimization Tool For Heat Exchanger Design*, Ph.D. Thesis, Department of Mechanical Engineering, University of Maryland, 2003

4. Simulation of a Heat Exchanger

A secondary objective of the dissertation is the development of a heat exchanger model built upon the Junction Solver. A heat exchanger of the type simulated in this chapter, represents a compound thermo fluid energy system, where two systems interact, specifically a refrigerant system and an air system. Many heat exchanger models exist, and the objective of this simulation is the illustration of a specific incarnation of the Junction Solver. The example demonstrates the Junction Solver, and accomplishes three goals:

1. Validation of the Junction Solver general purpose thermo fluid simulation tool
2. Utilization of the Junction Solver to solve a compound system where two dependent thermo fluid systems interact
3. Verification of a simulation model which will be used in a system simulation model, VapCyc, and facilitate its validation

The actual heat exchanger models are designed to represent physical pieces of hardware which have been instrumented and tested in the laboratory (Hwang et al. 2004).

4.1 Model Components

4.1.1 Tubes

Tubes are finned conduits for fluid flow. As are all other thermo fluid energy systems, they are characterized by mass, momentum, and energy conservation. For the scope of this heat exchanger study, kinetic and potential energies are considered

negligible, and the steady state balance equations necessary to simulate the tube are:

(Bird et al., 1960)

Continuity:

$$0 = \sum \dot{m}_{in} - \sum \dot{m}_{out} \quad (4.1)$$

For the pipe geometry, the continuity equation simply states the mass flow into the pipe is equivalent to the mass flow out of the pipe.

Momentum:

$$0 = -\Delta(\mathbf{v}\dot{m} + p\mathbf{S}) - \mathbf{F} + m_{tot}\mathbf{g} \quad (4.2)$$

The momentum equation is the relationship between the flow, and the pressure difference required to drive the fluid through the resistance the pipe walls exert upon the fluid. The unknown quantity in the momentum equation is this force \mathbf{F} . Typically, correlations created from empirical data are used to estimate this force.

Energy:

$$0 = -\Delta \left[\left(u + \frac{P}{\rho} \right) \dot{m} \right] + Q - W \quad (4.3)$$

The energy equation governs the change between fluid internal energy, $\dot{m} u$, flow work, $\dot{m} \frac{P}{\rho}$, heat transfer out of the control volume, Q , and work done by the fluid, W .

For a pipe, there is no (shaft) work done on the working fluid.

The tube has three distinct modes of operation.

1. When supplied appropriate internal boundary conditions of pressures and enthalpy values for all inlet streams, and external boundary conditions (air temperature, flow rate), the tube will resolve its mass flow rate and exit

enthalpy state which meets both the thermal and hydraulic relations for the model.

2. When supplied an inlet pressure and outlet pressure, a mass flow rate, inlet enthalpy value, and external boundary conditions, the tube will resolve an exit enthalpy which meets the thermal relations. Note: The resulting solution may violate hydraulic relations for the model.
3. When supplied with boundary pressures and enthalpy values, the tube will resolve a mass flow rate which meets the hydraulic relations. Note: The resulting solution may violate the thermal relations for the model.

4.1.1.1 Hydraulic Relations

White (1986), under the assumptions of incompressible flow, gives a method for obtaining the pressure drop through a pipe as

$$\Delta P \approx 0.158L\rho^{3/4}\mu^{1/4}d^{-7/4}V^{7/4} \quad (4.4)$$

For this calculation, density, ρ , viscosity, μ , tube diameter d and fluid flow velocity, V are representative of average values. This formula can be rearranged to an explicit form for mass flow

$$\dot{m} \approx 2.8702 \left(\frac{d^{3/4}\Delta P}{L\mu^{1/4}\rho^{3/4}} \right) \rho A \quad (4.5)$$

It is noted here, that this correlation is valid in the range $4000 < Re_d < 10^5$. While the simulation may have values for Re_d outside of this range, the relationship captures the characteristic behavior of the flow for most applications, and serves the purposes of this work.

Additionally, a correction term, DP_{Factor} is included in the model. This term is a constant that can be adjusted to fine tune the tube performance.

$$\dot{m} \approx 2.8702 \left(\frac{d^{5/4} \frac{\Delta P}{DP_{\text{Factor}}}}{L \mu^{1/4} \rho^{3/4}} \right) \rho A \quad (4.6)$$

It is stressed here that this model is dependent upon average values for the fluid thermophysical properties.

For a pipe, with a specified mass flow rate and thermodynamic state, the pipe model can then calculate a mass flow rate based upon this thermodynamic state. If the calculated mass flow rate does not agree with a specified mass flow rate, a mass flow residual, representing the tube's violation of its mass balance is given by

$$r_{\text{mass}} = \dot{m}_{\text{specified}} - \dot{m}_{\text{calculated}} \quad (4.7)$$

4.1.1.2 Thermal Relations

Relations between the heat transfer and the heat exchanger geometry can be calculated using an effectiveness-NTU method (White, 1984).

$$\begin{aligned} \varepsilon &= \frac{T_{1,o} - T_{1,i}}{T_{2,i} - T_{1,i}} = \frac{\dot{Q}}{\dot{Q}_{\text{Max}}} \\ NTU &= \frac{UA}{c_1} = \frac{UA}{(\dot{m}c_p)_1} \end{aligned} \quad (4.8)$$

The effectiveness (ε) is a ratio of heat transfer to maximum possible heat transfer, while the number of transfer units (NTU) is a “thermal size” relating heat transfer area (A) and heat transfer coefficient (U) to heat capacity rate ($\dot{m}c_p$). The heat transfer kinetic relations for the two fluids are then used to complete the effectiveness-NTU relationship.

$$U = \frac{1}{\frac{1}{h_o} + R_{Contact} + R_{Fouling} + \frac{1}{h_i FR}} \quad (4.9)$$

Here the overall heat transfer coefficient is described in terms of the outer (air) and inner (refrigerant) heat transfer coefficients (h) as well as thermal resistances for conduction, surface fouling and an outside to inside surface area ratio (FR).

The tubes are modeled as air to refrigerant heat exchanger in cross-flow, with the air side mixed. The effectiveness-NTU relationship for this is (White, 1984)

$$\begin{aligned} \varepsilon &= 1 - \exp\left[-\frac{1 - \exp(-NTU C^*)}{C^*}\right]; \quad C_{\min} \text{ mixed}, C_{\max} \text{ unmixed} \\ \varepsilon &= \frac{1}{C^*} (1 - \exp(-C^* [1 - \exp(-NTU)])); \quad C_{\max} \text{ mixed}, C_{\min} \text{ unmixed} \end{aligned} \quad (4.10)$$

The heat capacity rate of a fluid $\dot{m}C_p$ is denoted by C, and C* is the ratio of two heat capacity rates, such that $C^* \leq 1$.

Although this relationship is fine for super-heated and sub-cooled liquids, this relationship does not apply to fluids in a two-phase state. A condensing or evaporating fluid can be thought to have an infinite heat capacity since its temperature does not change with an increase in energy. Therefore when a single phase fluid is in thermal contact with an evaporating or condensing fluid, the effectiveness is expressed as:

$$\varepsilon = 1 - \exp(-NTU) \quad (4.11)$$

4.1.1.3 Refrigerant Side Heat Transfer Coefficients

For simplicity, all flows are modeled as turbulent. Single phase heat transfer coefficients are calculated using Gnielinski's correlation (Kakac, 2002, p.96), given as

$$f = (1.58 \ln(Re) - 3.28)^{-2}$$

$$Nu_d = \frac{(f/2)(Re - 1000)Pr}{1 + 12.7(f/2)^{1/2}(Pr^{2/3} - 1)} \quad (4.12)$$

$$h = Nu_d k / d$$

In this case, the correlation is valid for fully developed turbulent flow with a constant wall heat flux, $Pr > 0.7$.

Condensation heat transfer coefficients are calculated using Shah's (1989) relationship

$$h = h_{lo} \left((1-x)^{0.8} + \frac{3.6x^{0.76}(1-x)^{0.04}}{\left(\frac{P}{P_{Cr}}\right)^{0.38}} \right) \quad (4.13)$$

where h_{lo} is the heat transfer coefficient were the fluid only liquid, and in this case is calculated by Dittus and Boelter (1930) as

$$h = 0.023(k/d)Re^{0.8}Pr^{0.3} \quad (4.14)$$

The above correlation is valid in the range $Re_{lo} > 350$, $Re_{vo} > 35000$.

4.1.1.4 Air Side Heat Transfer Coefficients

Likewise Kim et. al (1999) provide heat transfer calculations in terms the fin geometry (height, depth, spacing), maximum air velocity and air thermophysical properties.

$$h_{air} = 0.163Re_d^{-0.369}(Height_{Fin}/Depth_{Fin})^{0.106}(Spacing_{Fin}/d)^{0.0138}Pr^{-2/3}\rho_{air}u_{max}Cp_{air} \quad (4.15)$$

For the geometry and air flow given in Figure 4-4, this results in $h_{air} = 68.99 \text{ W/m}^2\text{-K}$.

4.1.1.5 Air Surface Area Calculations

The air side calculations are independent of the refrigerant side of the tube. A specified tube geometry and air flow rate facilitates the calculation of the air heat transfer coefficient, fin ratio and fin efficiency.

Utilizing relations for fin ratio and fin efficiency given in Chang et al. (1997)

$$A_{Tube} = \pi d(L - N_{Fins} Thickness_{Fins}) \quad (4.16)$$

The fin surface area is the cross-sectional area of the fin parallel to the air flow.

$$A_{Fins} = 2N_{Fins}(Depth_{Fin}Height_{Fin} - A_{cross,tube}) \quad (4.17)$$

The fin ratio is defined as the ratio of fin surface area to tube surface area were no fins present.

$$FR = \frac{A_{Tube} + A_{Fins}}{\pi dL} \quad (4.18)$$

4.1.1.6 Distributed Models

The effectiveness-NTU relationship provided by equation (4.8) is only adequate for constant values of the overall heat transfer coefficient U. In situations where a tube has both heat transfer between air and a single phase fluid, as well as heat transfer between air and a condensing fluid, the value of U will change radically along the length of the tube. In this thesis, a distributed heat transfer model, where each heat transfer mode (desuperheating, condensing and subcooling) is accounted for separately is implemented to help account for the radical change in thermophysical properties.

4.1.2 A Refrigerant Circuit

For the purposes of heat exchanger modeling, a refrigerant circuit is defined here as the flow path of refrigerant from an entrance to an exit. A refrigerant circuit may branch off into two or more paths, re-combine and/or exit at one or more locations. The refrigerant circuit is simply all tubes in hydraulic contact with one another between inlet and outlet. Figure 4-1 illustrates a possible complex refrigerant circuit.

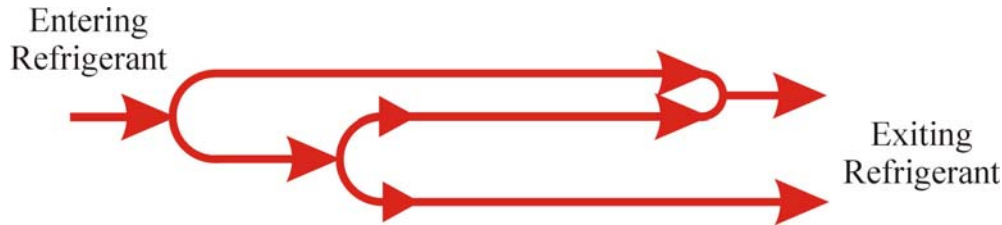


Figure 4-1: A Refrigerant Circuit

4.1.3 An Air Circuit

The tubes modeled earlier in this are prescribed a certain air velocity, or flow rate. In the tube models, air properties are attributes of the tube itself. When given the appropriate air boundary conditions, inlet temperature and flow rate, the outlet air temperature is calculated. Realistically, the air flow is an energy system distinct from the refrigerant. Although in thermal contact with the refrigerant circuit(s), the air flows along its own path, and is therefore hydraulically independent from a tube.

For simplicity in this thesis, the equations governing the hydraulic flow of the air are ignored, and a flow rate is imposed upon the air circuit. Consequently, continuity is implicit. Figure 4-2 illustrates a single air circuit flowing over three refrigerant segments.

In this particular case, the air is flowing in the +y direction, over three separate tube segments, where the refrigerant flows along the x-axis. Note there is no assumption that the tube segments be part of the same refrigerant flow stream.

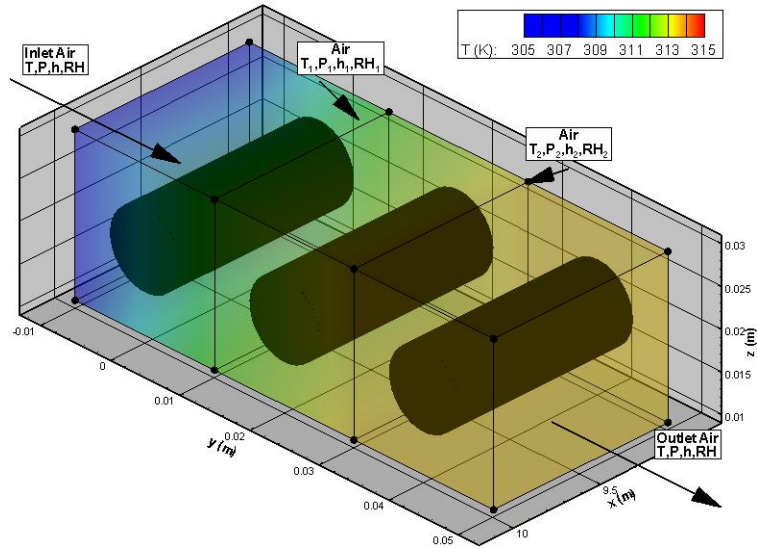


Figure 4-2: An Air Stream: Air Flows from Left to Right, Through Three Air Segments, Simulating Air Flow Over Three Tube Segments

Comment: In figure caption, explain in more detail the geometry.

Each air circuit has one or more tube segments over which it flows. The three individual “cells” are air segments, combined to make the single air stream.

Using the energy equation given in Table 2.1, an energy balance, the governing equation of energy for the air stream is:

$$\dot{m}(h_{air,out} - h_{air,in}) + \dot{Q} = 0 \quad (4.19)$$

$$\dot{Q} = \sum_{i=1}^{NSegments} \dot{Q}_i = \sum_{i=1}^{NSegments} -\dot{Q}_{i,Ref}$$

In the case where moist air has water vapor condense on the tube segment, conservation of species (water vapor/liquid water) is no longer implicit, and species balances must also be included. The mass and energy equations for moist air condensing on heat exchanger segments are (McQuiston and Parker, 1988):

$$\begin{aligned}
\dot{m}_{air}(\omega_{out} - \omega_{in}) &= \dot{m}_{H_2O,out} \\
\dot{m}_{air}(h_{air,out} + \omega h_{H_2O,out} - h_{air,in} - \omega h_{H_2O,in}) + \dot{Q} + \dot{Q}_{fg} &= 0 \quad (4.20) \\
\dot{Q} = \sum_{i=1}^{NSegments} \dot{Q}_i &= \sum_{i=1}^{NSegments} -\dot{Q}_{i,Ref}
\end{aligned}$$

Recalling that junctions have no interaction with the fluid, the junction energy balance is:

$$\begin{aligned}
\dot{m}_{air}(h_{air,out} + \omega h_{H_2O,out} - h_{air,in} - \omega h_{H_2O,in}) &= 0 \\
\text{Or, by defining: } \bar{h} &= h_{air,out} + \omega h_{H_2O,out} \quad (4.21) \\
\bar{h}_{out} - \bar{h}_{in} &= 0
\end{aligned}$$

4.2 The Actual Heat Exchanger

The actual heat exchanger modeled is a cross flow tube-fin heat condenser. Three distinct circuits enter the heat exchanger, and combine into a single stream, which serves as a sub-cooler.

Figure 4-3 illustrates the configuration of the heat exchanger modeled in this chapter, while the physical parameters for the heat exchanger are given in Figure 4-4.

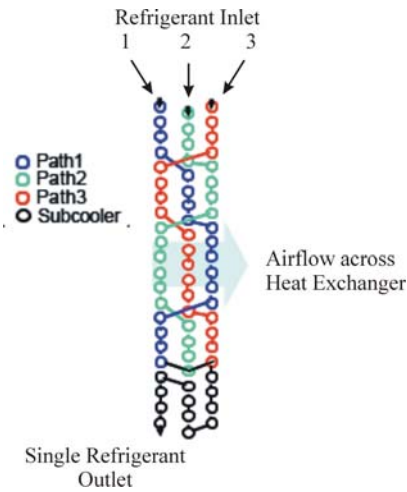


Figure 4-3: Cross Section of Heat Exchanger Configuration. Air Flows Across Banks of Tubes from Left to Right

Specification		Condenser
Dimension	W x H x D [cm]	99 x 70 x 8
	Frontal area [m ²]	0.69
Air flow	Air flow rate [m ³ /s]	1.6
	Frontal air velocity [m/s]	2.35
Fin	Shape	Wavy
	Fin pitch [mm]	2.1
	Thickness [mm]	0.13
Tube	No. of row	3
	No. of tubes per each row	22
	Tube diameter [mm]	9.52
	No. of circuit	3 (R-404A, R-290), 2 (R-410A)
	Tube shape	Inner grooved

Figure 4-4: Heat Exchanger Physical Parameters

The condenser given in (Hwang, et. al 2004) was part of a system, and thus subject to various system test points. Table 4-1 details the operating point of the condenser under the various system test points for the R-404a case. Similarly, Table 4-2 details the operating point of the condenser for the various system test points for the R-290 case.

Test Parameter	Test1	Test2	Test3	Test4
Pin (kPa)	2422.5	2515.1	2709.6	1743.3
Tin (K)	353.0	355.2	360.8	337.0
Pout (kPa)	2285.5	2384.9	2589.3	1606.9
Subcooling (K)	8.2	13.5	17.8	12.0
Air Inlet Temperature (K)	308.7	308.3	308.3	291.5
Mass flow rate (g/s)	100.0	99.9	98.5	100.0
Q (kW)	16.04	16.72	17.02	17.65

Table 4-1: Test Data for R-404a Condenser

Test Parameter	Test1	Test2	Test3	Test4	Test5
Pin (kPa)	1764.9	1786.1	1820.8	1877.3	1238.9
Tin (K)	353.3	354.2	355.1	357.7	331.2
Pout (kPa)	1667.0	1695.4	1735.5	1795.2	1130.8
Subcooling (K)	8.1	10.2	12.5	15.4	6.1
Air Inlet Temperature (K)	308.0	307.9	307.9	307.8	292.0
Mass flow rate (g/s)	41.7	41.6	41.7	41.2	42.7
Q (kW)	15.89	16.1	16.35	16.45	16.77

Table 4-2: Test Data for R-290 Condenser

4.3 The Heat Exchanger Model

A heat exchanger is constructed by combining refrigerant circuits, as defined in section 4.1.2, with air streams, as defined in section 4.1.3. The two energy systems are then coupled by associating the air properties of individual air segments with the air properties of the appropriate refrigerant segment.

The heat exchanger represented by Figure 4-3 and Figure 4-4 is modeled here in with a slight geometric modification. For a given refrigerant circuit path, when the circuitry snakes back upon itself, while staying in the same plane, perpendicular to the air flow, the tubes in that plane are combined into a single tube, thus combining multiple “passes” into a single “pass”. This process is demonstrated in Figure 4-5, where the total number of tubes is reduced from 66 to 15. As a consequence, a “pseudo length factor” (PLF) is introduced. This factor multiplies a given tube’s surface area and length for the purposes of calculating heat transfer and pressure drop for individual tubes.

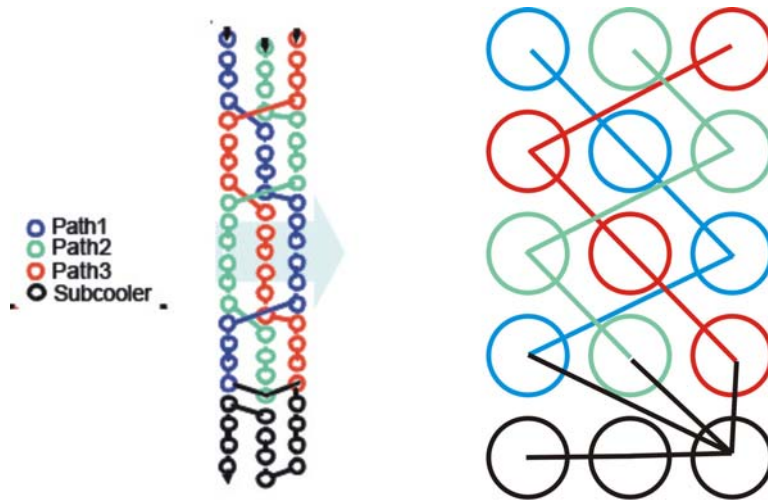


Figure 4-5: Geometry Modification For Heat Exchanger Model. The Real Condenser on the Left is Approximated with Circuitry on Right

Some factors which may contribute to the inaccuracy of the model to represent the actual heat exchanger include:

- The reduction of many tube passes down to a single tube pass
- The use of a single segment to represent a relatively long length of tube (0.99 m) with lumped pressure drop and heat transfer correlations
- The possibility of the flow characteristics being beyond the acceptable range for the given correlations

As the scope of this work is more the use of the heat exchanger to validate a simulation framework, rather than an accurate heat exchanger model, three additional correction factors (CF) are introduced to the model, for the purposes agreement between experimental data and the model. Equations (4.22) - (4.25) show how the *pressure drop correction factor* (CF_{PD}), *liquid heat transfer correction factor* (CF_{LHT}) and *two phase heat transfer correction factor* (CF_{TPHT}) are used in the modeling equations.

$$\Delta P \approx 0.158L PLF CF_{PD} \rho^{3/4} \mu^{1/4} d^{-5/4} V^{7/4} \quad (4.22)$$

$$h_{Liquid} = CF_{LHT} Nu_d k / d \quad (4.23)$$

$$h_{TwoPhase} = CF_{TPHT} Nu_d k / d \quad (4.24)$$

$$NTU = \frac{UA}{c_1} = \frac{UA PLF}{(\dot{m}c_p)_1} \quad (4.25)$$

For the purpose of modeling the heat exchanger, the values for the correction factors are determined using a design test point (test 1) from Table 4-1 and Table 4-2. The values are determined by solving the following equations:

$$\dot{m}_{calc} - \dot{m}_{test} = 0 \quad (4.26)$$

$$h_{out,calc} - h_{out,test} = 0 \quad (4.27)$$

$$h_{subcooler\ inlet, calc} = h_{Sat} \quad (4.28)$$

Equations (4.26) - (4.27) represent an agreement between experimental and simulated mass flow rate and outlet enthalpy (thus heat transfer), while equation (4.28) attempts to meet a design criteria that the final three tubes in the refrigerant circuit be used as a sub-cooler circuit.

Results for the correction factors with the condenser using R-404a are given in Table 4-3, while results for the condenser using R-290 are given in Table 4-4.

CF _{PD}	CF _{LHT}	CF _{TPHT}
4.25	0.7	0.9

Table 4-3: Correction Factors for Modeled Heat Exchanger (R-404a)

CF _{PD}	CF _{LHT}	CF _{TPHT}
6.0	0.2	2.5

Table 4-4: Correction Factors for Modeled Heat Exchanger (R-290)

The relatively high value of the correction factors indicate that the modeling assumptions and correlations used are not the best for simulating this particular heat exchanger. They do however allow this model to accurately simulate this heat exchanger at the design point, and allow for the model to represent the actual heat exchanger in the vicinity of the design operating point.

4.3.1 Wrapping the Junction Solver

The Junction Solver is a non-hierarchy system, meaning the sub-systems; in this case, tubes are not allowed to communicate. Consequently, the coordinating system, in this case, the heat exchanger itself, must facilitate communication between sub-systems, (components) such that convergence criteria are met.

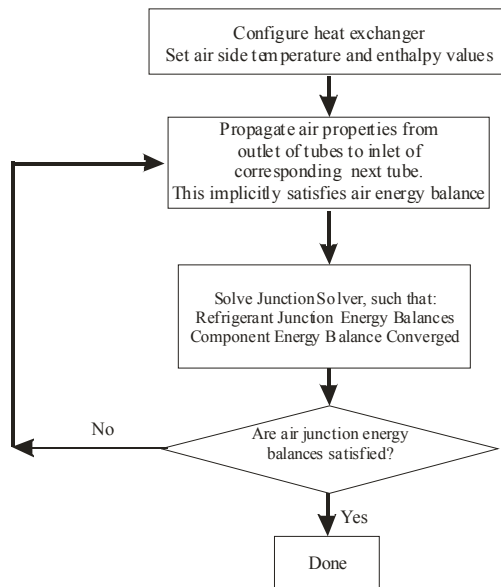


Figure 4-6: Heat Exchanger Solution Flow Chart

Figure 4-6 illustrates the process of wrapping the Junction Solver within a routine which updates the heat exchanger air side. The Junction Solver is responsible for

resolving the refrigerant side, while the coordinating heat exchanger is responsible for providing air property values (the tube external boundary conditions) to the Junction Solver sub-systems.

4.3.2 Seeding the Junction Solver for the Heat Exchanger

The Junction Solver can be filled with any potential solution. When no a priori guess value is known, the junction solver initializes to a linear pressure distribution and a constant thermal distribution.

The algorithm for determining the linear junction pressure distribution is given in Figure 4-7.

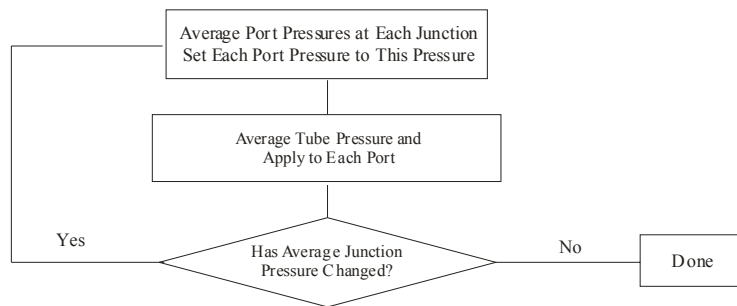


Figure 4-7: Algorithm for a Linear Pressure Distribution

The value for the junction enthalpies is set to the heat exchanger inlet enthalpy. The air temperature is set to the heat exchanger inlet air temperature at all air junctions.

4.3.3 Scaling Parameters for the Model

Recall, that for conditioning purposes, the Junction Solver has the parameters built in for scaling: P_{high} , P_{low} , h_{high} , h_{low} , \dot{m}^* , \dot{E}^* . Recall the junction solver scales independent variables in the form of

$$P_i^* = \frac{P_i - P_{low}}{P_{high} - P_{low}} \quad (4.29)$$

$$h_i^* = \frac{h_i - h_{low}}{h_{high} - h_{low}} \quad (4.30)$$

Similarly, the Junction Solver balance equations are scaled as

$$r_i = \frac{\sum_{p=1}^{Nports} \dot{m}_p}{\dot{m}^*} \quad (4.31)$$

$$r_j = \frac{\sum_{p=1}^{Nports} \dot{m}_p (h_p + \frac{\mathbf{v}_p^2}{2} + g z_p)}{\dot{E}^*} \quad (4.32)$$

For heat exchanger simulation, the values of P_{high} and P_{low} come from the heat exchanger boundary pressures, specifically the inlet and outlet pressure respectively. For a condenser, the refrigerant inlet enthalpy is the highest value, and represents h_{high} , while h_{low} is calculated as the minimum enthalpy the refrigerant can achieve, that which corresponds to the air inlet temperature at the heat exchanger outlet pressure. As the heat exchanger models are designed after a specific heat exchanger, values for mass flow and heat of rejection at the design point are used for \dot{m}^* , \dot{E}^* . Taking the Test 1 condition from Table 4-1 the independent variables are scales as

$$P_i^* = \frac{P_i - P_{low}}{P_{high} - P_{low}} = \frac{P_i - 2285.5}{2422.6 - 2285.5} \quad (4.33)$$

$$\begin{aligned} h_{high} &= h(P_{in} = 2422.6, T_{in} = 352.95) = 419.830 \text{ kJ / kg} \\ h_{low} &= h(P_{out} = 2285.5, T_{air,in} = 308.65) = 250.741 \text{ kJ / kg} \\ h_i^* &= \frac{h_i - h_{low}}{h_{high} - h_{low}} = \frac{h_i - 250.741}{419.830 - 250.741} \end{aligned} \quad (4.34)$$

While the residual equations are scaled as

$$r_i = \frac{\sum_{p=1}^{Nports} \dot{m}_p}{0.100 \text{ kg / s}} \quad (4.35)$$

$$r_j = \frac{\sum_{p=1}^{N_{ports}} \dot{m}_p h_p}{16.04 kW} \quad (4.36)$$

Recall from Figure 4-6 that a fractional step solution is employed. In this particular case, a simultaneous solution of the hydraulic equations is utilized, while a successive substitution routine is employed for the thermal step. Conditioning of the equations for use in iterative linear algebra routines is therefore only necessary for the hydraulic step.

Figure 4-8 illustrates the effect the scaling of the Junction Solver independent variables and residual equation has on the solution process. Each curve in Figure 4-8 represents an individual hydraulic step of the algorithm shown in Figure 4-6, which itself has one or more iterations to solve.

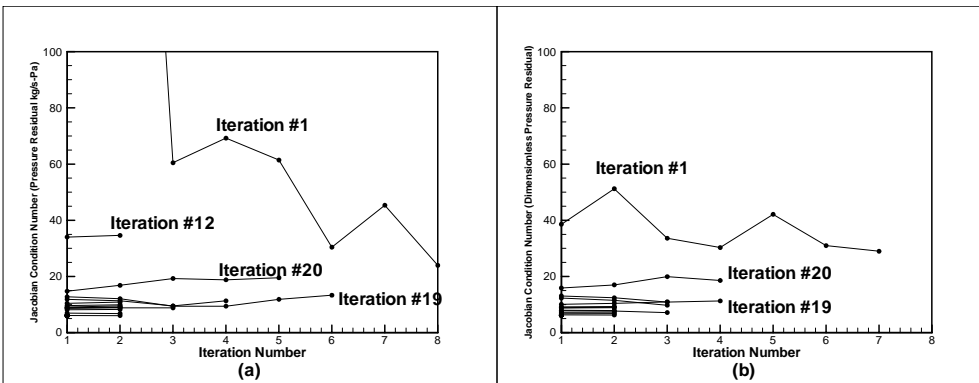


Figure 4-8: Condition Number of Jacobian Matrix versus Hydraulic (Inner) Iteration Number. The Iterative Hydraulic Routine is Wrapped in a Successively Substituted Outer Thermal Routine. Scaling of the Independent Variables and Residual Equations Reduces the Condition Number of the Jacobian Matrix of the Hydraulic Routine.

Comment: This needs more detail. What does "Inner" mean? Or refer to text in figure caption.

Figure 4-8 shows curves representing the condition number of the Jacobian matrix used for solving the pressure calculations in the fractional step technique. Each curve represents the value of the Jacobian condition number at individual iterations during the solution of the pressure routine.

Figure 4-8(a) has curves showing the condition number of the Jacobian when unscaled independent variables and residual equations are used, while the curves shown in Figure 4-8(b) has curves representing the Jacobian when independent variables and residual equations are scaled in the manner given in equations (4.33) and (4.35).

It is seen that due to the nature of the equations, most of the pressure calculations are converged to tolerance within two iterations.

It is evident from the plots that performance improvements are seen. For the scaled variables and equations, the initial iteration has a reduced condition number, and completed in one less iteration. Improvements in later iterations are seen, as after the first iteration, no pressure routine for the scaled variables and equations requires more than four iterations, where some of the iterations for the unscaled variables and equations do.

4.4 Validation

For verification, the heat exchanger model is run for the R-404a “test 1” case, as given in Table 4-1. Graphical output is used to verify that the model is indeed behaving in a manner similar to that of a real heat exchanger.

For test purposes, convergence tolerances are set to 0.01 W and 10^{-8} kg/s for the heat exchanger simulation. Plots of the heat exchanger thermophysical properties are made to verify that the model is solved properly, when convergence occurs.

4.4.1 Thermophysical Property Distributions

Figure 4-9 displays the refrigerant pressure distribution and quality distribution. The refrigerant pressure and quality behave in a predictable manner. Figure 4-9(a) illustrates

a gradual decrease in refrigerant pressure along the flow path(s), while Figure 4-9(b) shows a steady drop in the refrigerant quality, along the condenser flow path(s).

Figure 4-10 displays the temperature distribution for the heat exchanger. The temperature distribution of the heat exchanger is as expected. Figure 4-10(a) shows the refrigerant cooling off along the length of the heat exchanger path(s). Due to the condensation, the temperature remains near constant, while it condenses. The figure is consistent with Figure 4-9(b), in that the nearly constant temperature section corresponds to the sections of the heat exchanger where the refrigerant quality is greater than 0, but less than 1. Figure 4-10(b) (note the heat exchanger is viewed from the opposite side) illustrates the air temperature distribution superimposed upon the refrigerant tubes. In this figure, air is flowing from left to right, and the temperature rises along the air flow path.

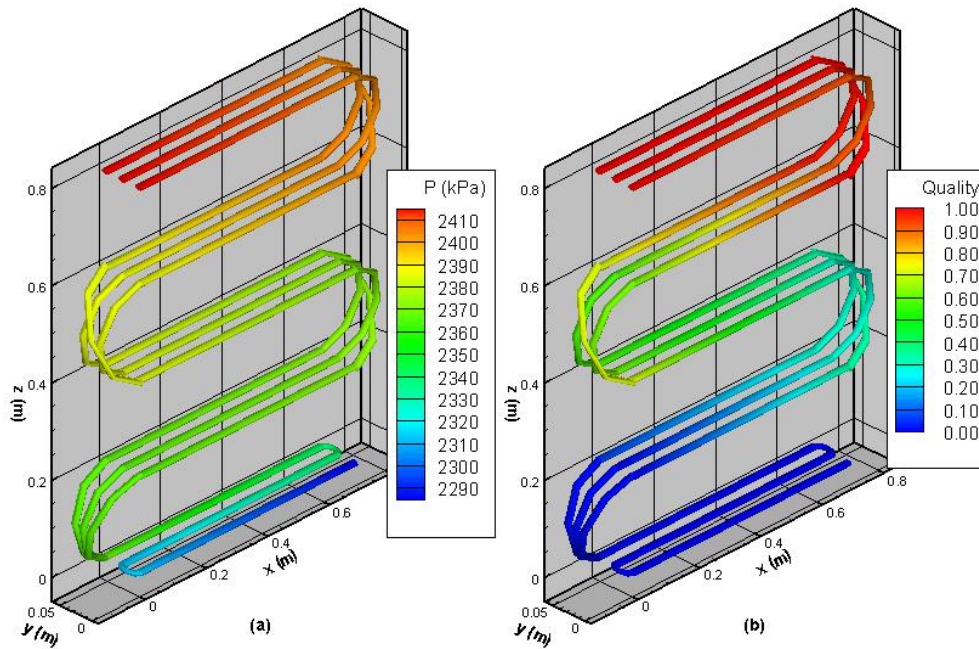


Figure 4-9: Refrigerant (a) Pressure and (b) Quality Distribution. Air Flow, Not Shown In the Figure, Flows from Left To Right Over the Tube Banks, Cooling Them.

Comment: Better show airflow too

4.4.2 Mass Balances and Energy Balances

Figure 4-11 illustrates the refrigerant junction and tube mass and energy balances.

Figure 4-11(a) illustrates that at convergence, the junction and tube mass balances are less than the convergence criterion 10^{-8} kg/s (in absolute value). Similarly, Figure 4-11(a) illustrates that the refrigerant tubes and junction energy balances are less than the specified 0.01 W.

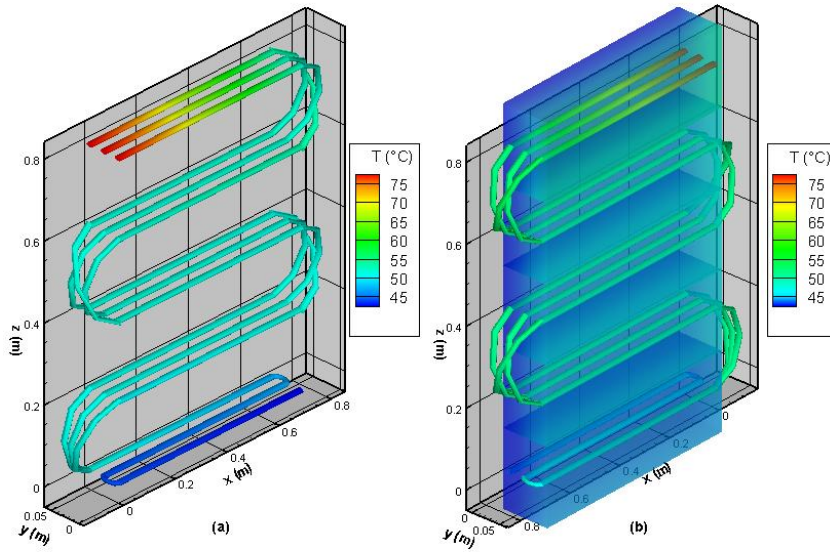


Figure 4-10: Heat Exchanger Temperature Distribution –
 (a) Refrigerant Only and (b) Refrigerant and Air

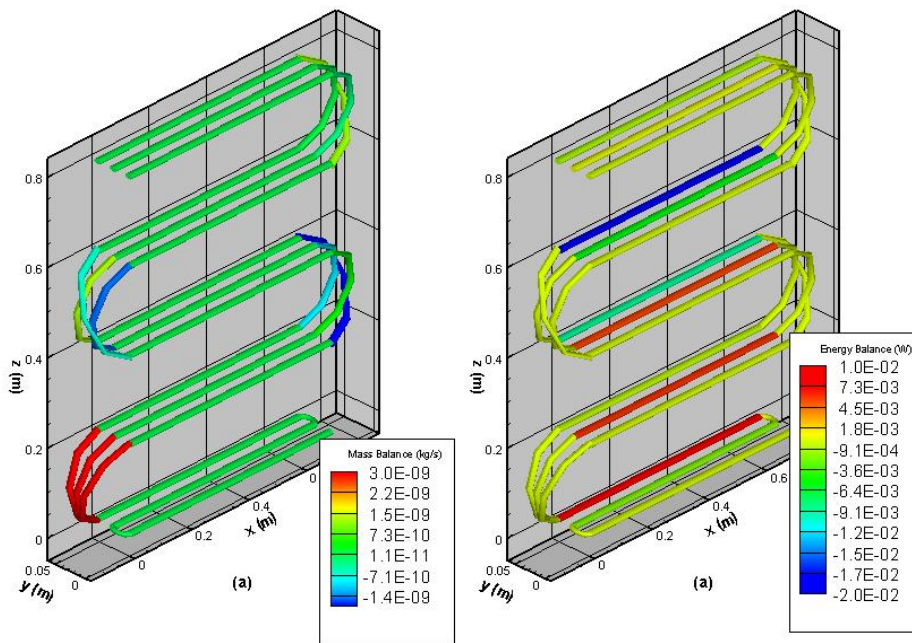


Figure 4-11: Heat Exchanger (a) Mass Balance and (b) Energy Balance

4.5 Verification

The performance results predicted by the simulation are compared against the experimental data provided and displayed in Table 4-1, Table 4-2, and Figure 4-12 - Figure 4-17. Table 4-1 gives a numerical comparison of the R-404a condenser, while Table 4-2 gives a numerical comparison of the R-290 condenser. See Hwang et al (2004), for a detail on the actual experiment.

The experimental data represents the single heat exchanger geometry under a variety of test conditions, with two working fluids. Given the specified boundary conditions on the condenser, namely inlet and outlet pressures, refrigerant inlet state, air inlet temperature and flow rate, the relevant performance properties, which correspond to model outputs are: refrigerant mass flow rate, heat rejection and refrigerant exit thermodynamic state represented by the degree of subcooling.

Figure 4-12 graphically displays the comparison of simulated to measured mass flow rate for the R-404a condenser for the four test points. The first three test points, which correspond to the same inlet air temperature, but different refrigerant pressures and inlet energy state have good agreement (within 5%) to the test data.

The final test point, which corresponds to a test point with a lower inlet air temperature, shows a deviation of approximately 6%.

Figure 4-13 shows that under all test points, the heat of rejection reported by the simulation is within 5%.

Figure 4-14 illustrates agreement within 5% for all test points except low inlet air temperature test point, which deviates by approximately 11%.

Figure 4-15 graphically displays the comparison of simulated to measured mass flow rate for the R-404a condenser for the four test points. The first three test points, which correspond to the same inlet air temperature, but different refrigerant pressures and inlet energy state have good agreement (within 5%) to the test data. The final test point, which corresponds to a test point with a lower inlet air temperature, shows a deviation of approximately 10%.

Figure 4-16 shows that under all test points, the heat of rejection reported by the simulation is within 5%.

Figure 4-17 illustrates subcooling agreement within 5% for all test points except low inlet air temperature test point, which deviates by approximately 11%.

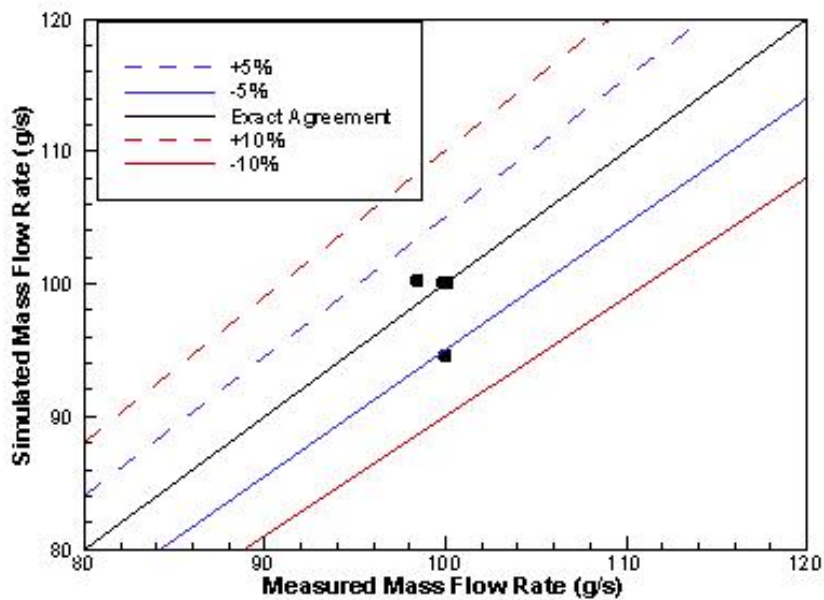


Figure 4-12: Mass Flow Rate Comparison for R-404a Condenser (See Table 4-1)

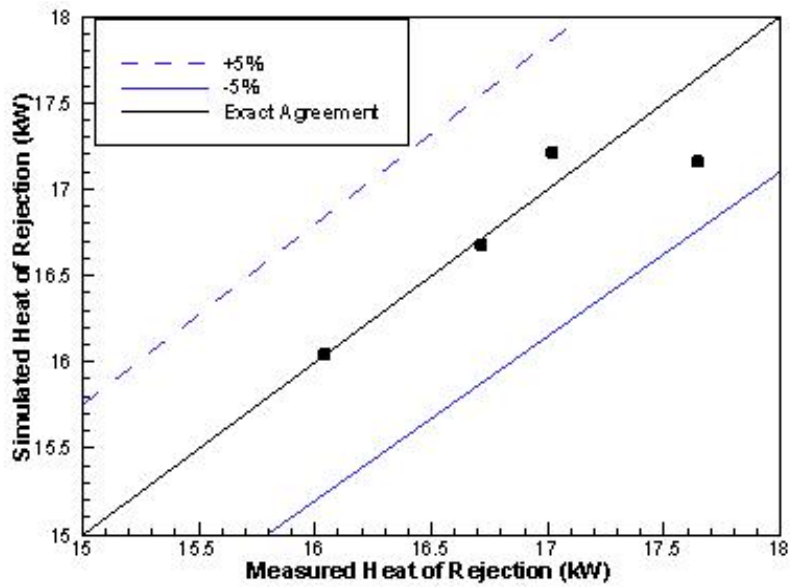


Figure 4-13: Heat of Rejection Comparison for R-404a Condenser (See Table 4-1)

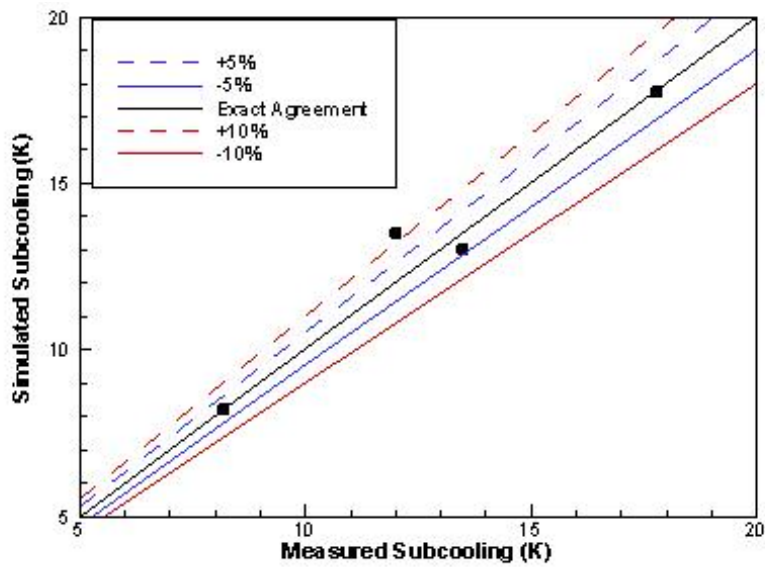


Figure 4-14: Subcooling Comparison for R-404a Condenser (See Table 4-1)

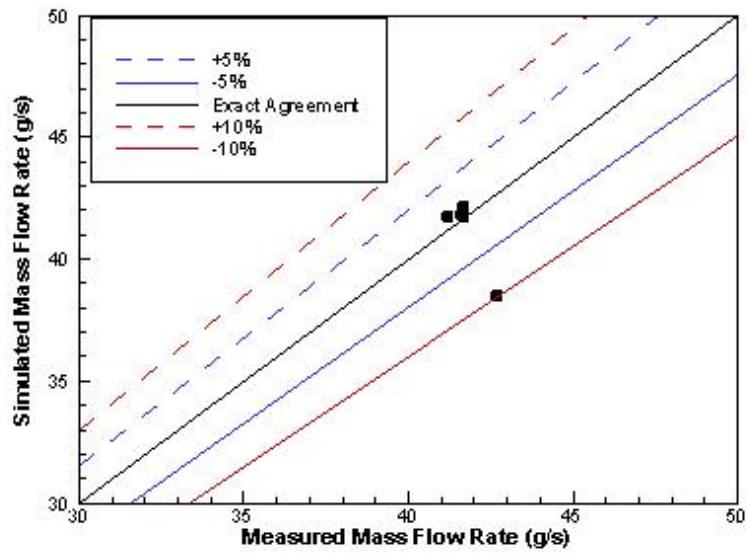


Figure 4-15: Mass Flow Rate Comparison for R-290 Condenser (See Table 4-2)

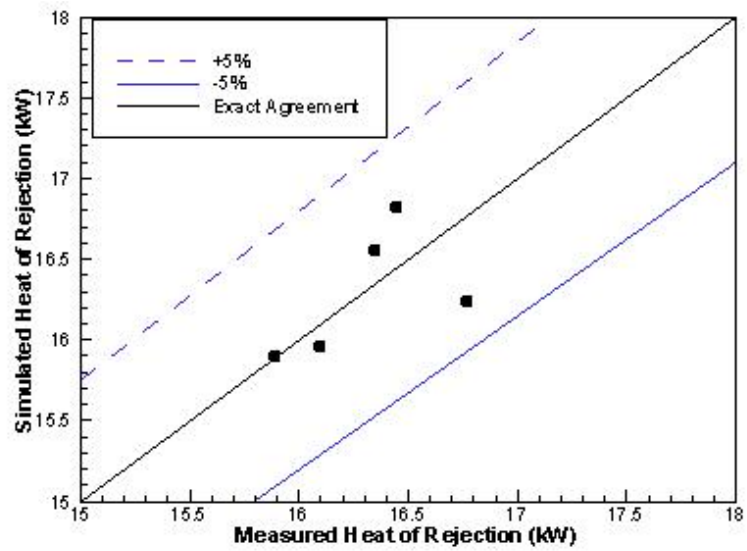


Figure 4-16: Heat of Rejection Comparison for R-290 Condenser (See Table 4-2)

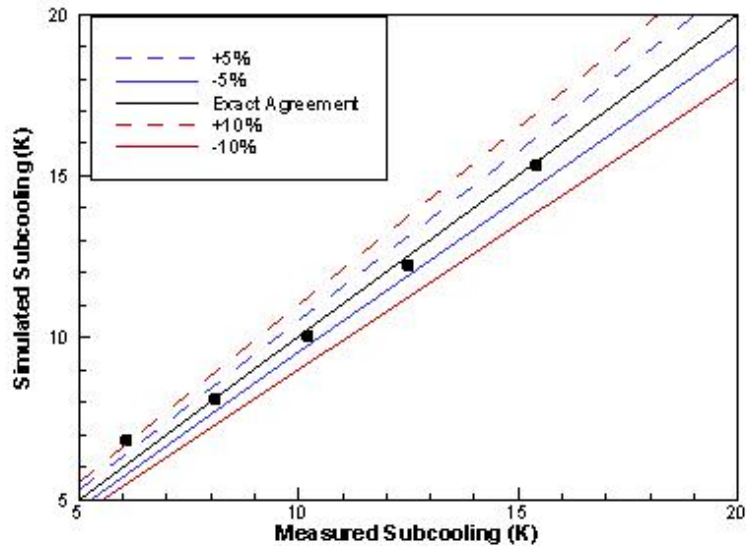


Figure 4-17: Subcooling Comparison for R-290 Condenser (See Table 4-2)

4.6 Summary

In this chapter, the secondary objective of the dissertation, the development of a heat exchanger model built upon the Junction Solver is accomplished. The model facilitates (a) validation of the Junction Solver as a general purpose thermo fluid simulation tool, and (b) a heat exchanger model for use in VapCyc to facilitate its validation.

In this chapter the necessary simulation models of tube and air segment are combined via the Junction Solver creating a condenser, which is suitable for use in VapCyc. The actual heat exchanger models are designed to represent physical pieces of hardware which have been instrumented and tested in the laboratory (Hwang, et al. 2004).

The tube models allow for three correction factors (pressure drop correction factor (CF_{PD}), liquid heat transfer correction factor (CF_{LHT}) and two phase heat transfer correction factor (CF_{TPHT})), which are determined numerically to match a specific test case for the heat exchanger, for each of the two tested fluids.

Due to the factors, the simulation data produced from the simulation results matches the experimental data to within a few percent for flow rate and heat of rejection.

This chapter achieves the goal of validating the Junction Solver as a general purpose thermo fluid simulation tool by its success in the application of modeling a specific thermo fluid system, namely an air cooled condenser. In addition, the R-134a and R-290 condensers are general purpose components, suitable for use in VapCyc, another specific application of the Junction Solver.

4.7 References

Chang, Y., Wang, C., 1997, "A Generalized Heat Transfer Correlation for Louver Fin Geometry", *International Journal of Heat and Mass Transfer*, Vol.40, pp. 533- 544.

Dittus, S.J., Boelter, L.M.K., 1930, "Heat Transfer In Automobile Radiators Of The Tubular Type", University of California Publications on Engineering, Vol.2, pp.443-461.

Hwang, Y., Dea-Huan, J., Radermacher, R., "Comparison Of Hydrocarbon R-290 And Two Hfc Blends R-404a And R-410a For Medium Temperature Refrigeration Applications", *ARI: Global Refrigerant Environmental Evaluation Network (GREEN) Program*, Final Report, March 2004

Kakac, S., Liu, H., 2002, *Heat Exchangers: Selection, Rating, And Thermal Design*, 2th edition, CRC Press.

Kim, N.H., Young, B., Webb, R.L., 1999, "Air-Side Heat Transfer and Friction Correlations for Plain Fin-and-Tube Heat Exchangers with Staggered Tube Arrangements", *Journal of Heat Transfer, Transactions of ASME*, Vol.121, pp.662-667.

Nash, G. S., and Sofer, A., 1996. *Linear and Nonlinear Programming*, McGraw –Hill publishing

Shah, M.M., 1989, "A General Correlation for Heat Transfer during Film Condensation inside Pipes", *International Journal of Heat and Mass Transfer*, Vol.22, pp.547-556.

White, F. M., *Fluid Mechanics*, Second Edition, McGraw-Hill, 1986

5. VapCyc – Vapor Compression Refrigeration System

To this point, this dissertation has focused on the creation of a generalized thermo fluid energy system simulation tool, the Junction Solver. The power of this simulation tool is realized when sub-systems of different character are combined into a larger system.

One objective of this section is the development of a general-purpose comprehensive vapor-compression refrigeration simulation tool, VapCyc, for analysis and design of the vapor-compression system and its components. Building VapCyc upon the Junction Solver allows for the following objectives to be met:

- The ability to constrain and specify the mass of working fluid for the system simulation. For a vapor-compression refrigeration application, this is referred to as “Charge Management”.
- The functionality to change, or swap, component models used in a fixed-component simulation at run-time thus replacing components without the need to re-compile the simulation code itself.
- The functionality to add or subtract components at run time, without the need to re-compile the simulation code itself.
- A (limited) user defined component connectivity.
- The selection from several different working fluids/refrigerants.
- Flexibility to alter component independent variables, independently from the main simulation program.
- Incorporation of both “interactive” and “automated” modes of operation

- Off design simulation

This chapter presents the VapCyc tool, as a specific implementation of the Junction Solver. Examples are used to validate and verify against data from a research project in the CEEE heat pump laboratory (Hwang et al. 2004), as well as demonstrate the overcoming of the numerical issues of conditioning and providing a necessary initial guess. This method of providing an initial guess to the system solution is provided via the introduction of a novel concept called the “context”.

5.1 VapCyc Component Specifications

Section 3.2 detailed component specification for use within the Junction Solver framework. VapCyc is a simulation which incorporates the Junction Solver. While the components used within VapCyc *must* have all the specifications of a component used within the Junction Solver, VapCyc requires specifications beyond those of the basic Junction Solver specification. For example, since VapCyc is a vapor compression refrigeration simulation, it must report the system’s coefficient of performance (COP). This is calculated by dividing the system’s cooling capacity by the power consumption. Thus all VapCyc components must provide both their cooling capacity (typically only the evaporator would have a non-zero value) and their power consumption.

The object oriented manner in which components are designed allows application programs, such as VapCyc to incorporate component properties which are non-traditional in standard simulations. For example, a component property, “Lead Time”, gives an idea of how quickly a system could be built. Utilizing the properties of “gravity” (gravitational constant) and component “mass” allows the study of systems in different applications, such as earth and in space.

Table 5-1 illustrates the component properties that all VapCyc components are required to provide, for this introductory version of VapCyc. The manner in which VapCyc is written allows additional properties to be added later.

Property	Description
Charge	The mass contained within the component at the current operating point.
Cost	The cost to purchase/manufacture the component
Gravity	The value of the gravitational potential experienced by the component
HeatOut	The energy rejected to the environment via heat transfer at the current operating point
Lead Time	Time (in weeks) required to obtain/manufacture the component
Manufacturability	A numeric value representing the ability/cost to manufacture the component
Mass	The mass of the component (devoid of working fluid)
Port Enthalpy(p)	The enthalpy of the working fluid, at port p, at the current operating point
Port Massflow(p)	The mass flow of the working fluid, at port p, at the current operating point
Port Pressure(p)	The pressure of the working fluid, at port p, at the current operating point
Port Velocity(p)	The velocity of the working fluid, at port p, at the current operating point
Port Position(p)	The position in a gravitational field of the working fluid, at port p, at the current operating point
Power Consumption	Power consumed by the component at the current operating point
Refrigerant	The working fluid
Reliability	A number representing the reliability of the component at the current operating point
SystemUnits	The unit system which the component must comply with when receiving/giving numeric values
Volume	Volume of the component
WorkOut	The energy rejected to the environment via work (rate) at the current operating point

Table 5-1: VapCyc Component Properties

5.2 The Test System

Due to growing environmental awareness and resulting concerns, refrigerants, the working fluids for refrigeration systems, heat pumps and air conditioners, have attracted considerable attention. Following policies to reduce global warming, industry is developing technologies that can reduce emissions and improve energy efficiency (Hwang, et al. 2004). The scope of the project is to evaluate the performance of a medium temperature refrigeration unit with respect to the working fluid, R-404a, R-290 and R-410a.

This test project is a useful vehicle for the validation and verification of VapCyc. Rather than completely reproduce the results of the test project in its entirety, only the R-404a and R-290 test results are reproduced.

The tests performed for the test project include a “charge optimization” which includes the design point of the systems, and a “part load” test. The charge optimization consists of a fixed system configuration with environmental boundary conditions (air conditions at both the condenser and evaporator) are held constant while the refrigerant charge is varied to maximize the system COP. The part load test consists of the fixed system configuration at the “optimum” charge, evaluated at a second set of air conditions.

Appendix I contains the data points taken from the test project, as well as the process for the creation of the TEST components. The TEST components are used in VapCyc to verify the simulation against the test data.

5.3 Simulation of Vapor Compression Refrigeration

Vapor compression (v-c) refrigeration represents a large part of the United States energy consumption, and therefore a practical target for energy savings.

Experimental analysis of v-c refrigeration is difficult, as even a simple system can have many independent variables; making complete control of the system during experimentation difficult if not impossible. Logically, simulation is a good vehicle to analyze system performance over a rigidly controlled set of independent system variables. Simulation also offers an opportunity to save time and effort when optimization of a system is the goal.

The study of vapor compression refrigeration has been facilitated by many computer simulations created over the years, dating back to approximately 1976. These simulations are created for the purposes of design and analysis, but rarely optimization of the system. The large majority of the simulations created are very specific to a single system configuration. Typically, to make a very specific simulation more flexible a large number of independent variables are allowed in the simulation.

The literature demonstrates moderate efforts to utilize optimization in vapor compression refrigeration at the system level. Previous optimization work focuses on (a) simulation programs designed for and integrated with optimization routines and (b) optimization results obtained from manually retrieving data from simulation programs. In previous work, programs designed for optimization (Rice, et al. 1981, Levins et al. 1996) are limited in nature due to the complex nature of the optimization/simulation integration. Manual optimization techniques (Fischer and Rice, 1986) are (i) tedious to perform and (ii) not conducive to expansion. Thus, an opportunity exists to create a

simulation structured such that a great deal of the functionality of the optimization routine is determined at run-time, thus separating the simulation and optimization routines, adding flexibility and utility to the optimization process.

5.4 Premise

VapCyc is a tool for a steady-state vapor compression refrigeration system simulation. The simulation is designed to be comprehensive and robust, as the goal is its utilization in optimization routines.

VapCyc is built around four independent component models, namely compressor, condenser, expansion device and evaporator. VapCyc simulates the performance of systems comprised of at least this basic set. The simulation consists of a set of independent variables, such as: system charge, component model numbers and component independent variables, and results in a set of dependent system variables, such as: COP, capacity, weight and volume. Using the general system simulation modeling technique outlined in chapter 2, VapCyc also facilitates the insertion of additional components, beyond the four basic components into the system. The additional components can be positioned in parallel; between any two junctions, or in series; between any two components.

Although the basic configuration of the system is fixed, the models and independent variables, the boundary conditions, for each component can be variable. Once the individual component models, the component boundary conditions and a value for the refrigerant charge are chosen, the complete set of independent variables for the system is specified. The simulation is achieved through the solution of the system level conservation laws of mass, and energy, as described in chapter two. The evaluation of

the components subject to their boundary conditions, allows for the solution of the system level conservation laws.

Component inter-changeability, coupled with a solver for the run-time determined set of independent variables is achieved through a two level object oriented scheme. The system level object is responsible for resolving the system conservation laws. The component level objects are responsible for the reporting of their mass flow, refrigerant charge and energy interactions with their environment to the system level when given boundary conditions.

5.5 VapCyc's Component Library

5.5.1 A Generic Compressor Model

A simple model for a compressor accounts for all compressor workings with two compressor properties; namely an isentropic efficiency and a volumetric efficiency. When subject to thermodynamic boundary conditions, the volumetric displacement per revolution and compressor speed, the operating point of the compressor is determined. In this context, the operating point refers to the flow rate through the compressor (both mass and volumetric flow rates), as well as the thermodynamic state of the refrigerant at the compressor discharge port.

The model inputs are,

- Volumetric Efficiency, $\eta_v = \frac{\dot{m}_{actual}}{\dot{m}_{max}} = \frac{\dot{m}_{actual}}{\rho V RPM}$
- Isentropic Efficiency, $\eta_s = \frac{\dot{W}_{Isentropic}}{\dot{W}_{Actual}}$
- RPM (Compressor speed in revolutions per minute)

- Displacement Volume (per revolution, V)

The results of the model, which solves for mass flow rate and outlet enthalpy are:

$$\begin{aligned} \dot{m} &= \rho_m V \text{ RPM } \eta_v \\ h_{out} &= h_{in} + \frac{(h_s - h_{in})}{\eta_s} \end{aligned} \quad (5.1)$$

The model is now ready for a general energy system simulation as described in chapter 2. With the energy system providing thermodynamic boundary conditions to the model, the compressor is able to provide a mass flow at each of its ports, and a value for the enthalpy at each of its outlet ports.

It is stressed here that this generic compressor model is a specific compressor model designed to execute quickly, to facilitate the validation of VapCyc. Although this model is fairly simply in nature, models of any level of sophistication can be used within VapCyc.

5.5.2 The Alpha Compressor

VapCyc is equipped with a library of components, which have a range of nominal capacities at the VapCyc air-conditioning context. The components are created in families, where a family of a certain component has the same operating characteristics as all others in the family. For example, using the techniques of section 5.5.1 a family of compressors called the Alpha Compressor was created. The Alpha Compressor is based upon the Generic Compressor model of section 5.5.1. The model has four independent variables, namely displacement per revolution, speed (rev/min), volumetric efficiency (η_v) and isentropic efficiency (η_s). The Alpha Compressor Family fixes RPM at 1000 rev/min, η_v at 0.95 and η_s at 0.65. Based upon R-134a performance, a family of

compressors is sized for displacement/revolution to give nominal capacities of 1.24, 2.44, 3.6, 5.0, 10.0, 11.0 and 20 kW.

Table 5-2 shows the Alpha Compressor Family, rated with different refrigerants, and Table 5-3 shows the Alpha Compressor Family, rated for low temperature refrigeration applications.

Compressor Name	Compressor Model	Compressor Capacity ⁶ (kW)				
		R-134a	R-22	R-12	R-404a	Ammonia
AlphaCompressor1240		1.24	1.93	1.20	1.95	2.17
AlphaCompressor2440		2.44	3.81	2.37	3.84	4.28
AlphaCompressor3600		3.60	5.63	3.50	5.67	6.32
AlphaCompressor5000		5.00	7.81	4.86	7.88	8.78
AlphaCompressor10000		10.0	15.6	9.71	15.7	17.5
AlphaCompressor11000		11.0	17.7	10.7	17.3	19.3
AlphaCompressor20000		20.0	31.2	19.4	31.5	35.1

Table 5-2: Alpha Compressors Rated For Air-Conditioning

Compressor Name	Compressor Model	Compressor Capacity ⁷ (kW)				
		R-134a	R-22	R-12	R-404a	Ammonia
AlphaCompressor1240		0.177	0.359	0.210	0.334	0.336
AlphaCompressor2440		0.349	0.707	0.414	0.658	0.662
AlphaCompressor3600		0.515	1.04	0.611	0.970	0.977
AlphaCompressor5000		0.716	1.50	0.849	1.35	1.36
AlphaCompressor10000		1.43	2.90	1.70	2.70	2.71
AlphaCompressor11000		1.57	3.18	1.86	2.96	2.98
AlphaCompressor20000		2.86	5.80	3.40	5.40	5.43

Table 5-3: Alpha Compressors Rated For Low-Temperature Refrigeration

6 Capacity based upon 40°F (4.4°C) / 115°F (46.1°C) evaporating / condensing temperature, 7°F superheat / 10°F (3.9-5.6°C) subcooling

7 Capacity based upon -40°F (-40.0°C) / 105°F (40.6°C) evaporating / condensing temperature, 7°F superheat / 10°F (3.9-5.6°C) subcooling

5.5.3 The Beta and Gamma Compressor

The Beta and Gamma compressors are essentially the same as the Alpha Compressors. The difference between models is the isentropic efficiency value. Beta compressor's have an isentropic efficiency of 0.85, while the Gamma Compressor has an isentropic efficiency of 0.55. The three similar compressors are provided to offer high, medium and low efficiency compressors of similar capacity.

5.5.4 The Delta Compressor

The Delta compressor is built upon the Alpha Compressors. To make the model more realistic, the isentropic efficiency of the compressor varies as a function of the compression ratio the compressor pumps against. The isentropic efficiency versus compression ratio is dependent upon the compressor's working fluid, and therefore currently the Delta compressor only supports simulation for R-134a applications.

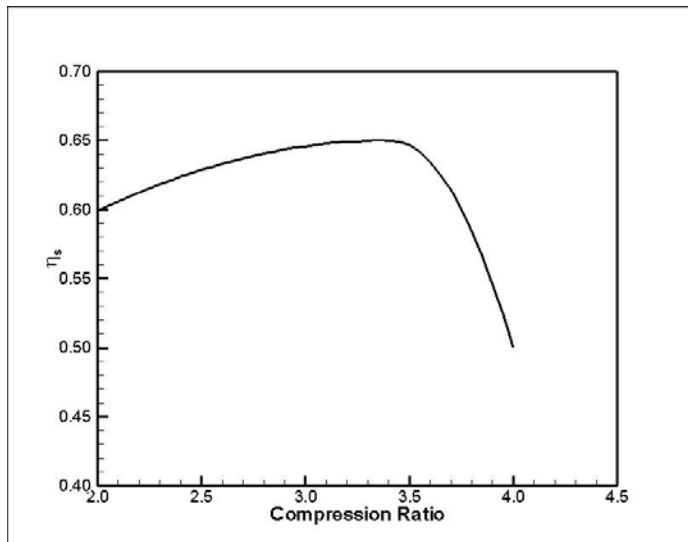


Figure 5-1: η_s versus Compression Ratio for Delta Compressor

Figure 5-1 shows the value of the Delta compressor's isentropic efficiency as a function of pressure ratio. The Delta compressor is designed to have an isentropic efficiency identical to that of the Alpha compressor at the air conditioning design point, $\eta_s = 0.65 @ r_p = 3.4$.

5.5.5 ARI 10 COEFFICIENT MODEL

The compressor model is based upon the 10-coefficient modeling method given in the Air Conditioning and Refrigeration Institute standard ARI 540-99 (ARI, 1999). For this model, a compressor's performance is described by polynomials of the form

$$X = C_0 + C_1S + C_2D + C_3S^2 + C_4SD + C_5D^2 + C_6S^3 + C_7DS^2 + C_8SD^2 + C_9D^3. \quad (5.2)$$

In this equation, X, represents a performance value, C_i are coefficients, specific to the performance value of interest, S is the *Suction Temperature* (in °F) and D is the *Discharge Temperature* (in °F). For this particular model, two sets of coefficients are used to represent the compressor mass flow rate and power requirement at a given operating point. Note, suction and discharge temperature are the dew temperature corresponding to the compressor's suction pressure and bubble temperature corresponding to the compressor's discharge pressure respectively.

Compressor manufacturer's supply the coefficients for their specific compressors by testing the compressor under the conditions given in the standard and fitting the coefficients to the test data.

5.5.6 The Generic Orifice

Stoecker (1958) gives a relation for the flow of refrigerant through an orifice as

$$\dot{m} = KD^2\sqrt{\Delta p \rho} \quad (5.3)$$

where K is a dimensionless, Δp has units of Pa, ρ has units of kg/m^3 and \dot{m} has units of kg/s . The generic orifice uses this formula to compute the mass flow rate through the orifice in terms of the fluid thermodynamic state and the specified pressure boundary conditions on the orifice.

5.5.7 Generic Heat Exchanger

A generic heat exchanger model is one which can offer a great deal of flexibility. One type of model is a single tube, with refrigerant flowing on the inside, and air flowing in a cross-flow configuration on the outside. If the model were to experience either condensation or evaporation, it divides into sub-segments, keeping heat transfer and momentum relations separate for the different heat transfer and momentum transport modes.

The single tube model is not realistic in practice. To keep a generic heat exchanger model as simple as possible the length of the heat exchanger is an equivalent length for a realistic heat exchanger with the given duty of that modeled heat exchanger.

The following model assumptions which apply to the condensing and evaporation sections, as well as single phase flow can be made for a computationally efficient heat exchanger model.

- Constant air flow rate and thermo-physical properties
- Heat transfer coefficients are functions of fin ratio and fin efficiency
- Refrigerant mass flow is a function of component boundary pressures
- All of the (refrigerant) pressure drop for the component occurs in the thermodynamic regime (vapor, two phase or liquid) of the entering fluid, and over the entire length of the heat exchanger

Using a classic relation of pressure drop to flow is (Bird, et al., 1960),

$$f = \frac{1}{4} \frac{D}{L} \left(\frac{\Delta P}{\frac{1}{2} \rho V^2} \right). \quad (5.4)$$

This relation is valid for laminar flow only, which leads to inaccuracies under conditions which result in turbulent flow. It does however qualitatively capture the behavior of the pressure – resistance characteristic.

An explicit form of mass flow can be determined, which is an advantage computationally.

$$\dot{m} = \frac{\pi}{4} D^{5/2} \sqrt{\frac{\Delta P \rho}{2fL}}. \quad (5.5)$$

Taking the heat exchanger inlet properties, and using the entire heat exchanger length as the parameter L, allows for a simple approximation of the heat exchanger mass flow rate as a function of pressure difference, inlet density (enthalpy) and heat exchanger geometry.

This radical simplification may differ greatly from the actual behavior one may expect in the laboratory, but it captures qualitatively the behavior looked for from the model, namely a decrease in flow with decreased pressure drops and/or increased inlet superheats. The main benefits of the simplification are the reduced calculation time and elimination of the need for iteration in the heat exchanger model.

It is stressed here that this generic heat exchanger model is a specific compressor model designed to execute quickly, to facilitate the validation of VapCyc. Although this model is fairly simple in nature, models of any level of sophistication can be used within VapCyc.

Heat transfer between air and refrigerant during the phase change is modeled with a simple relation,

$$\begin{aligned} Q_{ref} &= \dot{m}_{ref} C_p (T_{ref,Out} - T_{ref,In}) \\ Q_{ref} &= UA \left(T_{air,In} - \frac{(T_{ref,in} + T_{ref,out})}{2} \right) \end{aligned} \quad (5.6)$$

Combining the two expressions in (5.6), and rearranging terms, allows the refrigerant outlet temperature to be expressed as,

$$T_{ref,out} = \frac{2\dot{m}C_p T_{ref,in} + UA(2T_{Air,in} - T_{ref,in})}{2\dot{m}C_p + UA} \quad (5.7)$$

Heat transfer between air and refrigerant (either evaporating or condensing) is modeled through a simple relation,

$$\begin{aligned} Q_{Air} &= UA(T_{air,In} - T_{Evap/Cond}) \\ Q_{Ref} &= \dot{m}_{ref} (h_{ref,out} - h_{ref,in}) \end{aligned} \quad (5.8)$$

A caveat of this approach is a potential violation of the 2nd Law of Thermodynamics. This is guarded against by checking the outlet temperature of the refrigerant. If it is less than the saturation temperature, than the model is sub divided into sub-sections.

5.5.8 The Alpha Heat Exchangers

The Alpha heat exchangers are based upon the same model as the generic heat exchangers described in 5.5.7. In this case values for length and diameter of the heat exchanger tube are held constant, thus simulating a particular heat exchanger.

Using the VapCyc air-conditioning context, the Alpha Condenser and evaporator are sized by determining the appropriate length and diameter, which allows them to match a desired capacity at the nominal rating point.

5.5.9 The Eta and Epsilon Condensers

Although the Alpha Condenser family calculates quickly and qualitatively behaves well, it is not sensitive to air conditions. In reality, air conditions may affect the heat exchanger performance and therefore the system as a whole. The Eta and Epsilon condenser are constructed from distributed segments, and sensitive to air conditions.

ASHRAE (1988) recommends air face velocities around 6.5 ft/s (2 m/s) and air flow rates in the range of 600 to 1200 CFM /ton (80 to 160 l/s-kW), resulting in air temperature changes of 15 to 40°F (7 to 22°C).

The Eta Condenser family was designed to meet not only the refrigerant design specifications of the VapCyc air-conditioning context, but also the ASHRAE recommendations for the air side. As was revealed in chapter five, heat exchangers built from many segmented tubes can be very costly computationally. Therefore, a single segment which meets the above requirements was modeled.

Although the single segment may not represent a realistic model, it represents a scaled version of the desired condenser. It is therefore easy to imagine several of these segments in parallel. To get a desired capacity, the segment values need only be multiplied by a performance factor. Table 7-4 shows the dependent variables provided by the Eta Condenser at the VapCyc air-conditioning context.

Condenser Property	Nominal Value
Mass flow (g/s)	0.152
Heat Transfer (W)	29.04
Charge (g)	0.108
Inlet Air Temperature (°C)	35.0
Outlet Air Temperature (°C)	42.0

Table 5-4: Eta Condenser Nominal Operating Point for Air-Conditioning

Similar in nature to the Eta condenser family is the Epsilon condenser family. Rather than a single segment representing the heat exchanger, a single tube represents the heat exchanger. This tube can be split into an arbitrary number of segments, allowing the heat transfer and pressure drop to be evaluated with greater accuracy, with respect to the physical phenomena.

The Eta and Epsilon condensers also calculate air pressure drop. According to Shah in (Kreith,1999), the pressure drop through the core of an array of tubes covered with fins can be approximated as:

$$\Delta p = \frac{4fLG^2}{2g_c D_h} \left(\frac{1}{\rho} \right)_m \quad (5.9)$$

using the heat exchanger friction factor, f , the fin depth, L , the mass flux, defined as $G = \frac{\dot{m}}{A}$, the hydraulic diameter, D_h and the mean air density. The pressure drop is calculated from equation (5.9), assuming a constant density and a friction factor given by (Webb, 1994) as:

$$f = 0.508 Re_d^{-0.521} (X_t / d_0)^{1.318} \quad (5.10)$$

In this equation, X_t is the fin depth and d_0 is the tube diameter.

The fan power requirement is given as the thermodynamic power (sometimes referred to as *Brake Horse Power*) required to increase the given flow through the calculated pressure drop. The actual power requirement is the thermodynamic power divided by a fan efficiency and motor efficiency (White, 1986).

$$P = \frac{\rho g Q H}{\eta_{fan} \eta_{motor}} \quad (5.11)$$

In this case the air volume flow rate, Q is multiplied with air density, ρ , the gravitational acceleration constant and the head, defined as $H = \frac{\Delta p}{\rho}$. The Eta and

Epsilon condensers use $\eta_{fan} = \eta_{motor} = 1.0$;

5.5.10 Heat Exchanger Summary

A summary the heat exchangers presented in this chapter is detailed in, Table 5-5

Name	Features
Generic Heat Exchanger	<ul style="list-style-type: none"> • Constant air flow rate and thermo-physical properties • Heat transfer coefficients are functions of fin ratio and fin efficiency • Refrigerant mass flow is a function of component boundary pressures • All of the (refrigerant) pressure drop for the component occurs in the thermodynamic regime (vapor, two phase or liquid) of the entering fluid, and over the entire length of the heat exchanger
Alpha Condenser/Evaporator	<ul style="list-style-type: none"> • Based upon Alpha Heat Exchanger • Fixed Geometry
Eta Condenser	<ul style="list-style-type: none"> • Single distributed segment (see section 4.2) • Constant Geometry • Air pressure drop/power requirement
Epsilon Condenser	<ul style="list-style-type: none"> • A series of distributed segments • Constant Geometry • Air pressure drop/power requirement

Table 5-5: A Summary of VapCyc Heat Exchangers

5.6 VapCyc Utility

As an engineering tool, VapCyc provides a certain amount of utility, as outlined next. The tool is designed to offer a comprehensive set of features, which are currently lacking in existing software. The ability to add functionality is naturally facilitated via the generalized energy system modeling approach detailed in this thesis.

5.6.1 Charge Management

Recalling the modeling equations of chapter 2, as presented in Table 2-2; systems are categorized into two categories: open and closed. With regards to mass, an open system is one where mass is allowed to flow across the system boundaries, while a closed system contains a constant amount of mass at all times. Charge management is an accounting of the mass within a system, and thus utilizing the closed system mass balance, the charge can be specified, and accounted for at all system operating points.

5.6.2 Component Inter-Changeability

The generalized energy system modeling technique calls for a set of components, which operate in concert with one another via the solution of system level mass and energy balances. Thus an advanced tool, such as VapCyc, facilitates component inter-changeability by employing this technique. The components are distinct engineering models. The resolution of the modeling equations presented in Table 2-2 is accomplished independently from components.

Furthermore, by allowing the user to interface with the component models themselves, individual component boundary conditions can be modified independently of the solver resolving the system level equations of Table 2-2.

5.6.3 Multiple Refrigerants

Requiring the component models themselves to resolve their operating point given specified boundary conditions allows for a system level solver which is independent of choice of refrigerant. In VapCyc each component utilizes RefProp (NIST, 2001). VapCyc can poll components at run time to see which refrigerants from RefProp the

components are capable of utilizing. By taking all components into account, a list of refrigerants available to the user to utilize in the simulation is presented.

5.6.4 Multiple Unit Systems

Requiring component models to operate with a specified set of unit systems, allows the VapCyc interface to specify the units of the boundary conditions specified to components. This allows the user of the simulation to operate in a unit system of choice.

5.6.5 Physical Variables

A vapor compression refrigeration system simulation is used to estimate many physical variables over a range of operating conditions. The current list of physical variables calculated in the simulation includes:

- COP
- Capacity
- Weight
- Volume
- SEER – Seasonal Energy Efficiency Ratio

5.6.6 Economic Variables

A meaningful simulation also supplies economic variables of interest. A current list of variables includes:

- System cost
- System reliability
- System manufacturability

The system level variables are calculated using values obtained from the individual components, which are dependent upon individual models. The usefulness of this is realized when component values, such as component reliability are functions of

component performance. An example is a compressor which calculates a reliability value which is dependent upon the compressor's pressure ratio.

5.6.7 Seasonal Performance

ARI standard 210/240-94 "*Unitary Air Conditioning and Air-Source Heat Pump Equipment*" (ARI, 1996) provides a testing standard, which results in a seasonal figure of merit. This figure of merit is the Seasonal Equipment Energy Efficiency Ratio (SEER), a ratio of the equipment cooling or heating performance to the system's electrical requirements. The calculation procedure given in the standard is straightforward, and implemented in VapCyc.

5.6.8 Interactive Mode

The user of VapCyc is able to obtain meaningful results without a thorough understanding of the programming and/or mathematical modeling involved in the simulation. To this end, a "user-friendly" interface allows the user to modify the simulation interactively and view the results immediately.

Figure 5-2 is a screen shot of the VapCyc Cycle Calculation page, with the Alpha 3600 system loaded. This page represents the main interface for interactive use of VapCyc. Visible in the page are: the four icons, representing the four main VapCyc components, model numbers for the currently selected components, the selected refrigerant, the system charge, system performance variables (not all are available before the simulation is run), and the nominal capacity and charge of the four main components in the current context. For demonstration purposes, a figure of merit called "Final Energy

Balance” is given on this screen. This represents the total energy balance of the system after solution (in W).

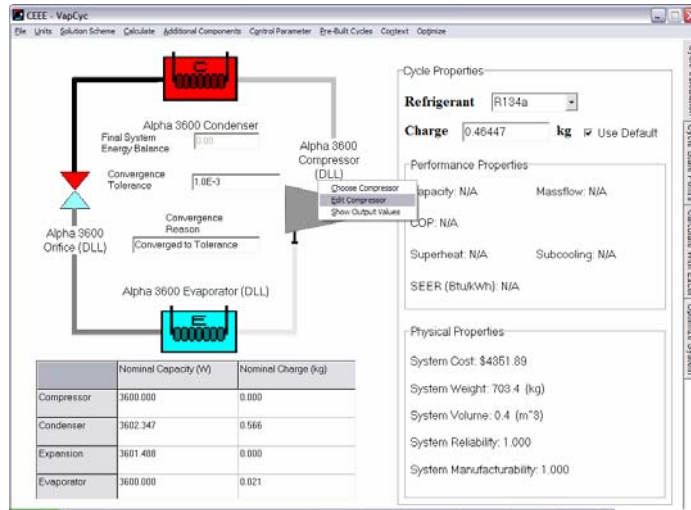


Figure 5-2: The Alpha 3600 System in Air-Conditioning Mode

The pictures representing models can be clicked with the mouse, allowing the user to (a) choose a component, (b) edit the current component or (c) view the component’s output variables (after run).

Figure 5-3 illustrates the changing of the compressor, in this particular case; the Alpha 3600 compressor is replaced with the Alpha 20000 compressor.

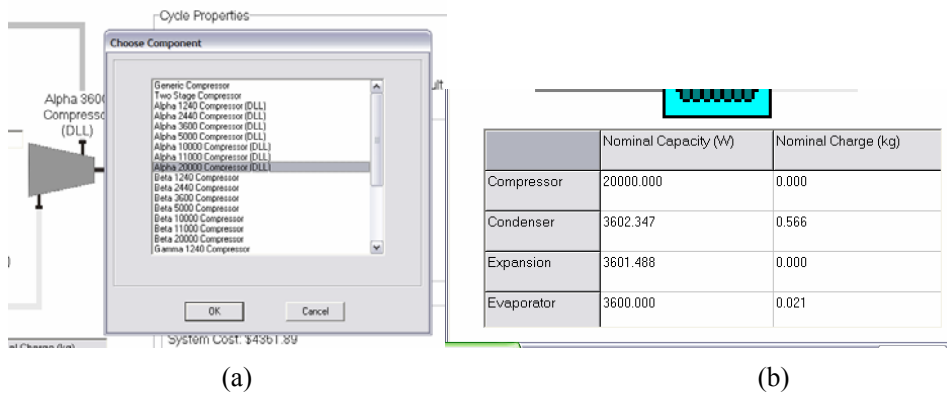


Figure 5-3: Changing of the Compressor

Note the change in the system variables that accompany the compressor change. Upon the change of any variable, the nominal system charge and/or nominal capacity may change. The component nominal capacities and charges can be seen in Figure 5-3(b). Here, it is seen that changing the compressor has not changed the nominal system charge, but the compressor's nominal capacity is now quite different than the other three components.

5.6.9 Automated Mode: Parametric Analysis

To be a useful tool for more than a single operating point, parametric analysis is offered through an automated mode. This automated mode, operating with Microsoft Excel, allows system configurations to be specified by text only. Multiple runs are then performed without monitoring the simulation.

5.7 VapCyc System Model

Figure 5-4 shows a simple four component vapor compression refrigeration system. The components are connected to one another via junctions. The components and their connectivity are specified to the Junction Solver to form the vapor compression refrigeration system.

5.7.1 Scaling the Independent Variables and Balance Equations

Recall, that for conditioning purposes, the Junction Solver has the parameters built in for scaling: P_{high} , P_{low} , h_{high} , h_{low} , \dot{m}^* , \dot{E}^* . Recall the junction solver scales independent variables in the form of

$$P_i^* = \frac{P_i - P_{low}}{P_{high} - P_{low}} \quad (5.12)$$

$$h_i^* = \frac{h_i - h_{low}}{h_{high} - h_{low}} \quad (5.13)$$

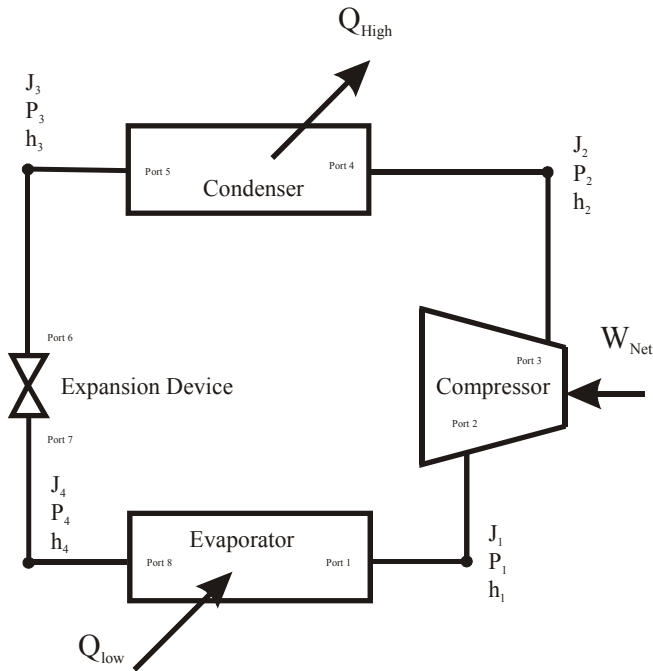


Figure 5-4: Simple Vapor-Compression Refrigeration System

Similarly, the Junction Solver balance equations are scaled as

$$r_i = \frac{\sum_{p=1}^{N_{ports}} \dot{m}_p}{\dot{m}^*} \quad (5.14)$$

$$r_j = \frac{\sum_{p=1}^{N_{ports}} \dot{m}_p (h_p + \frac{v_p^2}{2} + g z_p)}{\dot{E}^*} \quad (5.15)$$

The choice of scaling parameters should represent some physical meaning to the system. This often offers meaningful values to convergence criteria and independent variable bounds. The system depicted in Figure 5-4 however, does not give any

indication of characteristic pressures (P_{high} , P_{low}), enthalpies (h_{high} , h_{low}), mass flow rate, or energy flow rate.

5.7.1.1 The System Context

Scaling the variables and equations of a simulation when the result of the simulation is unknown is achieved using the “real life” concept of equipment rating. Figure 5-5 shows an example of a compressor manufacturer’s model number explanation (Tecumseh, 2004). Included in the compressor model number is a code for a nominal rating point, and nominal capacity at that rating point.

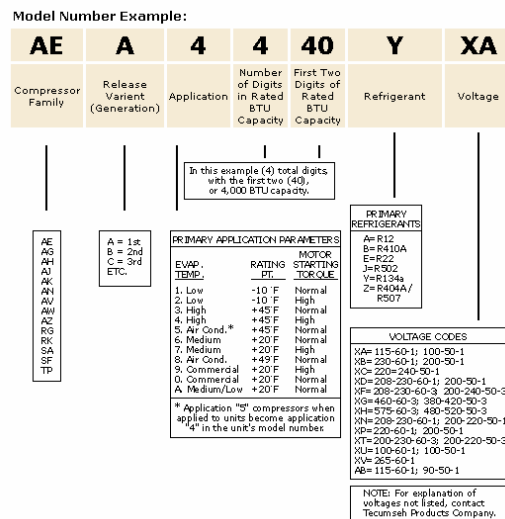


Figure 5-5: A Compressor Model Number (Tecumseh, 2004).

The compressor model number shown in Figure 5-5 represents a compressor for “high temperature application” (45°F / 7.2°C) and provides a cooling capacity of 4000 Btu/h (1.17 kW) at its rating point. The rating point, although not published in Figure 5-5 is defined in ARI standard 540-99 (ARI, 1999), and consists of suction and discharge pressures, as well as inlet gas temperature. The compressor’s flow rate can be

correlated to an evaporator capacity by multiplying it by a nominal heat transfer per unit mass. This nominal heat transfer is given in terms of an evaporating pressure, condensing temperature condenser exit liquid temperature (subcooling) and evaporator superheating, or compressor return gas temperature. The rating point and the compressor's cooling capacity at that point provide enough information to derive the compressor's suction and discharge pressures, as well as the suction temperature and mass flow rate.

Figure 5-6 shows rating points which vary with compressor application, as published in the 540-99 standard.

Suction Dew Point Temperature		Compressor Type	Discharge Dew Point Temperature		Return Gas Temperature	
°F	°C		°F	°C	°F	°C
45	7.2	All	130	54.4	65	18.3
20	-6.7	All ^a	120	48.9	40/65 ^a	4.4/18.3 ^a
-10	-23.3	Hermetic	120	48.9	40	4.4
-25	-31.7	All ^a	105	40.6	40/65 ^a	4.4/18.3 ^a
-40	-40	All ^a	105	40.6	40/65 ^a	4.4/18.3 ^a

If air flow across the compressor is used to determine ratings, it shall be specified by the compressor manufacturer.
^a For hermetic type compressors 40°F [4.4°C] return gas temperature shall be used.
 For external drive and accessible hermetic type compressors 65°F [18.3°C] return gas temperature shall be used.

Figure 5-6: ARI 540-99 Rating Points for Compressors

A similar set of rating points is given in the standard for compressors which are to be used as both cooling and heating compressors in a heat pump application. These rating points are shown in Figure 5-7.

Even though compressors have different operating characteristics, having similar ratings at a common operating point allows for an educated guess that the compressors should behave similarly, at least, in the region of this defined operating point.

Consider the two compressors represented by published manufacturer's data shown in Figure 5-8 and Figure 5-9, the Copeland CS10K6E-PFV and ARJ11K1E-CAV respectively (Copeland, 2004).

ANSI/ARI STANDARD 540-1999

Table 2. Standard Rating Conditions for Compressor(s) and Compressor Unit(s) Used In Air Conditioners and Heat Pumps (Based on 95°F [35°C] Ambient Temperature)						
Application Test Points	Suction Dew Point Temperature		Discharge Dew Point Temperature		Return Gas Temperature	
	°F	°C	°F	°C	°F	°C
Air Source (Cooling)	45	7.2	130	54.4	65	18.3
Air Source (Heating - High Temp.)	30	-1.1	110	43.3	50	4.4
Air Source (Heating - Low Temp.)	5	-15	95	35	25	-3.9
Water Source Cooling & Heating	45	7.2	120	48.9	65	18.3

If air flow across the compressor is used to determine ratings, it shall be specified by the compressor manufacturer.

Figure 5-7: ARI 540-99 Heating and Cooling Compressor Operating Points

At the ASHRAE high temperature rating point (marked in red) in Figure 5-8, these compressors are very similar, giving a cooling capacity of 11,600 and 11,300 Btu/hr (3.4 and 3.3 kW) respectively. The capacity difference at that particular rating point is only 2.6%. However, comparing the two displacement rates, it is seen that these are not as mechanically similar as the capacity at the ASHRAE high temperature rating point would suggest; with displacement rates differing by 7.5%. The capacity of the machines, when rated at the lower temperature rating point, corresponding to the second ASHRAE rating point in Figure 5-9 (marked in blue) differ by approximately 10%.

In this thesis, a distinct thermodynamic operating point, which is applied to an individual piece of equipment, is deemed to be a *context*. Using this nomenclature, each of the five entries in Figure 5-6 is a unique context at which a given compressor will have a distinct operating point and performance.

A compressor can be described with reasonable accuracy simply by denoting a nominal operating condition. Choosing either the high temperature (45°F / 7.2°C) or medium temperature (20°F / -6.7°C) as a context, a nominal operating point is established. Provided the solution of the system model does not force the compressor model to vary from this range significantly, its mass flow rate is fairly close to the nominal value. Thus, the nominal flow rate (or nominal capacity), as defined by the nominal rating point is a good choice for scaling the mass flow rate and mass balance equations.

CS10K6E-PFV

HFC, R-134a, 60 Hz, 1 - Phase, 208/230 V
High Temperature



Production Status: Available for sale to all U.S. customers. Please check with your local Copeland representative for international availability.

Performance			Mechanical	
Evap(°F)/Cond(°F)	45/130	20/120	Number of Cylinders: 2	CIR (in ³ /Rev): 2.256
RG(°F)/Liq(°F)	65/130	65/120	Bore Size (in): 1.516	CFH (ft ³ /hr): 274.1963
Capacity (Btu/hr):	11600	5840	Stroke(in): 0.625	
Power (Watts):	1300	937	Overall Length (in): 10.28	Mounting Length (in): 7.5
Current (Amps):	7.5	6.3	Overall Width (in): 9.09	Mounting Width (in): 7.5
EER (Btu/Wh):	8.9	6.2	Overall Height (in): 13.31	Mounting Height (in): 13.69
Mass Flow (lbs/hr):	203	93	Suction Size (in): 3/4 Stub	
Sound Power (dBA):		0 Ave 0 Max	Discharge Size (in): 3/8 Stub	
Vibration (mils(peak-peak)):		0.0 Ave 0.0 Max	Initial Oil Charge (oz): 45	
			Oil Recharge (oz): 43	
			Net Weight (lbs): 70.0	

Figure 5-8: Performance Summary of a Copeland CS10K6E-PFV Compressor (Copeland Web Site)

Similar to the compressor, the remaining components have standards for their rating. Evaporators are rated in such standards as ARI 420-2000, *Unit Coolers for Refrigeration* and ARI 440-1998, *Room Fan-Coils*. Condensers are specified in standards, such as ARI 450-99, *Water-Cooled Refrigerant Condensers, Remote Type*, and ARI 460-2000, *Remote Mechanical-Draft Air Cooled Refrigerant Condensers*. Expansion devices are rated according to ARI 750-2001: *Thermostatic Refrigerant Expansion Valves*. Each

standard defines a context in which the given piece of equipment can be, at least partially, characterized.

ARJ11K1E-CAV

HFC, R-134a, 60 Hz, 1 - Phase, 208/230 V
High Temperature



Production Status: This compressor has been discontinued for sale to U.S. OEM customers. A field replacement compressor is currently available through a U.S. Copeland Primesource Wholesaler. Please check with your local Copeland representative for international availability.

Performance			Mechanical			
Evap(°F)/Cond(°F)	45/130	20/120	Number of Cylinders:	1	CIR (in ³ /Rev):	2.099
RG(°F)/Liq(°F)	65/130	65/120	Bore Size (in):	1.688	CFH (ft ³ /hr):	255.0943
Capacity (Btu/hr):	11300	6400	Stroke(in):	0.938		
Power (Watts):	1560	1100	Overall Length (in):	9.625	Mounting Length (in):	8.0
Current (Amps):	7.3	5.9	Overall Width (in):	6.5	Mounting Width (in):	4.8
EER (Btu/Wh):	7.2	5.8	Overall Height (in):	10.75	Mounting Height (in):	11.375
Mass Flow (lbs/hr):	191	101	Suction Size (in):	1	Rotarlock	
Sound Power (dBA):		Ave Max	Discharge Size (in):	5/16	Stub	
Vibration (mils(peak-peak)):		Ave Max	Initial Oil Charge (oz):	30		
			Oil Recharge (oz):	28		
			Net Weight (lbs):	63.0		

Figure 5-9: Performance Summary of a Copeland ARJ11K1E-PFV Compressor (Copeland Web Site)

Although these standards were chosen because they are all refrigeration/air-conditioning components, the standards themselves are for individual components. There is no insinuation that these devices will work in concert with others. In practice, when assembled into a refrigeration system, the eventual operating point will likely not result in the specified cooling capacity of any of the individual components. However, if the individual pieces of equipment have similar nominal capacities, at the same nominal operating point, the system will operate near this nominal operating point.

Individual component models do not necessarily relate to one another in any intuitive fashion. The challenge to the simulation program is to establish if one piece of equipment (model) is compatible with another, and if so, how might it go about solving the system model.

VapCyc utilizes a *system context* to help overcome the challenges. From a simulation stand point, it is not unreasonable for VapCyc to supply components a context, or operating point, and request its flow rate and/or heat transfer in that context. This offers a first approximation of component compatibility for the system as a whole.

5.7.1.2 The VapCyc Air-Conditioning Context

Fixing at least four components in the cycle, the compressor, condenser, expansion device and evaporator, VapCyc deviates from a general energy system to a specific one. VapCyc allows for the addition of components and the ability to change model and independent variables of components, which makes it general, but ultimately, VapCyc is some sort of vapor compression refrigeration cycle.

Taking advantage of this allows a context to be specified. This context is used to scale the simulation independent variables and balance equations. One particular context employed by VapCyc is a standard air-conditioning system. The compressor pressures are set by determining the nominal evaporating and condensing temperatures. A typical air-conditioning application would be approximately 40°F (4.4°C) evaporating and 115°F (46.1°C) condensing. Nominal pressure drop through the condenser and evaporator are assumed to be 3%. Superheating (evaporator) and sub cooling (condenser) are typically in the 7-10°F (3.9-5.6°C) ranges.

Figure 5-10 and Figure 5-11 graphically depict the system context of this simple vapor compression cycle. Figure 5-10 shows the set of nominal values that make up the standard air-conditioning context for VapCyc, while Figure 5-11 is a Pressure-enthalpy diagram of the nominal cycle. Components rated at this context provide mass flow, outlet energy state and refrigerant charge.

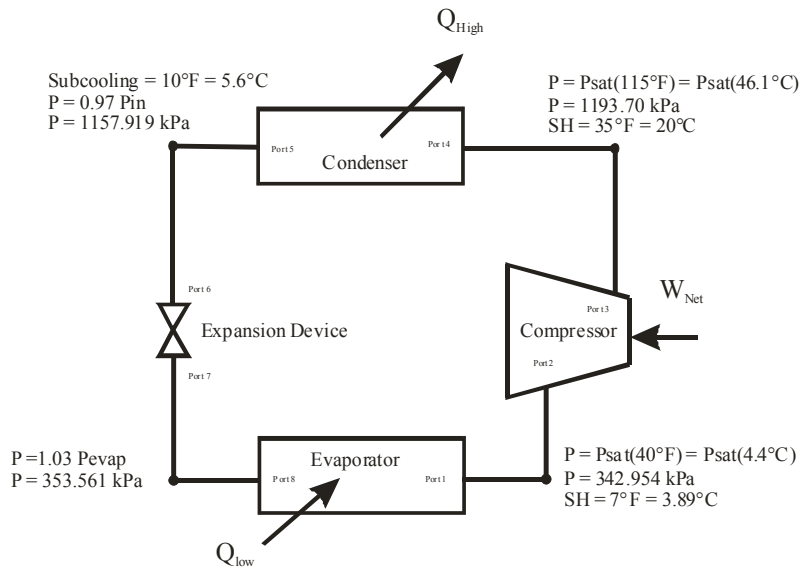


Figure 5-10: A Nominal Air-Conditioning Operating Point for VapCyc

Consider a context given to four components; were the condenser to have a rated mass flow value 20 times that of the other components, this is a sign that the system may be poorly matched. Poorly matched systems in practice may function to some degree, but in simulation often lead to a failure to find a solution (if it exists). The performance value of individual components at the given context allows the system to be analyzed for possible failure of the solution routine.

5.7.1.3 The Multiple VapCyc Contexts

VapCyc has five built in contexts: Residential Air-Conditioning (RAC), Residential Heat Pump (RHP), High Temperature Refrigeration (HT), Medium Temperature (MT) Refrigeration and Low Temperature Refrigeration (LT). Currently, all contexts have fixed values for condenser subcooling, 10°F/5.56°C; evaporator superheat, 7°F/3.88°C; compressor discharge superheat, 37.5°F/20.8°C; and heat exchanger pressure drop of 3%

of the inlet value. The evaporating and condensing pressures are given in Table 5-6 in the form of evaporating and condensing temperatures.

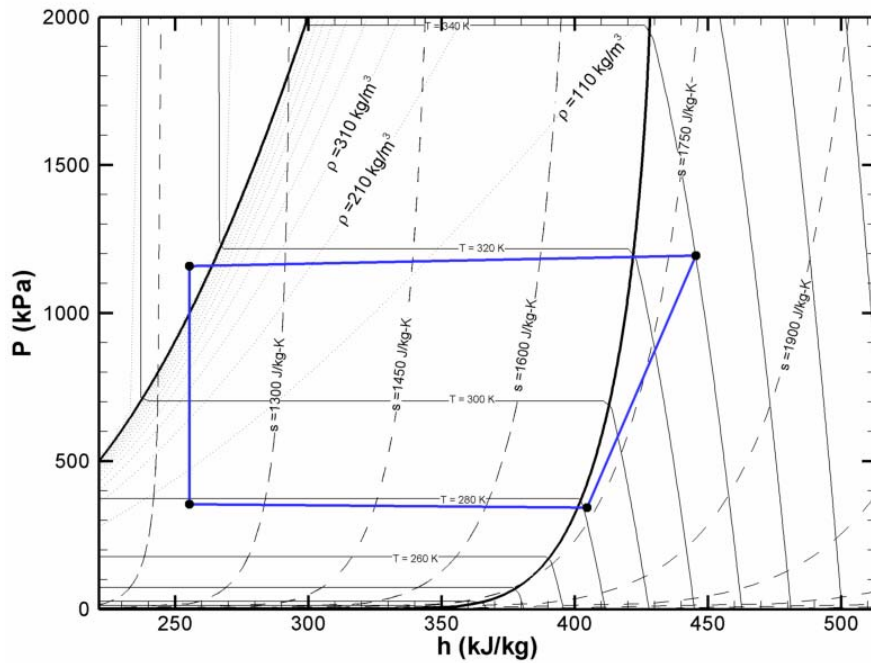


Figure 5-11: A Nominal Air-Conditioning Operating Point for VapCyc

Context	Evaporating Temperature (°F/°C)	Condensing Temperature (°F/°C)
Residential Air-Conditioning	45.0 / 7.2	130.0 / 54.4
Residential Heat Pump	30.0 / -1.1	110.0 / 43.3
High Temperature Refrigeration	20.0 / -6.7	120.0 / 48.9
Medium Temperature Refrigeration	-25.0 / -31.7	120.0 / 48.9
Low Temperature Refrigeration	-40.0 / -40.0	105.0 / 40.6

Table 5-6: VapCyc Evaporating and Condensing Temperatures

5.7.1.4 Scaling With the Context

The simulation equations and variables are scaled to the context. The nominal evaporating and condensing temperatures ($\bar{P}_{Evap} = P_{sat}(T_{Evap})$, $\bar{P}_{Cond} = P_{sat}(T_{Cond})$) provide good choices to replace P_{High} and P_{Low} in equation (5.12), as they represent the highest and lowest pressures in the air-conditioning context. Likewise the compressor discharge and evaporator inlet enthalpy values are the highest and lowest of the air-conditioning context. The air-conditioning context is for steady state, and thus each component has the same flow rate. Therefore the compressor nominal flow rate is chosen as the characteristic flow rate to scale mass flow residuals. The evaporator's nominal capacity is a good choice for a characteristic heat transfer value to scale energy balances with. The condenser's nominal heat rejection could be used just as easily. Also, at the given context each component will have a nominal charge. The nominal system charge is used to scale the system mass balance. Table 5-7 gives a full set of the scaled independent variables and equations used in VapCyc.

5.7.2 Control of VapCyc

The power of a simulation tool over many operating conditions for many components is useful, but the need to *specify* the system *completely* is problematic when design is the objective. VapCyc implements a control, which allows a user to specify an objective variable, and a variable to control it. For example, if one wants to size an expansion orifice to give a desired level of superheat at the evaporator outlet, VapCyc allows the user to specify the orifice diameter as an independent variable and the evaporator superheat as a dependent variable.

Variable / Equation	Scaled Variable / Equation
Junction Pressures, P*	$P_j^* = \frac{P_j - \bar{P}_{Evap}}{\bar{P}_{Cond} - \bar{P}_{Evap}}$
Junction Enthalpy	$h_j^* = \frac{h_j - \bar{h}_{Evap,in}}{\bar{h}_{discharge} - \bar{h}_{Evap,in}}$
Junction Mass Balance	$\frac{\sum_{p=1}^{Nports} \dot{m}_p}{\dot{m}_{Compl}} = 0$
Junction Energy Balance	$\frac{\sum_{pi=1}^{Nports,Inlet} \dot{m}_{pi} h_p - h_{po}}{\sum_{po=1}^{Nports,Outlet} \dot{m}_{po}} = 0$ Q_{Evap}
System Mass Balance	$\frac{\sum_{i=1}^{NComponents} Charge_i - System Charge}{Charge} = 0$

Table 5-7: Scaled Independent Variables and Balance Equations

5.8 VapCyc Examples of Capabilities and Verification

With the capability to change component models, as well as expand upon the basic system configuration, VapCyc represents a generalized energy system modeling approach to the standard vapor compression refrigeration cycle. VapCyc offers more utility than existing v-c simulations due to its set of features, namely, charge management, component inter-changeability, multiple refrigerants and seasonal performance.

Several examples are presented to validate the techniques described in this chapter which are used within VapCyc. Examples include: the affect scaling the independent variables and balance equations, the usefulness of the context as seen in the interactive mode, simulation and verification against actual experimental data, and seasonal performance of a system. The flexibility of the tool is demonstrated by modifications of the system configuration, by utilizing several evaporators in parallel and two compressors

in series. Lastly, VapCyc is used to perform a simple system optimization versus system charge and condenser air flow velocity.

5.8.1 Verification of the Model

The model is “solved” when the Junction Solver resolves all junction level mass and energy balances, in addition to the system mass balance. To confirm that the solution is indeed a physically meaningful one, P-h diagrams are generated and mass and energy balances are calculated. Three cases are examined for verification

1. The standard Alpha 3600 Air Conditioning System
2. The standard Alpha 3600 Air Conditioning System with the condenser inlet air set to 40°C
3. The standard Alpha 3600 Air Conditioning System, with a hot gas bypass valve installed between the compressor discharge and the evaporator inlet

In all cases, the convergence tolerance is set to 10^{-4} x compressor nominal capacity for energy ($3600 \text{ W} \times 10^{-4} = 0.36 \text{ W}$) and 10^{-4} x compressor nominal flow rate ($0.024 \text{ kg/s} \times 10^{-4} = 2.4 \times 10^{-6} \text{ kg/s}$).

	Junction Balance (Mass kg/s x 10 ⁻⁶ , Energy W)							
	1		2		3		4	
Test #	Mass	Energy	Mass	Energy	Mass	Energy	Mass	Energy
I	-0.565	-0.15	0.00934	0.01	0.00644	0.11	0.407	0.05
II	-1.67	0.11	-0.059	-0.03	0.180	-0.69	1.55	0.14
III	-1.50	0.29	0.150	-0.0021	0.059	0.0013	-1.74	-0.18

Table 5-8: Mass and Energy Balances for Test Configurations

Table 5-8 displays the final junction mass and energy balances for the three tests, numerically verifying the solution.

Figure 5-12 illustrates the results of the three cases used to validate VapCyc in graphical form using a P-h diagram. The shape of the P-h diagrams validates the model, being the expected shape for the four junction systems. The behavior of the system is as expected, and is seen increase in system condensing pressure for the increase in condenser air temperature, and the increase in suction pressure for the hot gas bypass configuration. The shape of the P-h diagrams, coupled with the mass and energy balances shown in Table 5-8 validate the simulation approach.

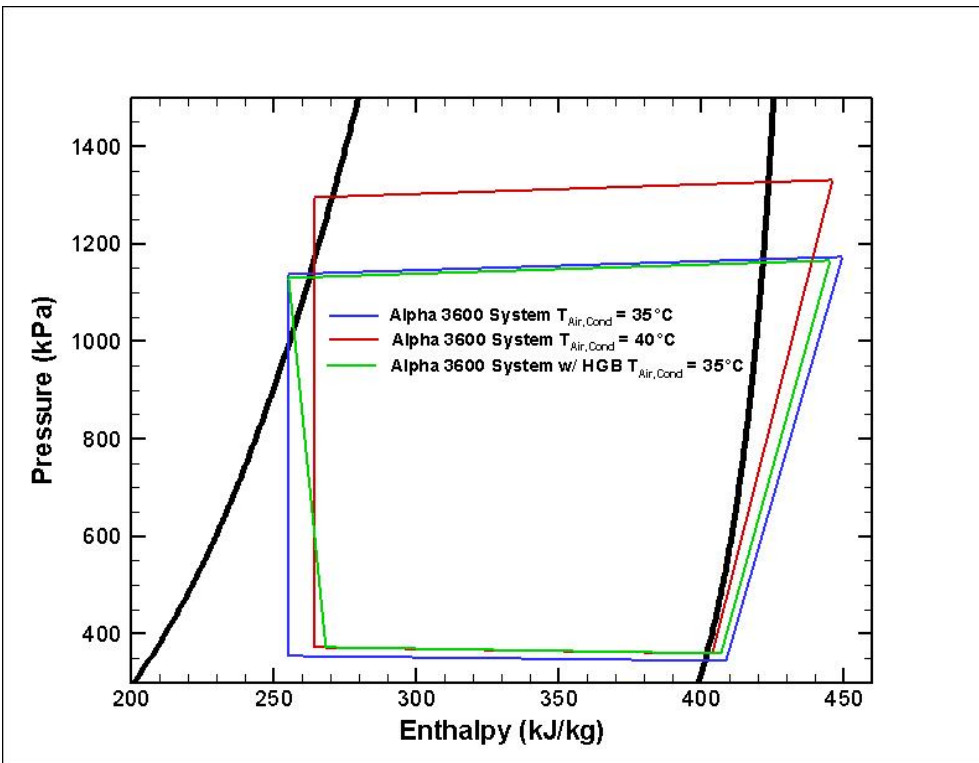


Figure 5-12: P-h Diagrams for Various Alpha 3600 Configurations

5.8.2 Verification of the Scaling

To verify the scaling is appropriate and validate the use of the context, a simple numerical experiment is performed. Two systems are simulated using the nominal charge, as provided by the component for the Air Conditioning context. The first system is the standard Alpha 3600 system (each of the four components is of the nominal 3.6 kW variety), while the second system is the Alpha 3600 system, but with an Alpha 10000 condenser. This second system is referred to here as the “hybrid system”. The simulation is performed using a NEWTON type non-linear equation. The significance of the NEWTON type solver is that a Jacobian matrix is calculated and a correction vector is calculated using this matrix. Large condition numbers for this matrix can result in erroneous solutions to this correction vector. The simulations are carried out with (a) with no scaling at all and (b) utilizing the Air-Conditioning context for scaling the independent variables and residual equations. In both cases the initial guess is provided to the system from the Air-Conditioning context.

Figure 5-13 gives a P-h diagram with three cycles overlaid. The first represents the VapCyc Air-Conditioning context, which serves as the initial guess for the two simulations, the second, is the solution of the Alpha 3600 system, with its nominal charge, and the third is the Alpha 3600 system with an Alpha 10000 condenser in place of the original. It is noted here, that as expected the Alpha 3600 system solution lies very near the VapCyc Air-Conditioning context, and except for the condensing temperature (pressure), the hybrid system does as well.

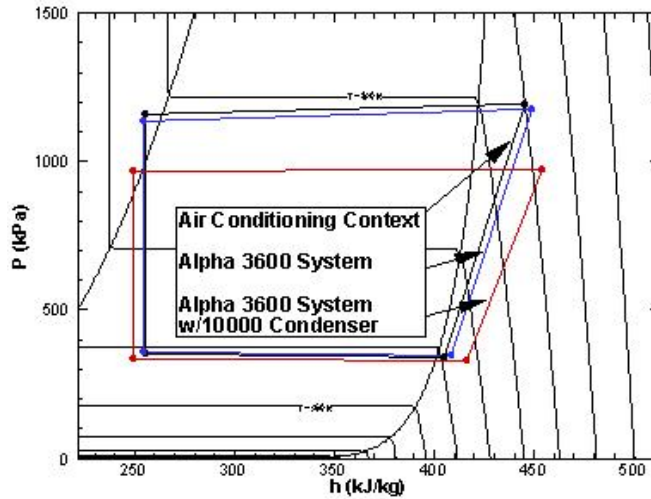


Figure 5-13: P-h Diagrams of Numerical Experiment Systems

Figure 5-14 shows a result of the Alpha 3600 system simulation. Two main points are evident, namely (1) The condition number of the Jacobian Matrix used for the NEWTON type solver at each iteration is of order 10^6 ($O(10^6)$) for the non-scaled case and of $O(10)$ for the scaled case, and (2) the non-scaled case requires 6 iterations, while the scaled case requires only 1.

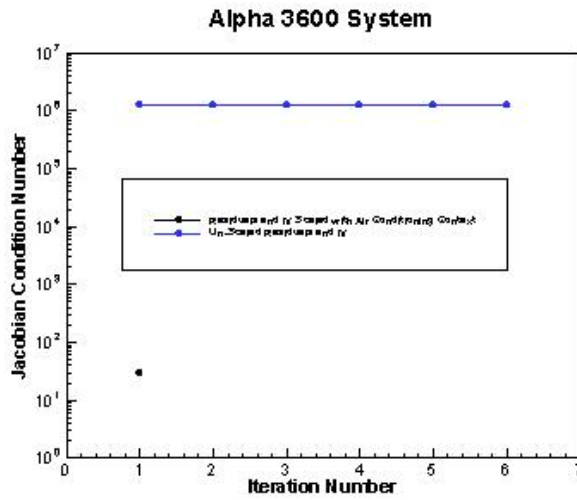


Figure 5-14: Jacobian Condition Number vs. Iteration Number

Figure 5-15 illustrates a result of the hybrid system simulation, with the larger condenser.

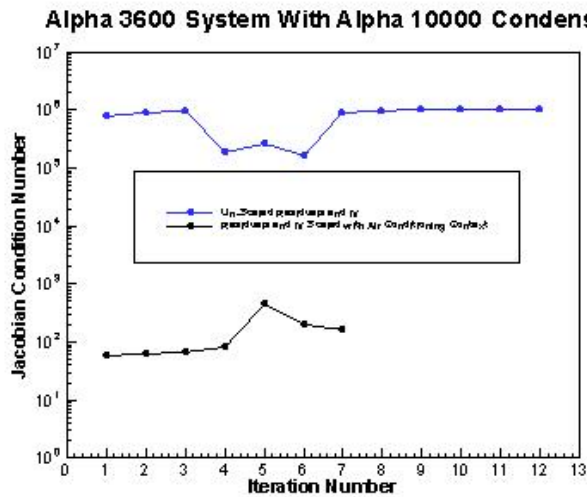


Figure 5-15: Jacobian Condition Number vs. Iteration Number

Points evident from this figure are: (1) The condition number of the Jacobian Matrix used for the NEWTON type solver at each iteration is again of order 10^6 ($O(10^6)$) for the non-scaled case and of $O(10)$ for the scaled case, and (2) the non-scaled case requires 12 iterations, while the scaled case requires only 7. Clearly, the numerical solver must work harder to solve this system of equations provided by this different set of components, and the use of the Context makes the computations easier.

The power of the Context as both a vehicle for an initial guess and a scaling tool is demonstrated by the ability to reduce the computational power to solve the problem and its ability to find a solution where a non-scaled problem may not be able to.

5.8.3 Examining the VapCyc Context

Figure 5-2 illustrates the main VapCyc interface. The screen shot shows the VapCyc calculation page, in the VapCyc RAC context. Recalling the ‘Alpha’ series components

were designed for air-conditioning application, using the Alpha3600 series for each component results in a nominal 3.6 kW capacity at the design point, which corresponds to the VapCyc RAC context.

Note, the option to use the default nominal charge is selected, so the sum of the nominal charge for each component (shown in the table in the bottom left corner) is equal to the specified system charge (in the upper right hand corner).

Table 5-9 shows the changes applied to the system when low temperature context (LT) is utilized. In this particular case, where both the evaporator and condenser have air as the secondary fluid, the only changes are the two secondary air temperatures.

	Air-Conditioning Context	<i>Low Temperature Refrigeration Context</i>
Condenser Air Temperature (°C)	35.00	35.00
Evaporator Air Temperature (°C)	26.67	-23.00

Table 5-9: Air Temperatures of A/C and Refrigeration Context

Figure 5-16, taken from the main VapCyc screen, details the Alpha3600 components, subject to the LT context. Each of the components behaves differently than when subject to the residential air conditioning (RAC) context.

	Nominal Capacity (W)	Nominal Charge (kg)
Compressor	515.817	0.000
Condenser	2931.433	0.566
Expansion	3430.277	0.000
Evaporator	382.628	0.002

**Figure 5-16: Performance of Alpha3600 Components:
Low Temperature Refrigeration Context**

The compressor, operating at a much lower suction pressure, has a drastically reduced capacity. The condenser, at nearly the same nominal operating point changes little in nominal capacity, and nominal charge contained within the condenser is virtually

indistinguishable. The changes in evaporating and condensing temperatures results in a slightly increased expansion device capacity. The evaporator experiences large differences in both nominal capacity and nominal charge, due to the large density differences between the two operating points. Comparing the nominal capacities of the components, the disparity between them is evident, and a solution to the model is unlikely; as expected, VapCyc cannot find a valid solution. Figure 5-17 shows a hybrid cycle of Alpha components, in the LT context. The Alpha 3600 orifice is replaced with the Alpha 500 orifice and the Alpha 3600 condenser is replaced with the Alpha 1240 condenser (smallest).

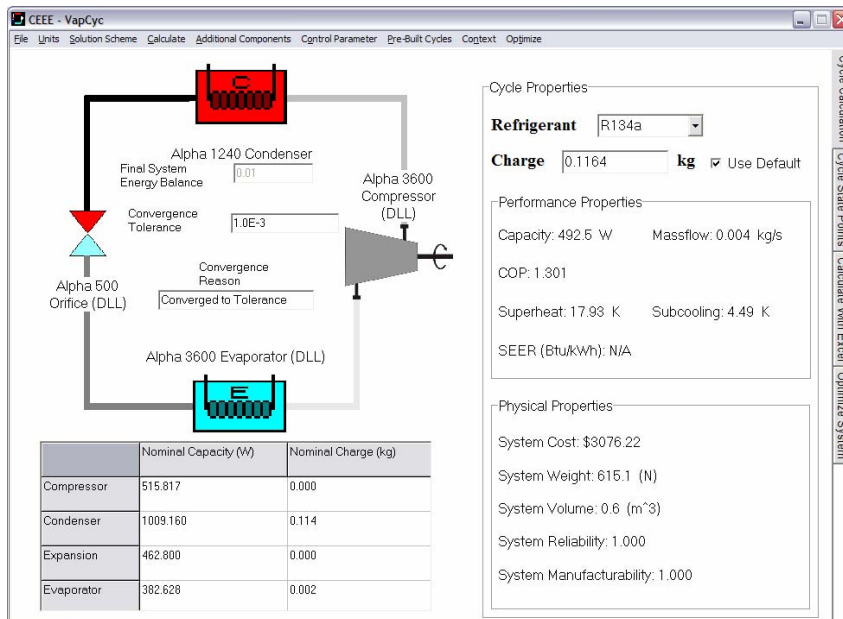


Figure 5-17: Low Temperature Solution to an Alpha Hybrid System

The VapCyc context has allowed the system to re-specify its operating condition from air-conditioning to low temperature refrigeration, as well as provide an initial guess value to the solver. Utilizing the nominal capacity allows for the identification of components which may offer difficulty in the cycle operating in an acceptable range, due to mis-

matched sizes. In this case the components are replaced manually to match the nominal capacities at the LT context better. Using the nominal capacity (or deviation in nominal capacity from other components) as a figure of merit, the process may be automated, allowing VapCyc to exchange equipment automatically. The ability of the context to offer insight to component compatibility as well as seeding of an initial guess to a solver gives VapCyc great power.

5.8.4 Comparison with Test Data

The comparison of the test data (Hwang, et al. 2004), and VapCyc simulations using TEST component models (see Appendix 1) is presented here for verification purposes. A main focus of the test project is the performance of the entire system, for a given refrigerant, as a function of the system charge, at constant evaporator superheat (TXV controlled).

The only degree of freedom in the test system experiment is the system charge. Two issues make comparison of refrigerant charge inventory between simulation and experiments difficult, namely (1) simulation models (Rice, 1987) can vary greatly in estimation of charge depending upon the correlations used to calculate them, and (2) as no information regarding piping dimensions is included in the report, the physical system piping connecting the various components are not accounted for in the simulation. The liquid line between condenser and expansion device in particular can hold a considerable amount of refrigerant.

Consequently, the VapCyc simulations are carried out at constant superheat, and refrigerant charge is varied to match the value of sub cooling at the condenser exit from

the given experimental test point. Results are given graphically in Figure 5-18 - Figure 5-21.

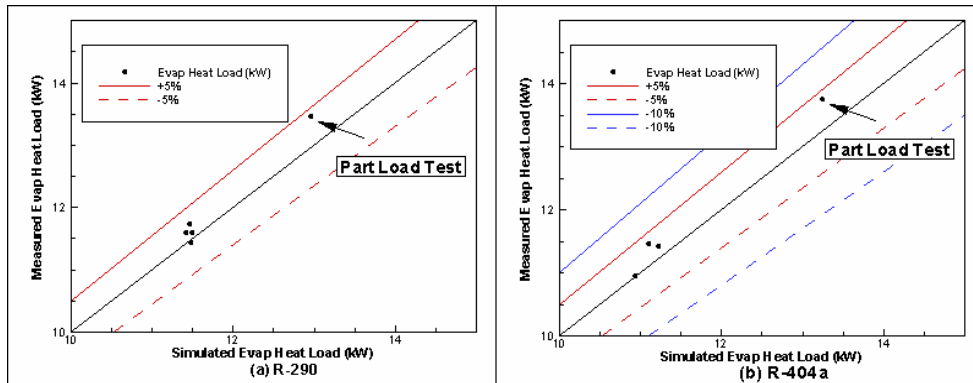


Figure 5-18: Simulated vs. Measured Evaporator Heat Load

Figure 5-18 demonstrates that the VapCyc simulation predicts the evaporator heat load within 5%, even for the part load test. The point corresponding to the “test 1” (Appendix I) data point agrees within 5%. This is not surprising, as the components were designed based upon this point. All other points slightly over predict the system’s cooling performance.

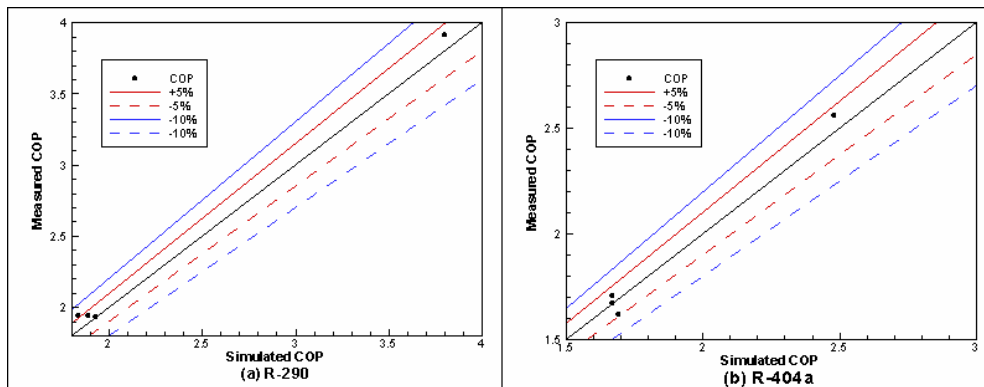


Figure 5-19: Simulated vs. Measured COP

Figure 5-19 illustrates results similar to that of the system’s cooling performance, namely the system COP is predicted within or near 5%. This is expected, as the cooling

load results are predicted well, and the compressor's off-design performance (see appendix 1) is very similar to the measured compressor off-design performance.

The good agreement of measured versus simulated compressor performance is also evident in Figure 5-20, where the system mass flow rate agrees with measured data to within 5%.

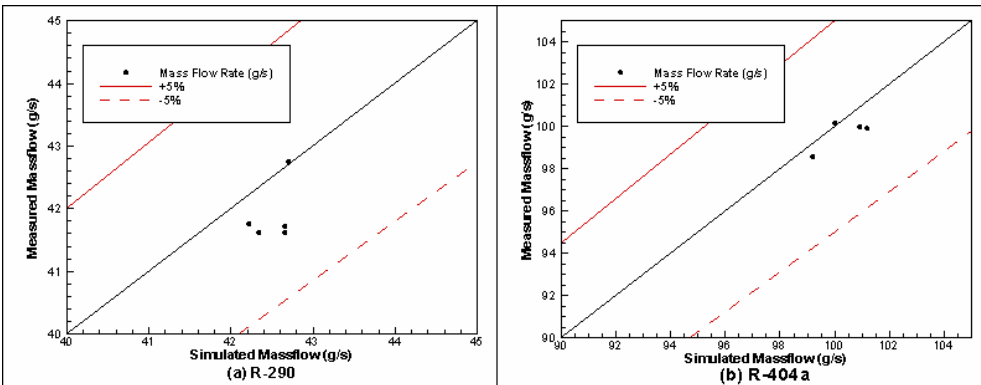


Figure 5-20: Simulated vs. Measured Mass Flow Rate

Figure 5-21 illustrates how difficult it is to simulate refrigerant charge inventory. To compare refrigerant charge in the system, a crude “charge correction” was made to the measured data. The charge correction is taken as the difference between the measured system charge at “test 1” from the test data (appendix 1) and the virtual charge from each component at “test 1”. This charge correction is assumed to be constant for each test point.

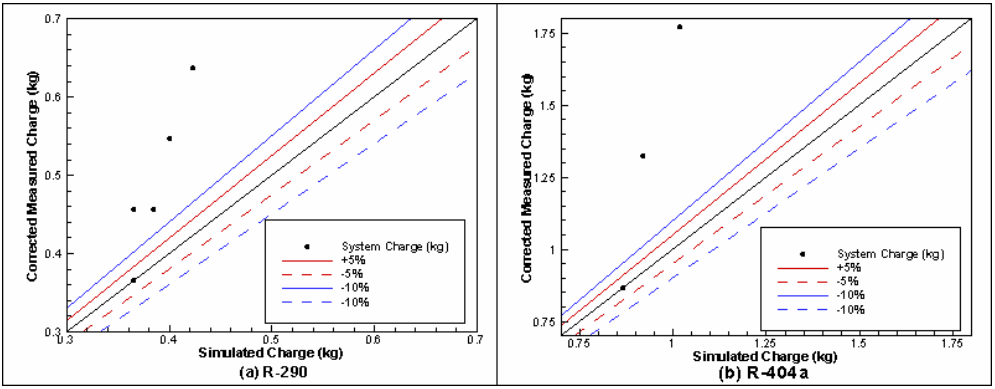


Figure 5-21: Simulated vs. Measured Refrigerant Charge

As expected, the “test 1” data point has perfect agreement between simulation and experiment; however, all remaining points disagree significantly. Clearly, if experimental and simulated charge management is to be in agreement, the components themselves must be modeled more accurately with respect to charge inventory, and piping must be accounted for.

To gain some insight to the difference between the simulation results and experimental results, Figure 5-22 presents the measured and simulated system suction and discharge pressures for the R-290 test data.

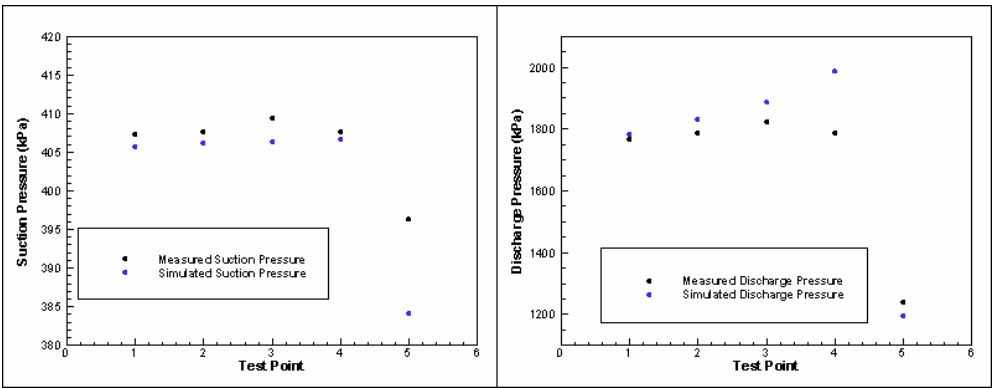


Figure 5-22: Suction and Discharge Pressure for R290 Test System

The figure reveals that the experimental pressures are not varying from one test to the next in the same manner as the simulation pressures. This phenomenon is consequence of two main issues: (1) refrigerant charge inventory models different from reality, and (2) off design component performance differing from experimental performance.

Table 5-10 and Table 5-11 provide numerical comparisons of both the R-290 system and the R-404a system. Comparisons of the simulated evaporator heat load, Q_{Evap} , COP, mass flow rate, corrected system charge, suction and discharge pressures to the measured values from the Test Project are provided.

		Q_{Evap} (kW)	COP	Massflow (g/s)	Charge ⁸ (g)	$P_{Suction}$ (kPa)	$P_{Discharge}$ (kPa)
Test 1	Measured	11.44	1.935	41.70	365	407.2	1764.9
	Simulated	11.48	1.927	42.67	365	405.7	1781.1
	% Deviation	-0.35	0.41	-2.32	0.0	0.37	-0.92
Test 2	Measured	11.59	1.943	41.62	456	407.6	1786.1
	Simulated	11.50	1.890	42.65	385	406.1	1829.6
	% Deviation	0.78	2.73	-2.48	15.6	0.37	-2.44
Test 3	Measured	11.74	1.940	41.75	546	409.3	1820.8
	Simulated	11.47	1.832	42.22	400	406.3	1884.4
	% Deviation	2.30	5.57	-1.13	26.74	0.74	-3.49
Test 4	Measured	11.59	1.870	41.62	637	407.6	1786.1
	Simulated	11.43	1.739	42.35	423	406.6	1985.0
	% Deviation	0.52	7.00	-1.75	33.60	0.25	-11.1
Test 5	Measured	13.47	3.911	42.73	456	396.4	1238.9
	Simulated	12.97	3.800	42.70	365	396.1	1250.5
	% Deviation	3.71	2.84	0.07	19.96	6.89	-0.94

Table 5-10: Measured vs. Simulated Data for R-290 Test System

⁸ Corrected Charge

		Q _{Evap} (kW)	COP	Massflow (g/s)	Charge ⁹ (g)	P _{Suction} (kPa)	P _{Discharge} (kPa)
Test 1	Measured	10.94	1.669	100.1	866	503.1	2422.5
	Simulated	10.94	1.668	100.0	866	405.7	1781.1
	% Deviation	0.0	0.06	0.10	0.0	19.36	26.48
Test 2	Measured	11.46	1.709	99.9	1320	504.2	2515.1
	Simulated	11.10	1.670	101.2	920	406.1	1829.6
	% Deviation	3.14	2.28	-1.30	30.30	19.46	27.26
Test 3	Measured	11.41	1.621	98.5	1773	504.3	2709.6
	Simulated	11.22	1.690	99.2	1020	406.3	1884.4
	% Deviation	1.67	-4.26	-0.71	42.47	19.43	30.45
Test 4	Measured	13.75	2.558	100.0	1320	482.9	1743.3
	Simulated	13.25	2.480	100.9	920	406.6	1985.0
	% Deviation	3.64	3.05	-0.90	30.30	15.80	-13.86

Table 5-11: Measured vs. Simulated Data for R-404a Test System

5.8.5 Optimization of Charge

A goal of the test project is the determination of the optimal charge for a given set of components at constant superheat at the evaporator exit, with COP being the objective function. In the test project, several data points are taken for different system charges. As demonstrated previously, meaningful comparisons between measured and simulated charge are difficult to achieve with the current data. Figure 5-23 offers comparison of simulated versus measured data by illustrating the optimum charge as a function of condenser exit sub-cooling.

⁹ Corrected Charge

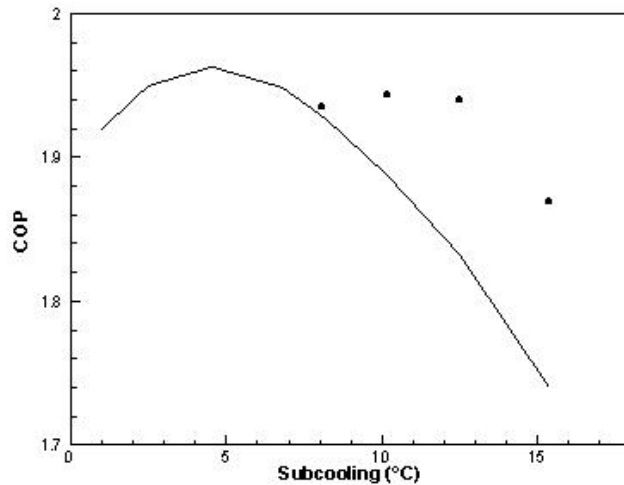


Figure 5-23: Measured and Simulated COP vs. Condenser Sub-cooling

The values of COP are similar in both cases, but the optimum COP is shifted to a lower value of sub-cooling for the simulation. This again is a consequence of both the component models off design performance and charge inventory behavior. The refrigerant charge inventory of a system is strongly related to the system's internal volume, which was not a focus of the of the current component models. To achieve better agreement, more data with respect to component charge management is needed, and the components need to be modeled to perform more accurately off the design point. The evaporator capacity, mass flow, suction and discharge pressure agree within 5% for a given evaporator superheat and condenser subcooling. Also, the character of the refrigerant charge curve with respect to condenser subcooling is consistent with test data. This demonstrates an opportunity to use VapCyc to tune the component models to help accurately simulate component charge as a function of operating point.

5.8.6 SEER Example

Determining the SEER of a given system using the standard requires the determination of the system's operation at two steady state operating points, namely the ASHRAE "A" and ASHRAE "B" rating points as defined in the standard. For reference Figure 5-24 illustrates the rating points.

ARI STANDARD 210/240-94									
Table 3. Operating Conditions for Standard Rating and Performance Tests Using Appendix A									
TEST		INDOOR UNIT				OUTDOOR UNIT			
		Air Entering				Air Entering			
		DB		WB		DB		WB	
		°F	[°C]	°F	[°C]	°F	[°C]	°F	[°C]
COOLING	Standard Rating Conditions "A" Cooling Steady State ¹	80	[26.7]	67	[19.4]	95	[35.0]	75 ²	[23.9]
	"B" Cooling Steady State	80	[26.7]	67	[19.4]	82	[27.8]	65 ²	[18.3]

Figure 5-24: ASHRAE "A" and "B" Rating Conditions for Air-Conditioning Systems

The performance of the Alpha 3600 air-conditioning system, with the nominal system charge (0.467 kg) at the ASHRAE "A" and "B" conditions is given in Table 5-12. Using these values for system performance, in accordance with ASHRAE standard 210/240-94 yields a system SEER of 13.76

ASHRAE Test	Q (Btu/hr / kW)	Power (Btu/hr / kW)	COP
A	12,490 / 3.66	3,316 / 0.92	3.77
B	12,523 / 3.67	2,771 / 0.82	4.52

Table 5-12: Alpha 3600 System Performance at ASHRAE "A" and "B" Conditions

Figure 5-25 illustrates the variation in the Alpha 3600 SEER as a function of system charge. It is noted here that as system charge increases, system superheat at the compressor suction port decreases, as sub-cooling at the condenser exit increases. The increase in SEER is due to this increase in sub-cooling. The increase in COP is seen to

level off at higher charges, as the condensing temperature of the system decreases and the condenser (refrigerant) exit temperature approaches the ambient air temperature cooling it.

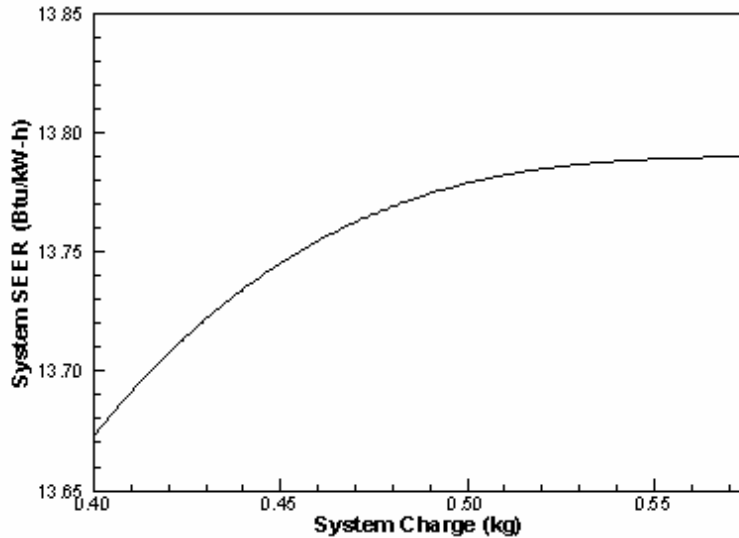


Figure 5-25: Alpha 3600 System SEER vs. System Charge

For a given operating point, the charge has a maximum value, as some degree of suction superheat must be maintained. The Alpha 3600 system must not exceed 0.57 kg of refrigerant charge for the ASHRAE “B” in order to maintain some degree of suction superheat.

5.8.7 Application Examples: Expandability of the Simple System

5.8.7.1 Evaporators in Parallel

The object oriented structure of the VapCyc simulation allows for the simple four component vapor compression system to be expanded at run time. The object oriented nature of the Junction Solver, which is the underlying solution engine allows a single solver to be utilized to solve many different systems.

A common practice in refrigeration and air conditioning is to serve several evaporators off of a single compressor and condenser. Figure 5-26 illustrates a “split” system, where five evaporators of varying size are connected in parallel to a single compressor, condenser and expansion orifice in series.

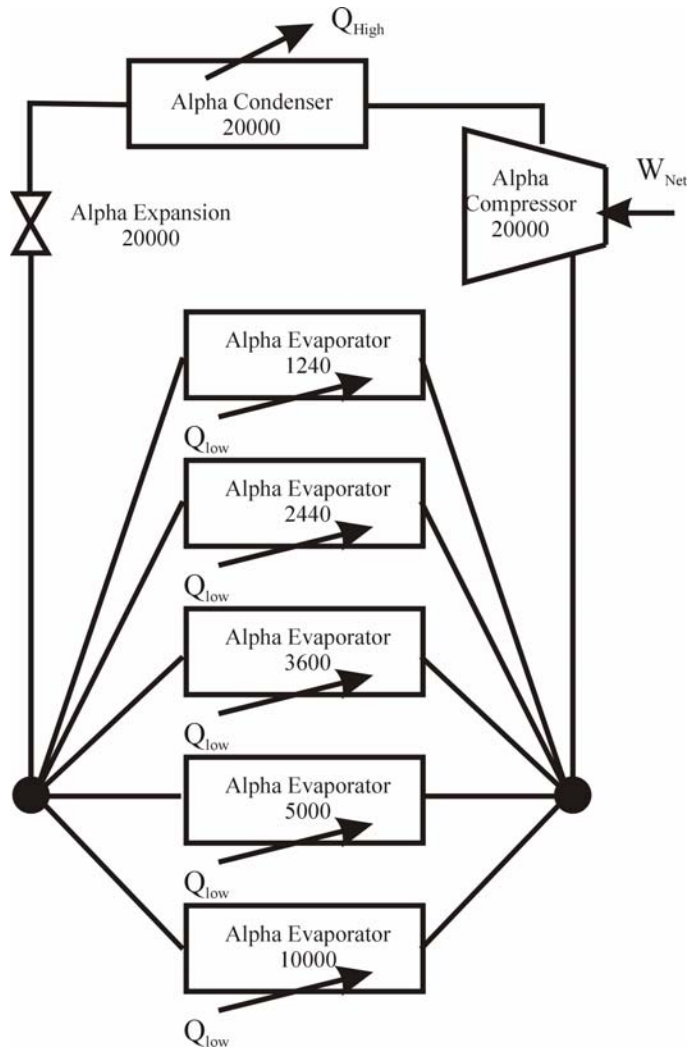


Figure 5-26: A Nominal 20 kW Alpha Split Air Conditioning System

Such a system may serve as an air conditioning system, where each evaporator would serve as the cooling coil for individual rooms requiring different cooling loads. The

system illustrated in Figure 5-26 is a nominal 20 kW system, consisting of the Alpha Compressor 20000, Alpha Condenser 20000, Alpha Orifice 20000 and a set of evaporators, Alpha 1240, 2440, 3600, 5000, and 10000. The system is configured in VapCyc, and the system charge is chosen to achieve acceptable levels of compressor suction superheat and condenser subcooling.

Table 5-13 displays the mass flow rate, power consumption and heat rejection by the individual components and the total system.

Component	Mass Flow Rate (g/s)	Power Out (kW)	Heat Out (kW)
Evaporator 1240	7.35	0.0	-1.16
Evaporator 2440	14.46	0.0	-2.29
Evaporator 3600	21.40	0.0	-3.39
Evaporator 5000	29.67	0.0	-4.70
Evaporator 10000	59.35	0.0	-9.41
Compressor 20000	132.22	-5.45	0.0
Condenser 20000	132.22	0.0	26.41
Orifice 20000	132.22	0.0	0.0
System Total	132.22	-5.45	5.46

Table 5-13: Results for Nominal 20 kW Alpha Split Air Conditioning System

As expected, the total flow rate through the evaporators is equivalent to that of the other three components. The evaporators each deliver different amounts of cooling to their specific zones, and the overall system energy balance is obeyed. This example demonstrates the expandability of VapCyc, and the underlying Junction Solver, and thus the flexibility of the simulation tool and simulation technique.

5.8.7.2 Compressors in Series

The Delta compressor presents a more realistic behavior of a compressor's isentropic efficiency versus its compression ratio. Figure 5-27 illustrates the performance of two systems as the ambient air temperature for the condenser varies. Each system contains an

Alpha3600 condenser, Alpha3600TXV (set for 10K superheat) and an Alpha3600 evaporator. One system contains an Alpha3600 compressor and the other a Delta3600 compressor. Recall, the Alpha compressor has a constant isentropic efficiency of 0.65, and the Delta compressor has a parametric isentropic efficiency, which varies with the compressor compression ratio, and has a value of 0.65 at the compression ratio of 3.42. It is shown in the figure that the system COP varies with the ambient air temperature, and this effect is more pronounced when the (more realistic) Delta compressor is used.

A combination of two Delta compressors in series can be used, rather than a single Delta compressor, when the rotational speed of the second compressor is reduced. Simulation demonstrates an increase in system COP by utilizing two delta compressors in series, rather than one.

Figure 5-27 illustrates the COP versus ambient air temperature of a system utilizing the single Delta compressor and two Delta compressors in series. The figure demonstrates that two Delta compressors in series can provide improvements in system COP at higher than design ambient temperatures.

It is illustrated in the figure that the system COP curve shifts up or down, depending upon the second compressor's rotational speed. The design speed of the second compressor would therefore be decided based upon the off design operation of the system.

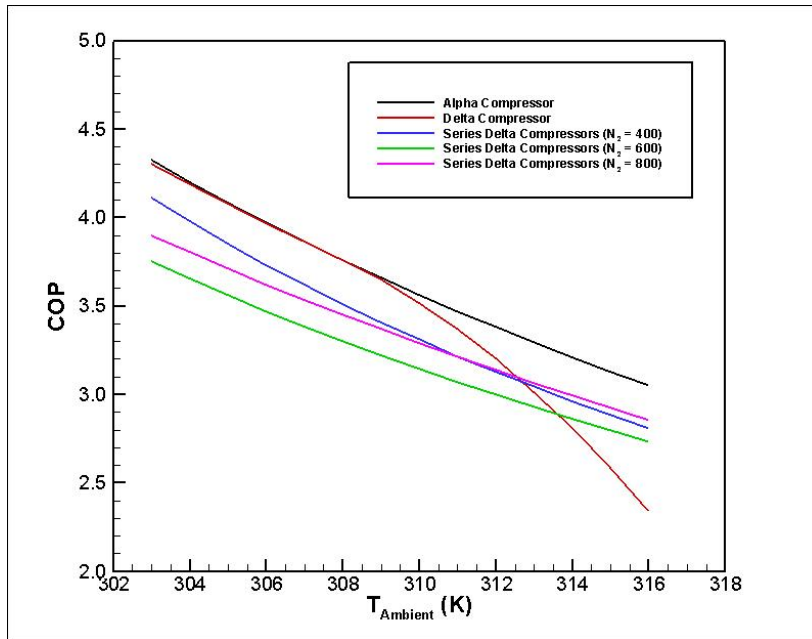


Figure 5-27: System COP versus Ambient Temperature

5.8.8 Parametric Study of COP With Respect to Charge and Air Velocity

Consider a situation where a set of vapor compression refrigeration components are supplied for a given application, such as the ASHRAE “A” condition, per the ARI standard 210/240. Specifically, consider the case of an Alpha 3600 compressor, Alpha 3600 Evaporator, Alpha 3600 Orifice and an Eta 3600 condenser.

VapCyc can be used to optimize the system. Figure 5-28 is a representation of the performance of such a system.

The performance is viewed as a function of system charge and condenser air velocity. The figure has lines of constant COP flooded with color and lines of constant cooling

capacity. Evident from the figure is the two dimensional maximum in COP.

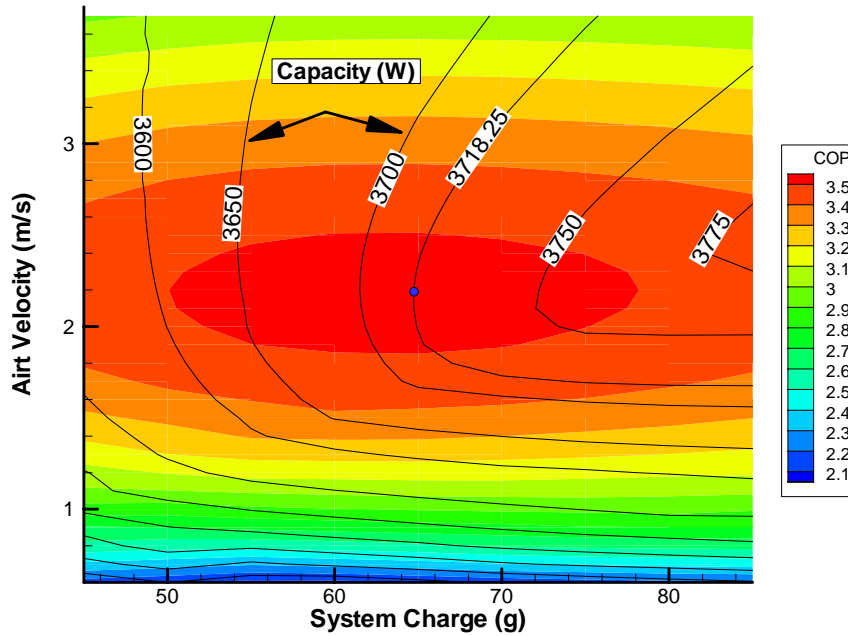


Figure 5-28: COP and Capacity versus Charge and Air Velocity

This system has a maximum COP of 3.53 when the condenser air velocity is 2.19 m/s and the system charge is 64.7g.

5.9 Summary

This chapter summarizes VapCyc, a comprehensive and robust tool for steady-state vapor compression refrigeration system simulation. The objective of the tool is for analysis, design and optimization of vapor compression refrigeration systems and their components. To mimic the behavior of real vapor compression refrigeration systems, a list of features including: charge management, component inter-changeability, a library of available refrigerants, a variety of unit systems, a series of physical variables, and the ability to utilize economic variables are integral to VapCyc. The tool can be used in an

interactive mode for simulating a single operating point as well as an automated mode for parametric analysis.

The VapCyc calculation engine is an object oriented entity, based upon the generalized energy system simulation principals given in chapter 2 of this thesis. The component library is built upon the component models given in chapters 4-6 of this thesis.

A major simulation challenge is the solution of the mathematical model. This thesis develops a simulation framework, and tool, the Junction Solver, to create systems in a general manner. The challenges of providing an initial guess and proper scaling of the conservation equations and independent variables fall upon the simulation program employing the Junction Solver. In this chapter, a novel concept called the *system context*, which is a logical extension of rating equipment, is developed. The system context is a link between the specific system being simulated and the general nature of the Junction Solver. Several contexts are defined for several different vapor compression refrigeration applications. The context allows for communication between the system level solver, and the component level models. The system level solver can utilize information provided by components for the specified context to intelligently develop an initial guess, and diagnose potential disastrous system configurations, composed of mis-matched component models.

VapCyc allows the research concerns of individual users to be addressed. These concerns include:

- The analysis and performance of particular pieces of hardware, over a range of operating conditions

- A forum for sharing of performance information, while maintaining propriety of such information
- A tools to access/publish/share this performance information
- An assembled libraries of components which can be used to assemble full systems
- The ability to explore the performance of specific systems over a range of operating conditions to which the system may be subject
- Tools to access/publish/share this system performance information
- An environment for re-usable experiments/simulations
- Component level experiments/simulation
- System level experiments/simulation
- Flexible and adjustable system design for experiments/simulations
- Interchangeable components
- Complete separation of component models from system models, enabling
 - Research to be conducted on systems, with individual component models being independent variables to the experiment/simulation
 - Research to be conducted on individual components
 - Research to be conducted on simulation solvers independently from components
- User friendly software
- Common simulation framework over multiple system simulations
- A component/system simulation environment that closely mimics that of the physical experimental environment

5.10 References

Air-Conditioning and Refrigeration Institute, 1994, *Unitary Air Conditioning and Heat Pump*, ANSI/ARI Standard 210/240.

Air-Conditioning and Refrigeration Institute, 1999, *Positive Displacement Refrigerant Compressors and Compressor Units*, ANSI/ARI Standard 540.

ASHRAE Handbook: Equipment, American Society of Heating, Refrigerating and Air-Conditioning Engineers, 1988

Copeland Corporation, Division of Emerson Climate Technologies, On-Line Catalog, 2004, <http://www.copeland-corp.com/americas/entercataloginfo.htm>

Davis, G.L., Scott, T.C., 1976. "Component Modeling Requirements For Refrigeration Systems Simulation", *Proceedings of the 1976 Purdue Compressor Technology Conference*, West Lafayette, USAA, July 1976, pp. 401-408

Fischer, S.K., and Rice, C.K., 1986 "System Design Optimization and Validation for Single-Speed Heat Pumps", *ASHRAE Transactions*, Vol. 91, Part 2

Hiller, C.C., Glicksman, L.R., 1976, "Improving Heat-Pump Performance Via Compressor Capacity Control – Analysis And Test". Volumes I and II, *Energy Laboratory Reports MIT-EL 76-0001 and MIT-EL 76-0002*, Department of Mechanical Engineering, Massachusetts Institute of Technology, Massachusetts, USA (January 1976)

Hwang, Y., Dea-Huan, J., Radermacher, R., "Comparison Of Hydrocarbon R-290 And Two Hfc Blends R-404a And R-410a For Medium Temperature Refrigeration Applications", *ARI: Global Refrigerant Environmental Evaluation Network (GREEN) Program*, Final Report, March 2004

Kreith, F., 1999, *Mechanical Engineering Handbook*, Boca Raton: CRC Press LLC, 1999

Levins, W., Rice, C.K., Baxter, V.D. 1996. "Modeled And Measured Effects Of Compressor Downsizing In An Existing Air Conditioner/Heat Pump In The Cooling Mode", *ASHRAE Transactions*, V 102, Pt 2

NIST, 2001, NIST Thermodynamic and Transport Properties of Refrigerants and refrigerant Mixtures-REFPROP Version 7.0, National Institute of Standards and Technology, Boulder, Colorado, USA.

Rice, C.K., Fischer, S. K., Ellison, R.D., Jackson, W.L. 1981. "Design Optimization of Conventional Heat Pumps: Application to Steady-State Heating Efficiency," *ASHRAE Transactions*. Vol. 87, pp1037-1054

Rice, C.K., 1987 “The Effect of Void Fraction Correlation and Heat Flux Assumption on Refrigerant Charge Inventory Predictions”, *ASHRAE Transactions*. Vol. 93, part 1, pp 341-367

Richardson, 1998. *Performance Models For Use in Computer Simulation of Energy Systems*, Masters Thesis, Department of Mechanical Engineering, Marquette University, 1998

Stoeker, W., *Refrigeration and Air Conditioning*, McGraw-Hill, 1958

Tecumseh Products, Model Chart, 2004, <http://www.tpc-nacg.com/mdlnmbr1.htm>

Tandon, A., *Object-Oriented Modeling of Vapor Compression Systems and Components*, M.S. Thesis, Department of Mechanical Engineering, University of Maryland, College Park, 1999.

Webb, R.L. 1994, *Principals of Enhanced Heat Transfer*, John Wiley & Sons, New York.

6. Conclusions

The objectives of this thesis are:

1. Develop and create a simulation framework, and subsequent simulation tool in an object oriented manner, which provides the ability to simulate experiments in a laboratory. The framework is developed with the following fundamental objectives:
 - a. Serve as a functional starting point which will appeal to the individual objectives of each individual user with a wide range of backgrounds: industrial uses (development of models and/or real equipment), academic professionals (utility of the simulation tool without an intimate understanding of its inner workings), and students (re-using of existing software to facilitate the growth of the project(s))
 - b. Serve as a simulation tool such that each component within the system is a stand-alone simulation model, enabling the analysis and if desired optimization of individual simulation models and/or the system as a whole
2. Verify the simulation tool with experimental data

6.1 Thermal Energy System Simulation Framework

The generalized methodology is built upon the structures of: components, ports and junctions.

Components are individual energy systems themselves, which are connected by the user at run time to create a larger system. Each component has ports with which

communication exist via junctions. Similar in nature to real world devices, boundary conditions are applied to a component, and a steady state solution results.

Through the iteration on junction independent variables, such as pressure and enthalpy, system level conservation equations are solved, consequently simulating the system.

6.2 Basic Components

In this thesis, the energy interactions for the simulation of the vapor compression refrigeration system are limited to thermo fluid interactions, such as thermal energy, heat transfer and work. Components are developed as sub-systems to be combined into larger systems.

Each type of component has unique characteristics which govern its specific relationships between boundary conditions; pressure and inlet energy state, to mass flow, heat transfer and work output. Individual components determine outlet conditions as well as heat/work interactions via a common set of boundary conditions. A generalized energy system simulation using such components can be constructed utilizing system level intensive variables as the component boundary conditions. In this particular case, the system level intensive variables; junction pressure and junction exit specific enthalpy are employed and communicated to components, via ports.

6.3 Component Models

Building energy systems from components and utilizing these energy systems as components in a larger system presents mathematical challenges for simulation. Two main challenges are solving the sub-system models to achieve a desired solution for a

system, and the mathematical character of the partial differential equations which describe the physics being modeled.

The main rule to focus upon when modeling components for use in larger energy systems are determined to be: proper scaling of independent variables and balance equations. The scaling is shown to affect the solution process, particularly when gradient based solution methods are employed.

The independent variable scales must be indicative of the physical phenomena being modeled. This thesis exploits component level boundary conditions for the scaling of independent variables.

6.4 Compound Energy Systems

Chapter four of this thesis broaches the concept of a compound energy system, which is the combination of two or more hydraulically independent systems. In chapter four, a refrigerant circuit and an air circuit are combined to make a heat exchanger. Later, in chapter five, a compressor, condenser, expansion device and evaporator are combined into a vapor compression refrigeration system. Also in chapter five, additional components are added, demonstrating the ability to create energy systems in an arbitrary fashion by combining individual components.

Compound energy systems have the same scaling issues as basic components. The scaling parameters for both independent variables and balance equations should be indicative of both independent systems. In the case of the air to refrigerant cross flow heat exchanger example, the air entrance enthalpy, coupled with the boundary conditions on the refrigerant side allow scaling values to be calculated.

Uncertainty propagation between components and the system generally needs to begin at the lowest levels, such that when propagated to the highest level, the convergence criterion of the highest system is met.

The character of the mass flow relations for the compound energy system is elliptic, while the energy relations are essentially hyperbolic. The flow arrangement of the two fluids however need not be hyperbolic in the same direction. Consequently, some form of de-coupled mass and energy relations are necessary. A fractional step method provided by Jiang (2003) is examined. Additionally, more complex methods which couple the mass and energy relations of the two systems to varying degrees are developed. Ultimately a robust method for solving the example compound system is developed.

The heat exchanger model developed in this thesis has been validated and verified against actual test data.

6.5 VapCyc

This thesis reaches a culmination with VapCyc, a comprehensive and robust tool for steady-state vapor compression refrigeration system simulation. The objective of the tool is for analysis, design and optimization of vapor compression refrigeration systems and their components. VapCyc serves as a simulation tool such that each component within the system is an individual program, enabling the analysis and if desired optimization of individual simulation models and/or the system as a whole

This simulates the behavior of real vapor compression refrigeration systems, where a list of relevant component and/or system features includes: charge management, component inter-changeability, a library of available refrigerants, a variety of unit

systems, a series of physical variables, and the ability to utilize economic variables are integral to VapCyc. The tool can be used in an interactive mode for simulating a single operating point as well as an automated mode for parametric analysis.

The VapCyc calculation engines, as well as its component library are object oriented entities, based upon the generalized energy system simulation principals.

The components themselves are completely unrelated unless connected to one another via junctions. Therefore, neither the system level balance equations, nor the independent variables can be scaled relative to the component models.

Overcoming the complexity of formulating an initial guess is accomplished through a novel concept called the system context, which is a logical extension of rating equipment. Several contexts are defined for several different vapor compression refrigeration applications. The context allows for communication between the system level solver, and the component level models. The system level solver can utilize information provided by components for the specified context to intelligently develop an initial guess, and diagnose potential disastrous system configurations, composed of mismatched component models.

Several examples demonstrate how the solution technique is employed in a flexible setting. Single point simulations, parametric analyses and optimizations are possible. Independent variables for the simulations are not limited to component independent variables, but the refrigerant, system charge and even component model.

VapCyc allows the research concerns of individual users to be addressed. These concerns include:

- The analysis and performance of particular pieces of hardware, over a range of operating conditions
- A forum for sharing of performance information, while maintaining propriety of such information
- A tools to access/publish/share this performance information
- An assembled libraries of components which can be used to assemble full systems
- The ability to explore the performance of specific systems over a range of operating conditions to which the system may be subject
- Tools to access/publish/share this system performance information
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- Component level experiments/simulation
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- Complete separation of component models from system models, enabling
 - Research to be conducted on systems, with individual component models being independent variables to the experiment/simulation
 - Research to be conducted on individual components
 - Research to be conducted on simulation solvers independently from components
- User friendly software
- Common simulation framework over multiple system simulations

- A component/system simulation environment that closely mimics that of the physical experimental environment

In this thesis, VapCyc has been validated and verified against actual test data.

6.6 Future Work

Future work opened up by this research takes several different forms.

Mathematically, there is much to be done. Although the complex nature of the momentum and energy relations cannot be determined before run time, there is much room in a generalized “junction solver” to interpret the solution process.

The generalized energy system process can be either applied to different systems and/or expanded. With energy transport being similar in nature from energy form to energy form, it is conceivable that many types of energy be incorporated. The generalized solver which is used to solve the simplified system given in VapCyc, is easily altered to include kinetic energy, potential energy electro-magnetic energy.

VapCyc has many potential uses. The component swapping capability, combined with the evolutionary optimization techniques make it an ideal candidate to create in industry wide simulation tool. Creating a component library from every major manufacturer, would allow a systems engineer combining equipment into systems to explore thousands of potential systems very quickly.

6.7 Major Accomplishments

This thesis accomplished the following objectives:

1. Develop and create a simulation framework in an object oriented manner, which provides the ability to simulate experiments in a laboratory

2. Develop a solution algorithm for the simulation tool where the system solution is separate from individual components
3. Create a heat exchanger simulation, operating with this tool
4. Verify the simulation tool with experimental data
5. Create VapCyc, a comprehensive (Graphical User Interface + Junction Solver + Components) steady state vapor-compression refrigeration simulation tool

The thesis developed and created the Junction Solver. The Junction Solver is an object oriented structure of premised upon the data structures of *components, ports and junctions* which are used to virtually construct thermo thermal energy systems. As detailed in chapter 3, the methodology allows for components to be treated as black box entities, which conform to specific input/output standards. The input/output parameters to the component models are intensive thermodynamic variables. This allows the individual component function to be separate from the overall system governing equations. The simulation is then deemed to be multidisciplinary in nature, as each component can be considered as an individual discipline, of which the Junction Solver (coordinating system) is independent. The Junction Solver provides the abilities to simulate experiments as in a laboratory, such as:

1. The ability to constrain and specify the mass of working fluid for the simulation of system simulation. For a vapor-compression refrigeration application, this is referred to as “Charge Management”.
2. The functionality to change, or swap, component models used in a fixed-component simulation at run-time thus replacing components without the need to re-compile the simulation code itself.

3. The functionality to add or subtract components at run time, without the need to re-compile the simulation code itself.
4. Allow user defined component connectivity.
5. Allow for the selection from several different working fluids/refrigerants.
6. Flexibility to alter component independent variables, independently from the main simulation program.

The heat exchanger simulation detailed in chapter 4 was used to verify and validate the Junction Solver. The heat exchanger model created to simulate a specific heat exchanger, and the model was verified against the experimental data.

VapCyc, a vapor compression refrigeration system simulation tool was created in chapter 5. As a specific application built upon the generality of the Junction Solver, the system context concept was created to overcome the numerical challenges of scaling and initial guess values. Examples are used to demonstrate the validity of the scaling provided by the system context and demonstrate the flexibility and expandability the Junction Solver provides to a specific application, such as VapCyc.

6.8 Major Engineering Accomplishments

This thesis advanced the state of thermo-fluid energy system simulation, specifically thermo fluid systems such as heat exchangers, pipe networks and closed loop systems such as vapor compression refrigeration. A generalized framework and simulation tools were developed. Engineering principles were applied to the framework and tools to demonstrate its use and develop an advanced vapor compression refrigeration simulation tool. Key engineering accomplishments differentiate this work from previous work, namely:

1. The framework has scaling of independent variables and conservation equations at the solver level. This frees the solution algorithm from determining application specific variables, allowing development of the mathematics separately from engineering.
2. The selection of dependent and independent variables. The Junction Solver has a set of independent variables, consisting of thermodynamic properties, which are intensive variables. These independent variables are the input to sub-system components, which then provide a set of dependent variables, of mass and energy flow rate through its ports, as well as the mass of working fluid contained within the component at the given operating point. The Junction Solver is resolved using the mass and energy flow rates flowing between components, and for closed systems, a total system mass balance.

The benefits of these choices are:

- a. While typical simulation programs utilize working fluid flow rates as independent variables. This is problematic:
 - i. When flows split, and/or flow direction is not known a priori.
 - ii. The order of magnitude of the flow throughout the system is not known. For example, a 1kW air conditioner and a 1000kW air conditioner have radically different flow rates, yet are essentially the same system.
- b. Pressure and Enthalpy are logical choices for system independent variables, as they are independent of system size, mass flow rate and energy flow rate.

3. A comprehensive, yet flexible simulation framework, which also allows for charge management.
4. With consistent independent variables, the framework does not require differentiation of components
 - a. Traditional simulations had different dependent/independent variables for different types of components. For example, compressors/valves may have boundary pressures and internal state as input and gave mass flow rate and exit state as output, while heat exchangers may have mass flow rate and inlet state as input and provided outlet state as output. This required a solution algorithm with system configuration knowledge.
 - b. The Junction Solver allows components to be connected in any order, and allows for a solution algorithm independent of component evaluation order.
 - c. The system configuration can change arbitrarily at run time, with no need to re-compile.
5. Implementation of the System Context: a vehicle for communication between components and the system which they create prior to attempting to resolve the simulation.
 - a. Simulation program can determine potential component mismatches.
 - b. Simulation program can provide appropriate scaling values to the junction solver based upon potential system operation.
 - c. Simulation program can provide an initial guess to the system solution.
 - d. Simulation program can provide an estimate of the system charge.

6. An intuitive interface that is useful to engineers with a diverse set of objectives
 - a. Component simulation
 - b. System simulation
 - c. Variation of working fluid
 - d. Solver development

A1. TEST System

A1.1 The TEST Compressor Model

The TEST Compressor models are modified versions of the ARI 540-99 model, as presented in chapter 5. The modification is slight, correcting for an “off design” operation. The test points given in the standard all correspond to a certain return gas temperature to the compressor. In reality, a change in temperature at constant pressure will change the gas density and thus affect the mass flow rate through the compressor.

Tandon (1999) introduces a scale factor of the form

$$s = \frac{\rho(P_{actual}, T_{actual})}{\rho(P_{actual}, T_{standard})}, \quad (A1.1)$$

where the actual density (ρ) is scaled by the design (standard) density for the given pressure. This scale factor, s , is used to correct the model mass flow rate for the difference between the actual temperature and the design temperature given in the standard.

Figure A1- 1 and Figure A1- 2 illustrate the comparison of the Green Compressor models (both R-404a and R-290) and the measured performance data from the report (Hwang et. al 2004). It is clear from both figures that, at least at the measured data points, the model predicts the compressor mass flow rate and power requirement within +/- 5%.

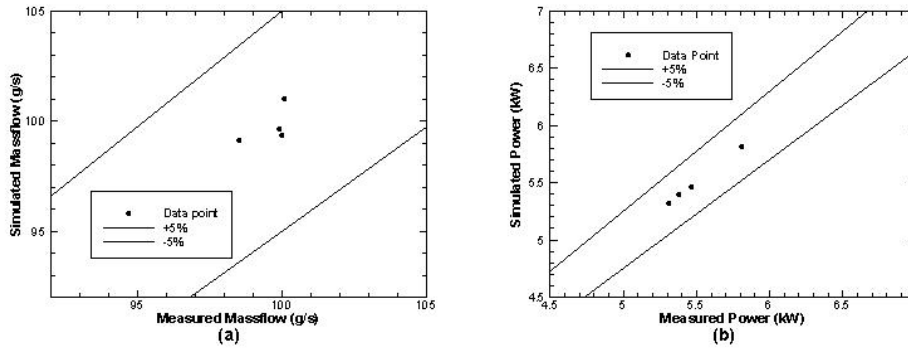


Figure A1- 1: TEST Compressor R-404a

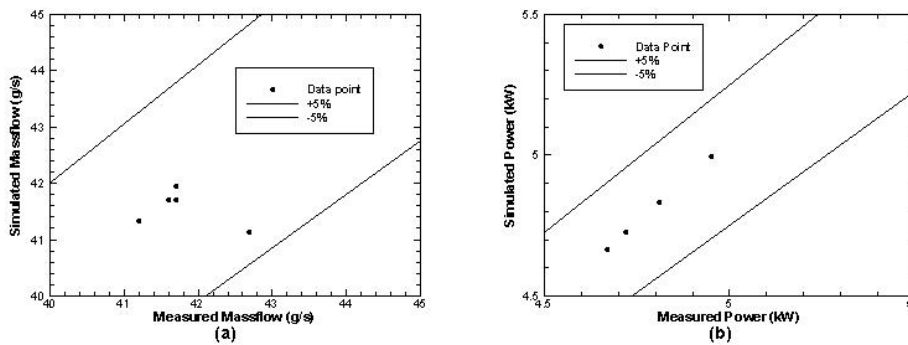


Figure A1- 2: TEST Compressor R-290

A1.2 The TEST Orifice Model

The TEST orifice is based upon the generic orifice, as presented in chapter 5. The orifice model has two degrees of freedom, namely the constant K and diameter D from equation (5.3). Using the first test point from test project data (Hwang, et. al 2004), and assuming a value of $K=1$, the TEST orifice is sized to give the corresponding mass flow rate.

Figure A1- 3 graphically displays the agreement between experimentally measured and calculated mass flow rate for the TEST orifice.

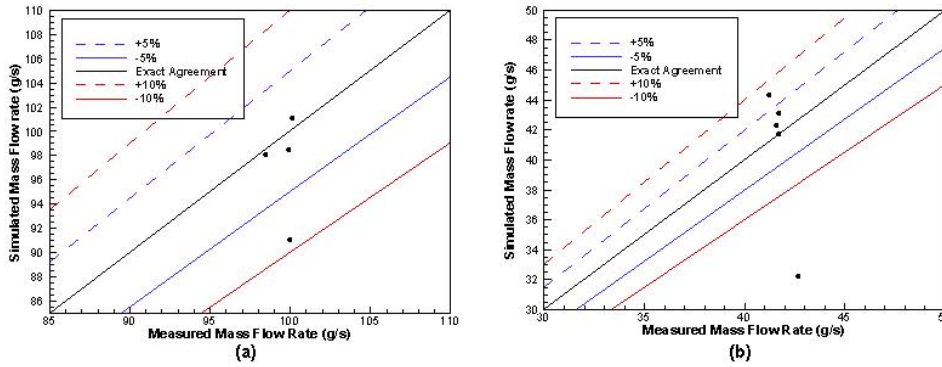


Figure A1- 3: TEST Orifice Performance

The figure shows some scatter in the data, with most points falling within 5% of the measured data. In both the R-404a and R-290 orifice, the deviation for the off-design test point is greater than for the others.

A1.3 The TEST Evaporator Model

The TEST evaporator is based upon the generic evaporator, as presented in chapter 5.

The length and diameter of the generic evaporator are sized such that the mass flow rate and refrigerant energy increase (refrigeration capacity) agree with the test data at test point 1 (see appendix I).

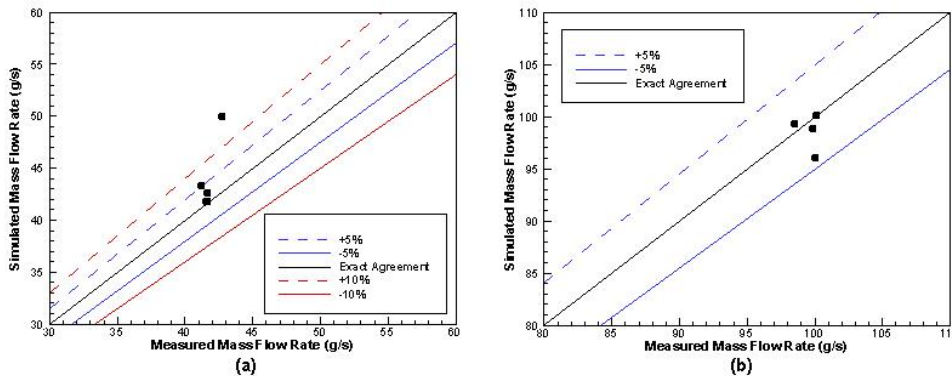


Figure A1- 4: Simulated Versus Measured Mass Flow Rate for TEST Evaporator (a) R-290 and (b) R-404a

Figure A1- 4 - Figure A1- 6 show the agreement of the simulated results versus the measured test points.

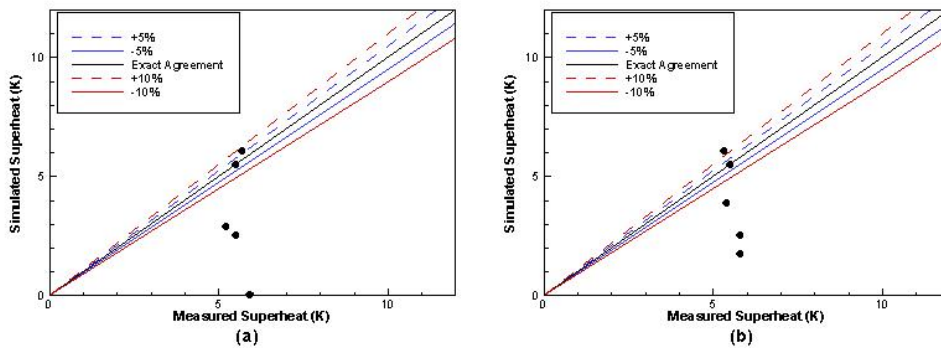


Figure A1- 5: Simulated Versus Measured Superheat for TEST Evaporator R-290 and (b) R-404a

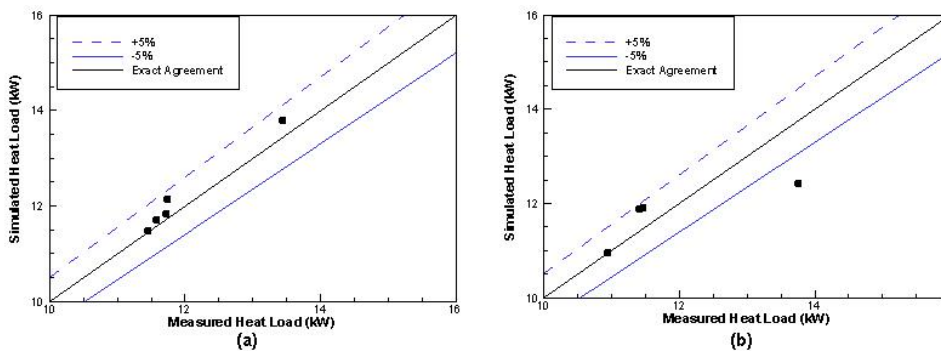


Figure A1- 6: Simulated Versus Measured Heat Load for TEST Evaporator (a) R-290 and (b) R-404a

The TEST evaporator simulation agrees with measured data to within 5% for all cases, except the R-290 part load case. The simulated superheat, a reflection of the outlet condition for the evaporator does not agree very well. The large deviation in superheat, however does not necessarily represent a large deviation in overall performance, as demonstrated by the fairly good representation of evaporator heat load. In all cases

except the R-404a part load test, the simulated heat load agrees with the measured heat load within 5%.

A1.4 The TEST Condenser Model

The TEST condenser is the condenser modeled in chapter 4. Results for the condenser's performance are given in that chapter.

A1.5Raw Data

R290	3.8 lb	Tair Evap	1.59	Tair Cond	34.79	Mass flow Rate	41.72	Charge*	1.00
	Pin	Pout	Tin	Tout	hin	hout	Q	W	Power Consumption
Compressor	400.87	1764.94	3.19	82.16	583.54	695.74	0.00	4680.47	5.91
Condenser	1764.94	1667.04	80.07	40.86	690.98	310.16	15886.29	0.00	0.00
Expansion	1595.33	424.16	38.67	-4.32	303.83	303.83	0.00	0.00	0.00
Evaporator	424.16	407.21	-4.32	0.13	303.83	577.96	11435.75	0.00	0.91
						Energy Balance	229.92	COP	1.94
R290	4.0 lb	Tair Evap	1.76	Tair Cond	34.69	Mass flow Rate	41.62	Charge*	1.05
	Pin	Pout	Tin	Tout	hin	hout	Q	W	Power Consumption
Compressor	401.42	1786.14	3.42	83.18	583.91	697.47	0.00	4726.15	5.96
Condenser	1786.14	1695.40	81.04	39.34	692.58	305.73	16099.41	0.00	0.00
Expansion	1622.57	424.07	37.34	-4.37	299.99	299.99	0.00	0.00	0.00
Evaporator	424.07	407.60	-4.37	0.39	299.99	578.39	11586.20	0.00	0.90
						Energy Balance	212.94	COP	1.94
R290	4.2 lb	Tair Evap	1.67	Tair Cond	34.66	Mass flow Rate	41.75	Charge*	1.11
	Pin	Pout	Tin	Tout	hin	hout	Q	W	Power Consumption

Compressor	403.15	1820.76	3.12	84.08	583.32	698.55	0.00	4810.23	6.05
Condenser	1820.76	1735.47	81.89	38.01	693.50	301.87	16349.45	0.00	0.00
Expansion	1662.73	425.76	36.14	-4.29	296.54	296.54	0.00	0.00	0.00
Evaporator	425.76	409.32	-4.29	0.05	296.54	577.73	11738.99	0.00	0.91
						Energy Balance	199.77	COP	1.94
R290	4.4 lb	Tair Evap	1.76	Tair Cond	34.69	Mass flow Rate	41.62	Charge*	1.16
	Pin	Pout	Tin	Tout	hin	hout	Q	W	Power Consumption
Compressor	401.42	1786.14	3.42	83.18	583.91	697.47	0.00	4726.15	6.20
Condenser	1786.14	1695.40	81.04	39.34	692.58	305.73	16099.41	0.00	0.00
Expansion	1622.57	424.07	37.34	-4.37	299.99	299.99	0.00	0.00	0.00
Evaporator	424.07	407.60	-4.37	0.39	299.99	578.39	11586.20	0.00	0.90
						Energy Balance	212.94	COP	1.87
R290	4.0 lb - PL	Tair Evap	1.59	Tair Cond	18.76	Mass flow Rate	42.73	Charge*	1.05
	Pin	Pout	Tin	Tout	hin	hout	Q	W	Power Consumption
Compressor	389.41	1238.89	1.53	59.84	581.16	662.54	0.00	3478.05	4.71
Condenser	1238.89	1130.82	57.95	25.26	658.56	266.25	16765.38	0.00	0.00
Expansion	1080.01	413.11	23.89	-5.26	262.51	262.51	0.00	0.00	0.00
Evaporator	413.11	396.35	-5.26	-0.36	262.51	577.59	13464.99	0.00	0.91

						Energy Balance	177.66	COP	2.86
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R404a	10 lb	Tair Evap	1.61	Tair Cond	35.50	Mass flow Rate	100.14	Charge*	1.00
	Pin	Pout	Tin	Tout	hin	hout	Q	W	Power Consumption
Compressor	503.06	2422.46	2.16	81.28	368.86	421.72	0.00	5293.63	6.56
Condenser	2422.46	16.00	79.83	40.94	419.87	259.74	-16035.44	0.00	
Expansion	2175.13	544.01	39.41	-3.67	257.28	257.28	0.00	0.00	
Evaporator	544.01	514.16	-3.67	0.05	257.28	366.56	10943.32	0.00	
						Energy Balance	201.51	COP	1.67
R404a	11 lb	Tair Evap	1.76	Tair Cond	35.15	Mass flow Rate	99.87	Charge*	1.10
	Pin	Pout	Tin	Tout	hin	hout	Q	W	Power Consumption
Compressor	504.19	2515.08	2.03	83.59	368.70	423.26	0.00	5448.74	6.70
Condenser	2515.08	2384.86	82.00	37.42	421.23	253.78	-16723.82	0.00	
Expansion	2276.97	544.52	36.09	-3.70	251.71	251.71	0.00	0.00	
Evaporator	544.52	515.12	-3.70	-0.04	251.71	366.46	11460.31	0.00	
						Energy Balance	185.23	COP	1.71
R404a	12 lb	Tair Evap	1.84	Tair Cond	35.08	Mass flow Rate	98.54	Charge*	1.20
	Pin	Pout	Tin	Tout	hin	hout	Q	W	Power Consumption
Compressor	504.33	2709.60	2.43	89.40	369.08	427.88	0.00	5794.44	7.04
Condenser	2709.60	2589.27	87.62	36.96	425.57	252.87	-17017.81	0.00	
Expansion	2481.26	543.50	35.63	-3.75	250.81	250.81	0.00	0.00	
Evaporator	543.50	514.76	-3.75	0.14	250.81	366.64	11413.61	0.00	

						Energy Balance	190.25	COP	1.62
R404a	11 lb - PL	Tair Evap	1.66	Tair Cond	18.34	Mass flow Rate	99.98	Charge*	1.10
	Pin	Pout	Tin	Tout	hin	hout	Q	W	Power Consumption
Compressor	482.88	1743.29	0.27	61.27	367.54	407.98	0.00	4042.97	5.38
Condenser	1743.29	1606.85	59.78	21.70	406.21	229.66	-17650.94	0.00	
Expansion	1527.59	523.96	20.99	-5.20	228.61	228.61	0.00	0.00	
Evaporator	523.96	495.25	-5.20	-0.84	228.61	366.17	13752.73	0.00	
						Energy Balance	144.75	COP	2.56

A1.6 References

Hwang, Y., Dea-Huan, J., Radermacher, R., “Comparison Of Hydrocarbon R-290 And Two Hfc Blends R-404a And R-410a For Medium Temperature Refrigeration Applications”, *ARI: Global Refrigerant Environmental Evaluation Network (GREEN) Program*, Final Report, March 2004

A2. VapCyc – The Program

A2.1 Basic Description

VapCyc is a general-purpose, steady state, vapor compression refrigeration system simulation tool. ISOC aspires to make VapCyc the *most advanced* vapor compression system modeling tool available. In this instance, most advanced, entitles a combination of simulation ability, coupled with functionality, currently not offered (at least in the public domain) in the field of vapor compression refrigeration simulation.

The desired applicability of VapCyc is system simulation, for the purposes of:

- Analysis of the system, as a function of component performance
- Analysis of a component, as a part of a system
- Optimization of system level boundary conditions
- Optimization of component level parameters

To achieve these goals VapCyc is a simulation of a fixed configuration vapor compression system, consisting of:

- 1 Compressor
- 1 Condenser
- 1 Expansion Device
- 1 Evaporator
- Expandable Components, both in Parallel and in Series with Four Main Components
- Piping Between Components
- Suction Line Heat Exchanger
- Accumulator

This system is general purpose in that each of the components can be selected and/or changed, and (if applicable) modified.

VapCyc has a library of components, consisting of “fixed” models, as well as “generic” components (components which allow some or all of their properties to be modified). Fixed components mimic “real life” components, in that their performance is dictated by their geometry, which cannot be changed.

An example of a fixed versus generic component is a compressor. A purchased compressor has a performance that is dictated by its boundary conditions and unchangeable geometry. A generic compressor allows the user to specify certain parameters. The Generic Compressor supplied with VapCyc has inputs for: Displacement Volume, RPM, Volumetric Efficiency and Isentropic efficiency.

It is desirable to offer a limited number of *system level* variables (those which would be controllable with a “real” machine) that can be modified by the user. The system level variables available in VapCyc are:

- The refrigerant type
- System charge
- System boundary conditions
 - Condenser coolant temperature (If applicable)
 - Condenser coolant flow rate (If applicable)
 - Evaporator heating fluid temperature (If applicable)
 - Evaporator heating fluid flow rate (If applicable)

A2.2 The Interface

The main VapCyc screen gives the user a snapshot of the basic system, with pictures representing the four main components, as shown in Figure A2- 1. Evident from the main interface page is the ability to change the refrigerant of the system, via a drop-down-box in the upper right hand corner. Also in this section, the user can specify the refrigerant charge, contained within the system. The check box to the right of the

refrigerant charge, allows VapCyc to calculate a nominal, or default charge, based upon the specified operating context and default charges for the currently selected components.

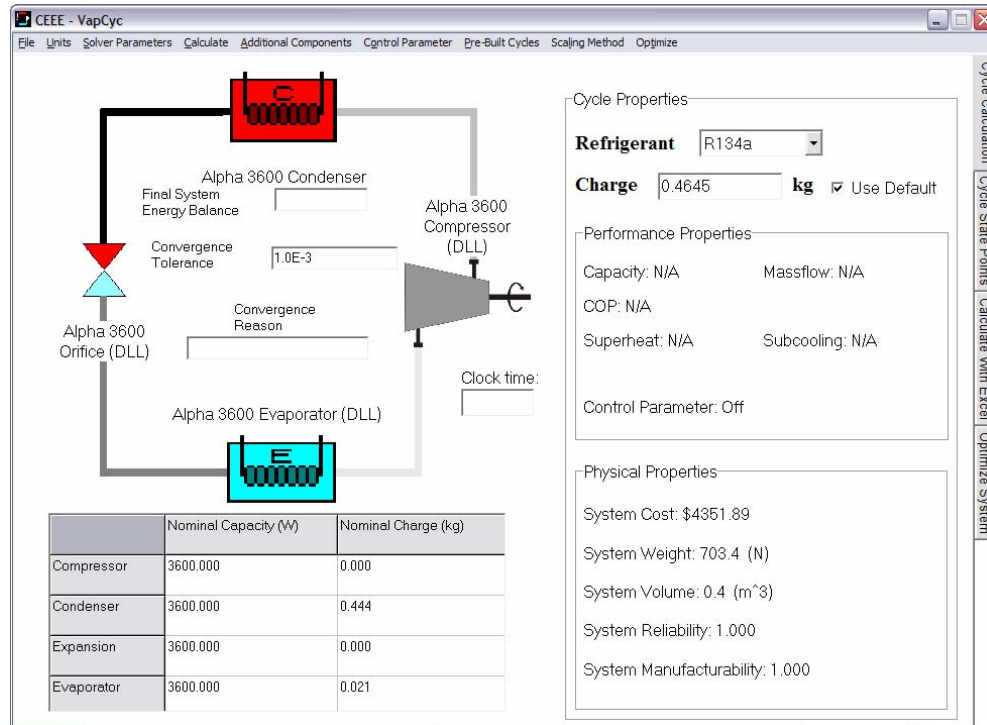


Figure A2- 1: VapCyc Main Interface Screen

In the middle of the right hand side of the main screen are some performance variables calculated by VapCyc when the simulation is run. Currently the system's cooling capacity, coefficient of performance (COP), superheat value at the evaporator outlet, condenser subcooling and refrigerant massflow are reported.

The bottom right hand corner of the screen shows performance variables which may be of interest. Physical variables of system cost, weight, volume are calculated as the sum of individual component values. System reliability and manufacturability have placeholders in the current version of the program, but are not yet fully functional.

A2.2.1 Choosing Components

Component models are pre-loaded, by default, the “Alpha3600” series is loaded. These components are designed to give a 3.6kW cooling capacity under air-conditioning conditions. Clicking the right mouse button while the cursor hovers over a component opens a dialog box allowing the user to select additional components. Figure A2- 2 demonstrated the selection of a different compressor from a list of available models.

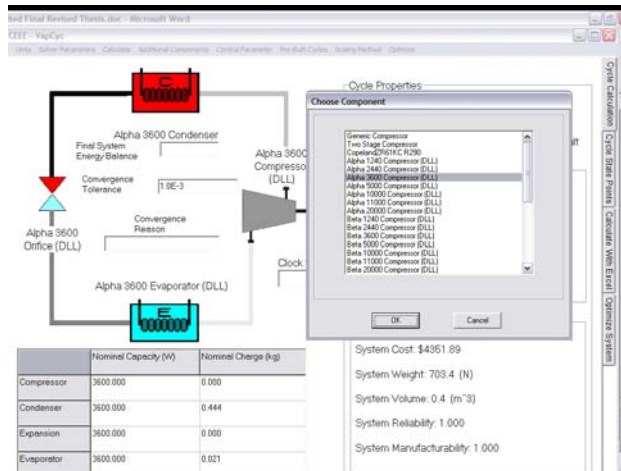


Figure A2- 2: Changing of a Component in VapCyc

When the cycle is defined, in terms of component models, refrigerant charge value and refrigerant, the system can be simulated by selecting “Run Current Cycle” from the Calculate menu. Alternatively, the user can press Cntrl-R as a short cut. At this point the solution values are displayed. Illustrates the result of running the default Alpha3600 system.

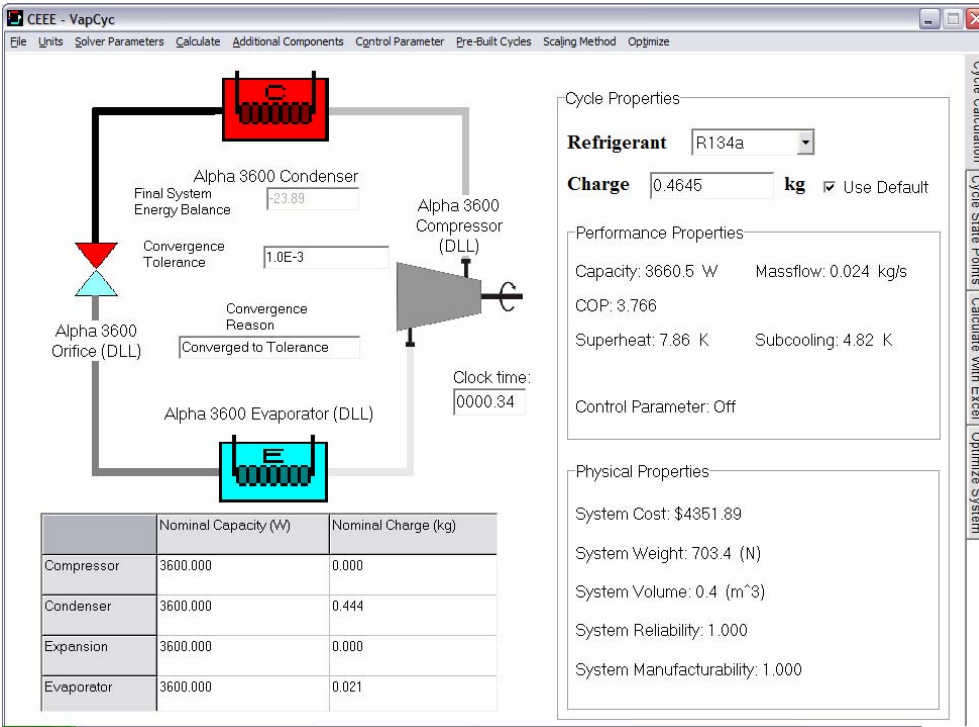


Figure A2- 3: The Solution of the Alpha3600 System at the Default Operating Point

A2.2.2 Editing Component Independent Variables

Component models may have independent variables which can alter their performance. An example of this is the air temperature of a condenser. The components themselves are responsible for providing the means to alter these values, which can be accessed by right clicking the right button, while hovering the mouse over the component of interest and selecting the “edit” function. Figure A2- 4 illustrates the edit window for the Alpha3600 condenser.

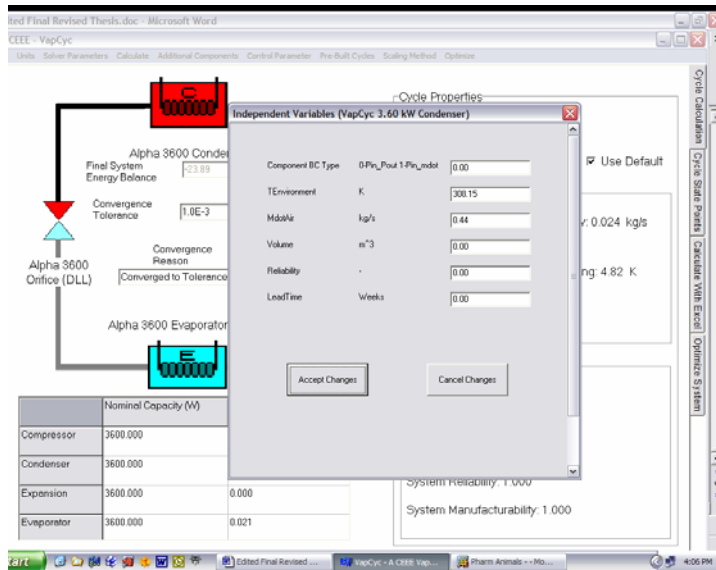


Figure A2- 4: Edit Menu for the Alpha 3600 Condenser

A2.2.3 The Parametric Interface

VapCyc has a parametric interface, which utilizes Microsoft Excel. Through this interface, the system is completely specified. Component models as well as their independent variables are accessed through this interface. Figure A2- 5 is a shot of the Excel interface for VapCyc. From the figure, it is seen that the component models are specified using their name and the system charge is varied for the individual run.

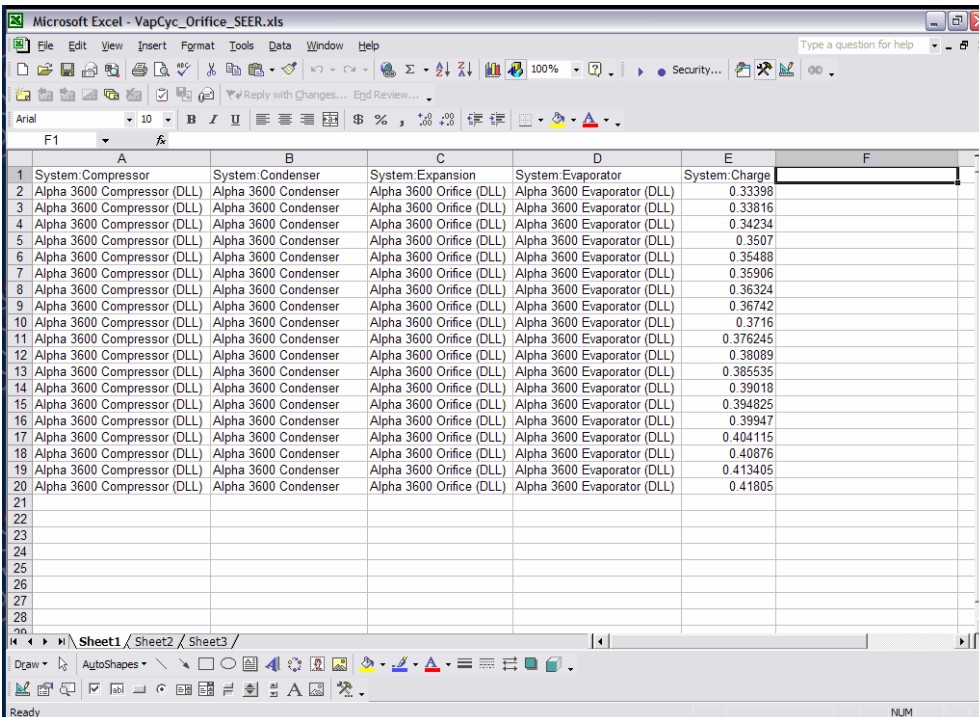


Figure A2- 5: Excel Interface for VapCyc's Parametric Input Mode

A2.3 The Solution Routine

VapCyc is an application program to simulate the performance of a specific vapor compression refrigeration system. As such, it has a Graphical User Interface (GUI) which implements a calculation engine. The calculation engine is a set of routines which receives information from the GUI and give the necessary information to the Junction Solver. After the Junction Solver has resolved the system, the calculation engine then retrieves information from the Junction Solver and feeds it to the GUI. This is achieved through three distinct phases of operation, namely a Pre Processor Phase, Solution Phase and Post Processor Phase.

A2.3.4 The Pre Processor Phase

The Pre Processor Phase is an interactive phase that allows the user to specify the system configuration and interpret information. VapCyc employs the specified system context to evaluate the current set of components and reports the results to the user. The pre processor is responsible for determining the nominal capacity and charge of a component for the specified system context. The capacity is a rough indicator of the flow rate the device will have at the operating component's operating point for the specified system context. If component's nominal capacities differ by a large degree, then the specified system context will be far from the solution point, if a solution for the specified system exists. The sum of the nominal charge contained in all components gives an indication as to the appropriateness of the specified system charge.

A2.3.5 The Solution Phase

The solution phase of the program begins when the user tells VapCyc to begin solving the currently configured cycle. At this point, the VapCyc GUI forms the JCP and provides the connectivity information to the calculation engine. The system context is then used to provide the calculation engine with the high and low pressure and enthalpy as well as the nominal system charge values to scale the Junction Solver independent variables and residual equations.

A2.3.6 The Post Processor Phase

The Post Processor Phase is the final communication between the calculation engine and VapCyc GUI. The calculation engine returns a value to the GUI communicating information about the success of the solver. The results are one of the following:

- The numeric solver in the calculation engine solved resolved to a specified tolerance.
- The numeric solver in the calculation engine solved a spurious convergence, or local minimum, but not to a specified tolerance.
- The numeric solver had solution problems and the returned solution is invalid.

At this point, the GUI queries the calculation engine for solution variables, including: evaporator cooling capacity, condenser heat of rejection, power consumed by each component, system COP, evaporator superheat, condenser subcooling, and compressor mass flow rate. The values of several components variables are available by hovering the mouse over a given component. Figure A2- 6 shows the screen after solution, and illustrates the presentation of compressor variables.

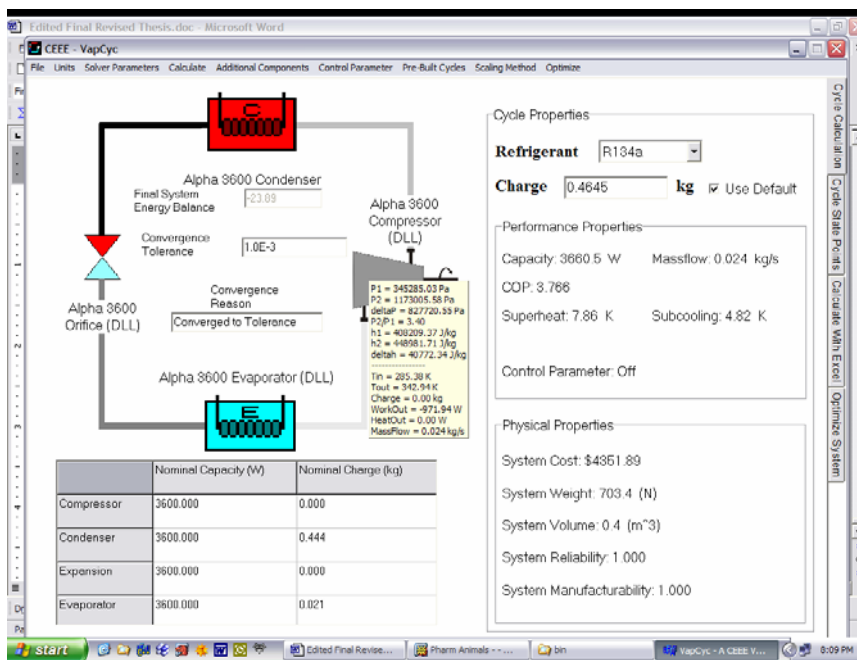


Figure A2- 6: VapCyc Main Screen After Solution, Showing Compressor Variables