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A GENERIC BENCHMARK FOR A MINI-SPLIT HEAT PUMP SYSTEM

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A GENERIC BENCHMARK FOR A MINI-SPLIT HEAT PUMP SYSTEM

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A GENERICAL BENCHMARK FOR MINI-SPLIT HEAT PUMP SYSTEM

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Heating, Ventilation, and Air Conditioning (HVAC) accounts for half of the building energy consumption in the U.S where Mini-Split Heat Pumps (MSHPs) are an emerging type of HVAC system. Their utilization has greatly increased by 34% from 2009 to 2013 and high potential EER is recognized for MSHPs. However, there is limited research involving MSHPs systems, and there is no generic benchmark for system testing and modeling. The available simulation tools such as VapCyc, GreatLab, and CYCLE_D are either too complicated, difficult to access, or not freely available. Therefore, an accurate and public share generic benchmark is essential and will be researched for researchers and scientists.

In this study, the Heat Pump Design Model (HPDM) is utilized to investigate MSHP performance values. There are five different kinds of input parameters necessary for the HPDM, namely a general system description, system refrigerant-side balancing, compressor characteristics which need a compressor scaling method, fin-and-tube heat exchanger parameters, and system operating conditions. Based on systematic inputs of the HPDM, several key outputs can be obtained, including system capacity, power consumption, and mass flow rate. By comparing output values with existing data sets, the capability of a generic model for MSHP can be identified.

In order to validate the methodology analyzed above, two kinds of case studies will be presented. In the first study, a comparison of lab data and simulation results is presented, whereas in the second one, a comparison is conducted between manufacturing data and simulation results. By identifying all of the input parameters for the specified unit, which is the LG LA096HV in this study, the HPDM can obtain simulation results immediately. As indicated by simulation results, the HPDM can be a generic benchmark in a certain temperature range with a relative error below 5%.

ACKNOWLEDGEMENT

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NOMENCLATURE

T_{odb}	outdoor dry bulb temperature
T_{idb}	indoor dry bulb temperature
T_{owb}	outdoor wet bulb temperature
T_{iwb}	indoor wet bulb temperature
P	finned face area
a	longitudinal center-to-center distance between tubes
b	transverse center-to-center distance between tubes
h	height for the heat exchanger
d	depth for this heat exchanger
#r	number of tubes in each row
#n	number of row

CHAPTER 1. INTRODUCTION

1.1. Background

With the development of economy and society, energy consumption is increasing for both developed and developing countries. During the years between 1984 and 2004, the consumption of primary energy sources which include non-renewable energy and several renewable energy sources from nature (WIKIPEDIA, 2016) increased by 43% (Pérez-Lombard, Ortiz, & Pout, 2008). In addition, the trend of increasing energy consumption will continue. The rapid speed of economic increase for developing countries is higher than for developed countries, with 3.2% annual rate for emerging countries and 1.1% annual rate for developed countries, separately (Pérez-Lombard et al., 2008). Oil, coal, and natural gas combustion are in the primary consuming position (IEA, 2014) in the year 2012, shown in Figure 1-1. As shown in Figure 1-2, the primary energy sources are mainly exploited in the developed countries, but those energy sources are also applied greatly in the developing nations, especially in China with 21.8% of the total energy consumption.

According to the Energy Information Administration (EIA), the energy consumption of emerging economies will surpass developed nations in 2020, shown in Figure 1-3. Typically, energy consumption can be divided into three sectors: industry, transport and other uses. Building energy consumption accounts for 40% for the total energy consumption in the U.S. (Cho, Li, Park, & Zheng, 2015) and amounts to about 39 quadrillions Btu, a large amount of total energy consumption (EIA, 2016).

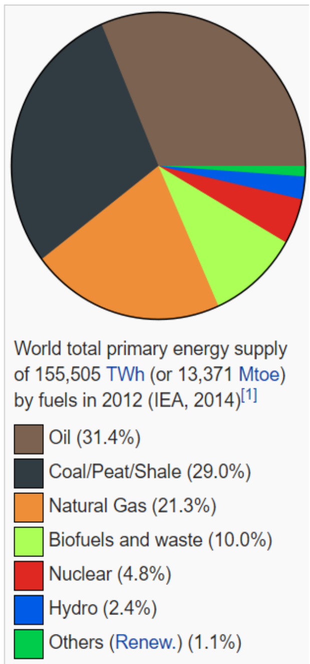

Figure 1-1 Primary energy supply by fuels

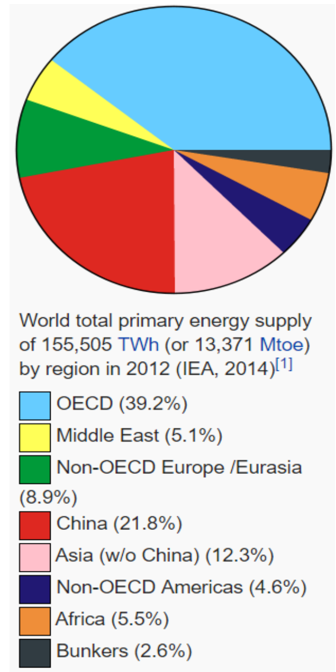

Figure 1-2 Primary energy supply by regions

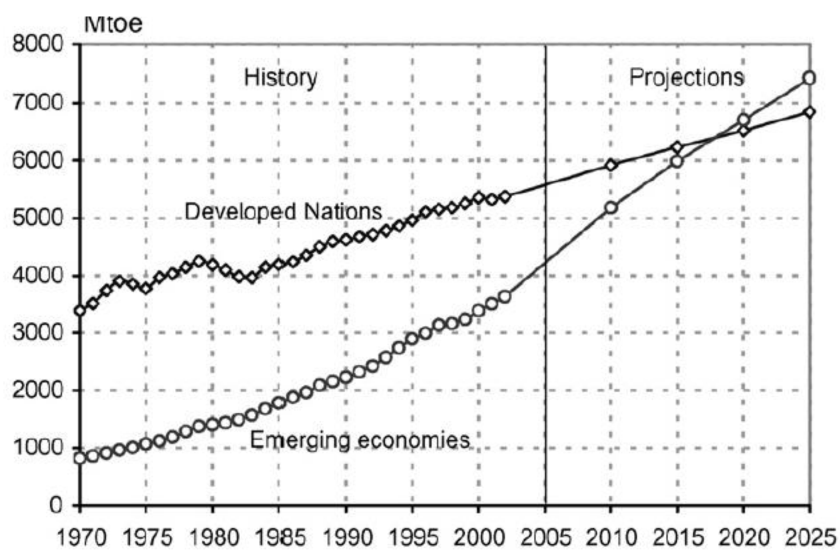

Figure 1-3 Developed and developing countries' energy consumption Source: Energy Information Administration (EIA)

Heating, ventilation, and air conditioning (HVAC) systems play a significant role in commercial and residential building energy consumption. According to data provided by the U.S. EIA and European Commission in 2014, half of building energy consumption is due to HVAC system operation (Knight, 2012) (U.S. EIA, 2014). Therefore, there is an extremely necessary and valuable importance to investigate HVAC consumption classification. Residential energy consumption (45% of the total) and commercial energy consumption (55% of the total) are included in building energy consumption (U.S. DOE, 2011). The amount of cooling and ventilation energy usage is 16% for commercial buildings while heating energy is 26%. In terms of residential energy consumption, 46% is for heating, which is much higher than commercial building energy usage. Meanwhile, 9% is for cooling energy consumption, which is lower than commercial buildings (Mitchell & Braun, 2011).

Typically, there are ducted systems and ductless systems utilized in buildings. Ducted systems are most traditional systems in the United States, while ductless systems are rather new in the U.S., though they have a large market share in Asian countries (Carmichael, Bielecki, Meyer, & Salvador, 2015). Rooftop packaged units (RTUs) are utilized in commercial buildings in the U.S., while Mini-Split systems are widely used in residences, particularly in Asian countries. Moreover, the residential HVAC market is increasing in the U.S., while the commercial HVAC market is expected to steadily grow in the future (News, 2016). The Mini-Split Heat Pump (MSHP) system is a very high-efficiency HVAC system and therefore its market share has a promising future, with only 5% of the total cooling and heating system market in the U.S. (Green Building Advisor, 2015). Additionally, the overall MSHP market increased between 2009 and 2013 by

34%(Navigant Consulting, 2014), which means that urgent research is required in the residential HVAC market.

A trustworthy simulation model or benchmark can be incredibly prominent for operation. An outstanding model could provide several advantages as follows:

- Large savings on experimentation costs
- Increased efficiency for testing unit performance
- Undisturbed experimental conditions
- Increased safety for difficult experiments (Ron, 2016).

In order to design higher efficiency residential HVAC systems, the simulation software used for analysis and application should be based on an accurate model or benchmark. There are many RTUs developed for researching performance in commercial buildings. Also, there are many that investigate system simulation programs. For example, Trane company investigates the TRACE 700 which is helpful for simulating building energy consumption. EnergyPlus, funded by the U.S. Department of Energy Building Technologies Office, is a whole-building energy simulation program for modeling energy consumption applied in buildings due to cooling and heating depletion, lighting, ventilation, and other loads. (U.S. DOE, 2015) Both types of software can conduct reliable simulation processes.

For residential buildings, especially for mini-split systems, there are limited researches about simulating the entire vapor compression cycle. Few software programs can be achieved for this purpose. For instance, EES can simulate a refrigeration cycle with its programming but it is time-consuming. Some well-known companies, like

Carrier and Danfoss, create databases for their customers. They can simulate refrigeration cycles only within their own units or equipment.

It is clear that a generic benchmark for the mini-split system would be proposed of great value. The Heat Pump Design Model (HPDM) developed by Oak Ridge National Lab for the U.S. Department of Energy (DOE) has been widely used in the residential building market, but only for split systems. Since the HPDM works well in split systems, there is a possibility that it can simulate mini-split systems well also.

1.2. Literature Review

There are limited models and simulation software packages devoted to simulating a mini-split heat pump refrigeration cycle. Some simulation programs focus on calculating system loads and analyzing power consumption, for example, Trace 700 and EnergyPlus. There are also some simulation models made by companies, for instance, Simtools, made by Carrier, and T-Rex, created by Trane. But because of commercial confidentiality agreements, those companies communicate with others rarely. (Chunlu, 2012) Some programs, like Dymola, ASPEN, AMESim, and SimulationX, can make a simple refrigeration cycle simulation, but they can only simulate uncomplicated refrigeration components. For complex vapor compression cycles or modeling, the speed of simulation processing will be slow and the simulation results will not be robust enough. Therefore, these programs are not good at simulating complicated refrigeration cycles.

Richardson, et. al (2006) developed a component-based platform for simulations of steady-state cycles at the Center for Environmental Energy Engineering (CEEE) at the

University of Maryland. Named VapCyc (Andresen, 2009), this software can achieve more complex refrigeration cycle simulations and utilizes geometry input for processing. It also calculates the charge inventory of the simulation cycle. Another combined software also provided by the CEEE is Coil Designer, which is a sophisticated tool for design and optimization of air-cooled heat exchangers (CEEE, 2016). It is a professional and highly customizable software for designing heat exchangers using the tube-in-tube method. When users finish their heat exchanger design, the VapCyc can be inputted within geometry files engendered by Coil Designer. The VapCyc requires users to choose individual component models (compressor, condenser, evaporator and expansion device models) and a value for system refrigerant charge. Once those parameters are fixed, the VapCyc executes vapor compression cycle simulation (Richardson & Jiang, 2002). The main projects the the VapCyc model are rooftop units' projects, especially for the supermarket.

The National Institute of Standards and Technology (NIST) developed a basic simulation software named CYCLE_D which focuses on vapor compression refrigeration cycles using pure refrigerants or blends of refrigerants (Brown, Domanski, & Lemmon, 2009). It has been investigated for a long time and now there is a version 5.1 presented for users. Not only can CYCLE_D simulate simple vapor compression cycles which include one compressor, a condenser, an evaporator and an expansion device, but it can also simulate subcritical cycles which may contain a second compressor, economizers or an intercooler. Through great work done by NIST, the CYCLE_D has become an easy and convenient program aiming for basic refrigeration cycle simulations, two-stage economizer cycle, and three-stage economizer cycle simulations. Additionally, the program can generate simulation results on P-h state diagram and T-s state diagram. One more advantage is that the compressor model can be represented as a 10-coefficient formula based on ARI Standard 540-2004. However, users may not select their individual heat exchanger geometry with this software and they need to figure out the condenser and evaporator saturation temperature in this model.

Chunlu Zhang and his research group introduced a newly developed general simulation platform, GREATLAB, in the College of Mechanical Engineering at Tongji University in Shanghai, China (Chunlu, 2012b). Several component models are included in this platform. The bifurcation diagram is illustrated in Figure 1.2. The CoilLab concentrates on coil design and optimization. The CompLab focuses on compressor modeling construction within AHRI 10-coefficient compressor maps, while the FanLab designs for fan design. GREATLAB combines all the sub-software into its platform to achieve the vapor compression cycle simulation. Users can define their own vapor compressor cycle in GREATLAB pro, which is a more expansive version than the standard one. The major projects for the GreatLAB are RTUs, train air-conditioners, and electronic cooling.

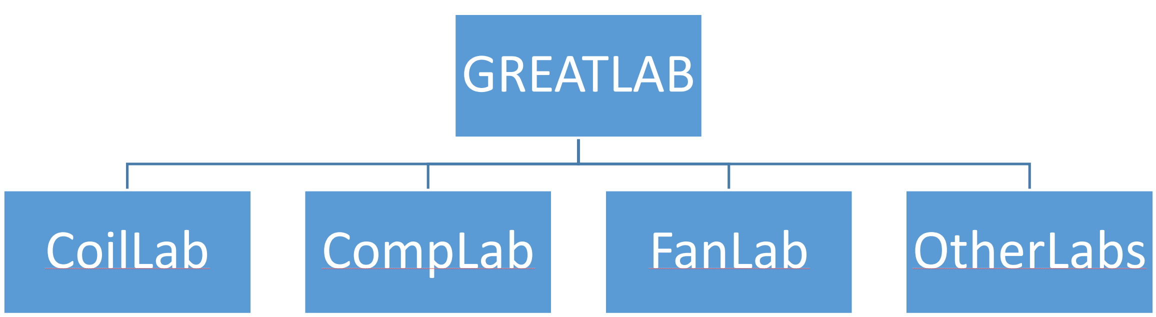

Figure 1-4 GreatLab and its components

FrigoSim is a vapor compression cycle software particularly aimed at refrigeration plants and heat pump systems. Semi-hardware based models of heat exchangers are included in the FrigoSim software (Andresen, 2009).

Sarkar et al. (2006) represented a simulation tool focusing on refrigeration cycle, which is a hardware-based one. This program package can optimize the cycle for maximum COP and equation sets that can be solved by the Newton-Paphson method (Sarkar, Bhattacharyya, & Gopal, 2006).

The Heat Pump Design Model (HPDM) is a steady-state design analysis research tool for heat pumps and air conditioning systems (Rice, 2015). Hiller and Glicksman, at the Massachusetts Institute of Technology (MIT), developed the original basic models for compressors and heat exchangers in 1976. In 1978, the first HPDM version was created by Ellison and Creswick using FORTRAN programs. The Mark I version was generated by Fischer and Rice in 1983, and the first PC version was released as Mark III in 1985. Followed by the version Mark IV, the variable-speed model, was achieved with design parametric capabilities and added electronically commutated motors (ECMs) (Heat et al., 1997). The latest Heat Pump Design Model version is Mark VII, which was upgraded in 2005 and 2006 within the ASHRAE Technical Research Project (TRP)-1173 (B. Shen, July 2006). The major projects that the HPDM operating are RTUs. It requires several heat exchanger geometries, compressor map input parameters, system operation conditions, and other values. Users can define their own heat exchangers and reasonable compressor map representations. Simulated by the HPDM, the system capacity, power consumption, mass flow rate and other key parameters will be displayed. Input and output will be more fully discussed in Chapter 3 of this thesis.

1.3. Motivation and Objectives

There is limited research on mini-split heat pump systems in the U.S., which are more and more significant on residential buildings. In addition, the models that people can find are mainly for RTUs. HPDM operated well in split systems within large DOE projects and provided accurate simulation results. Moreover, rooftop units have similar refrigeration components to mini-split heat pump systems. Therefore, an investigation of a generic benchmark for mini-split heat pump system is of interest and needs to be researched.

The objectives of this thesis are:

- Describe the methodology for utilizing the Heat Pump Design Model.
- Determine system inputs and outputs.
- Determine whether the compressor map scaling method in rotary compressor maps and scroll compressor maps can be utilized in a mini-split heat pump system.
- Compare and analyze lab and manufactural simulation data results obtained be the HPDM with lab and manufactural performance data for cooling and heating modes.
- Generate a generic benchmark for a mini-split heat system within a certain temperature range.

1.4. Methodology

The Heat Pump Design Model utilizes a physical model to achieve the stated objectives. With five different kinds of inputs, the HPDM can generate detailed simulated data which will be considered to compare with laboratory and manufactural performance data in both cooling and heating modes. The relative errors for several key outputs, like system capacity, mass flow rate, power consumption and other parameters, will be specified in order to investigate whether the HPDM can be a generic benchmark for a mini-split heat pump system.

1.5. Thesis Organization

The first chapter introduced the background for a mini-split heat pump system and the importance to generate a generic benchmark for that. Literature reviews about related models and their functions are provided. In addition, the research motivations and objectives are touched on and system methodology is discussed in the first chapter.

The second chapter focuses on the data sources. Laboratory data and manufactural data sets are offered to validate the research method. The third chapter illustrates the methodology for utilizing the HPDM, illustrates functions for five kinds of inputs and several key outputs after simulation. Two case studies are discussed in the fourth chapter. They are lab and manufactural simulation results compared to lab and manufactural performance data, separately. The final chapter summarizes several main conclusions for this thesis and suggestions for future studies.

CHAPTER 2. DATA SOURCES

2.1. Overview

There are two types of data sources used for simulations. The first one is the laboratory data and the second one is the manufactural data both for cooling mode and heating mode. The laboratory data (Cheung & Braun, 2014) was tested by Dr. Howard Cheung and Dr. David Yuill in the Herrick Laboratory at Purdue University. The manufactural data is provided by LG air conditioner engineer product data book. The testing unit is an LG unit, termed as LA096HV, which is a mini-split heat pump system with rated cooling capacity being 9000 Btu/hr and rated heating capacity being 11,700 Btu/hr.

2.2. Laboratory Data

The lab data can be divided into two sections which are cooling test performance sets and heating test performance sets, separately. The lab data sets include several performance data under the circumstance of the maximum compressor speed. Furthermore, the lab data sets were obtained at combinations of ambient temperature (T_{odb}) of 67, 87, 95, 105 and 115 F, indoor dry-bulb temperature (T_{idb}) of 74 and 80 F and, indoor wet-bulb temperature (T_{iwb}) of 56, 62, 66 and 67 F in the cooling mode. Simultaneously, the lab data sets were also gathered as the combinations of indoor dry-bulb temperature of 64 and 70 F, outdoor dry-bulb temperature of 7,17, 27, 35, 42, 47, 62 and 68 F, outdoor wet-bulb temperature of 6, 15, 23, 30, 35, 37, 40, 48, 51 and 53 F in the heating mode. Using the psychrometric chamber in the Herrick Laboratory at Purdue University, lab performance data could be obtained. The lab performance data include indoor coil refrigerant side cooling capacity, system refrigerant charge, the coefficient of performance (COP), energy efficiency ratio (EER), sensible heat ratio (SHR), refrigerant mass flow rate, power consumption and other parameters for both cooling tests and heating tests.

2.3. Manufacturing Data

There are also two components of manufactural data: cooling performance data sets and heating performance data sets. The manufactural data is also tested within the maximum compressor speed. The differences between lab data and manufactural data are the ambient temperature range and the indoor dry-bulb/wet-bulb temperature range. The range of manufactural data for T_{odb} , 68 F to 125 F, is larger than that of the laboratory data and T_{idb} is either 68, 71.6, 77, 80.6, 86 or 89.6 F. The range of laboratory data for T_{iwb} is smaller than the manufactural data, identified in Table 2-1 to be 57.2 F to 75.2 F. In general, manufacturers only provide total capacity, sensible capacity and power consumption for mini-split heat pump systems. However, SHR can be calculated by equation (2.1) and power consumption can be computed by equation (2.2). Therefore, the performance data can be more deeply investigated as total capacity, SHR, and power consumption.

$$\mathrm{SHR} = \frac{Sensible\ Capacity}{Total\ Capacity} \tag{2.1}$$

$$\text{Power consumption} =$$

$$compressor\ power\ consumption + indoor\ fan\ motor\ power\ consumption + outdoor\ fan\ motor\ power\ consumption \quad (2.2)$$

To be specific, Table 2-1 shows all performance parameters tested by the manufacturer with different T_{odb}, T_{idb}, and T_{iwb} in the cooling test mode.

Table 2-1 Manufactural testing data for LG 096HV in the cooling mode

Indoor Air temperature		Outdoor Air Temperature: DB (F)								
		68			77			89.6		
WB (F)	DB (F)	TC (Btu/hr)	SHR	PI (W)	TC (Btu/hr)	SHR	PI (W)	TC (Btu/hr)	SHR	PI (W)
57.2	68	8837	0.85	390	8462	0.90	410	7916	0.98	550
60.8	71.6	9383	0.79	530	8974	0.83	540	8462	0.90	650
64.4	77	9929	0.73	570	9519	0.77	580	9008	0.83	680
66.2	80.6	10202	0.71	580	9792	0.75	590	9281	0.80	690
71.6	86	11021	0.62	580	10611	0.65	600	10099	0.70	700
75.2	89.6	11567	0.57	580	11157	0.60	600	10611	0.64	720

Indoor Air temperature		Outdoor Air Temperature: DB (F)					
		95			104		
WB(F)	DB(F)	TC(Btu/hr)	SHR	PI(W)	TC(Btu/hr)	SHR	PI(W)
57.2	68	7677	1.03	610	7370	1.06	670
60.8	71.6	8223	0.94	700	7882	0.98	730
64.4	77	8769	0.87	720	8428	0.89	740
66.2	80.6	9008	0.84	710	8701	0.87	740
71.6	86	9827	0.73	740	9485	0.75	750
75.2	89.6	10372	0.67	760	10031	0.69	770

Indoor Air temperature		Outdoor Air Temperature: DB (F)					
		109.4			114.8		
WB(F)	DB(F)	TC(Btu/hr)	SHR	PI(W)	TC(Btu/hr)	SHR	PI(W)
57.2	68	7165	1.11	660	6960	1.15	590
60.8	71.6	7677	1.02	690	7472	1.05	600
64.4	77	8223	0.93	690	8018	0.96	590
66.2	80.6	8496	0.90	690	8291	0.93	580
71.6	86	9281	0.78	700	9076	0.80	580
75.2	89.6	9827	0.72	710	9622	0.73	590

For the manufactural performance parameters presented in Table 2-1, TC stands for the total capacity in a certain outdoor air temperature and indoor dry-bulb/wet-bulb temperature, while PI is the abbreviation for the total power input which means the same as total power consumption. The yellow highlighted cells are varied T_{odb} and The blue highlighted columns represent T_{idb} and T_{iwb}. The rated condition whose total cooling capacity is 9008 Btu/hr is shown in red font in Table 2-1.

Simultaneously, there are also detailed manufactural testing results for the heating mode within varied outdoor wet-bulb temperature (T_{owb}) and indoor dry-bulb temperature. Table 2-2 shows the results of the heating mode for this unit.

Table 2-2 Manufactural testing data for LG 096HV in the heating model

Indoor Air	Outdoor Air Temperature: WB (F)							
Temperature	5		14		23		32	
DB(F)	TC(Btu/hr)	PI(W)	TC(Btu/hr)	PI(W)	TC(Btu/hr)	PI(W)	TC(Btu/hr)	PI(W)
60.8	8803	750	9247	730	10031	770	10714	820
64.4	8701	760	9247	750	10031	790	10714	850
68	8665	770	9247	770	10065	810	10680	870
69.8	8632	780	9247	780	10065	820	10645	880
71.6	8632	790	9247	790	10031	830	10645	890
75.2	8396	810	9144	810	9929	860	10543	910

Indoor Air	Outdoor Air Temperature: WB (F)					
Temperature	42.8		50		59	
DB(F)	TC(Btu/hr)	PI(W)	TC(Btu/hr)	PI(W)	TC(Btu/hr)	PI(W)
60.8	11908	880	12556	920	13648	980
64.4	11806	900	12420	930	13614	990
68	11703	920	12317	950	13614	990
69.8	11635	930	12317	950	13546	990
71.6	11533	940	12317	960	13409	990
75.2	11464	950	12113	970	13273	1000

The manufactural parameters for heating mode are listed in Table 2-2. The yellow highlighted cells are varied T_{owb} from 5 F to 59 F. In addition, the blue highlighted columns representation of T_{idb}. The rated situation whose total heating capacity is 11703 Btu/hr is also marked in red font in this table.

CHAPTER 3. DOE/ORNL HEAT PUMP DESIGN MODEL

The DOE/ORNL Heat Pump Design Model is a very useful research tool for simulating refrigeration cycles for heat pump systems and air conditioners system. The software was developed by Oak Ridge National Laboratory for the U.S. Department of Energy. It is a no charge program for anyone to do simulations for vapor compression cycles. In section 3.1, the definition, strengths, the Heat Pump Design Model will be discussed. Moreover, the basic methodology of Heat Pump Design Models is desired to be presented in section 3.2.

3.1. Introduction to the Heat Pump Design Model

In this section, Heat Pump Design Model will be defined and a large number of advantages and a few of disadvantages will also be provided. The Heat Pump Design Model (HPDM) is a web-based research software platform that analyzes a steady-state design of air-to-air heat pumps and air conditioners, whose interface is presented in Figure 3-1 (ORNL, 2015). Additionally, the HPDM can be utilized this software online freely. This program comprises of several strengths which will discuss more in the next paragraph.

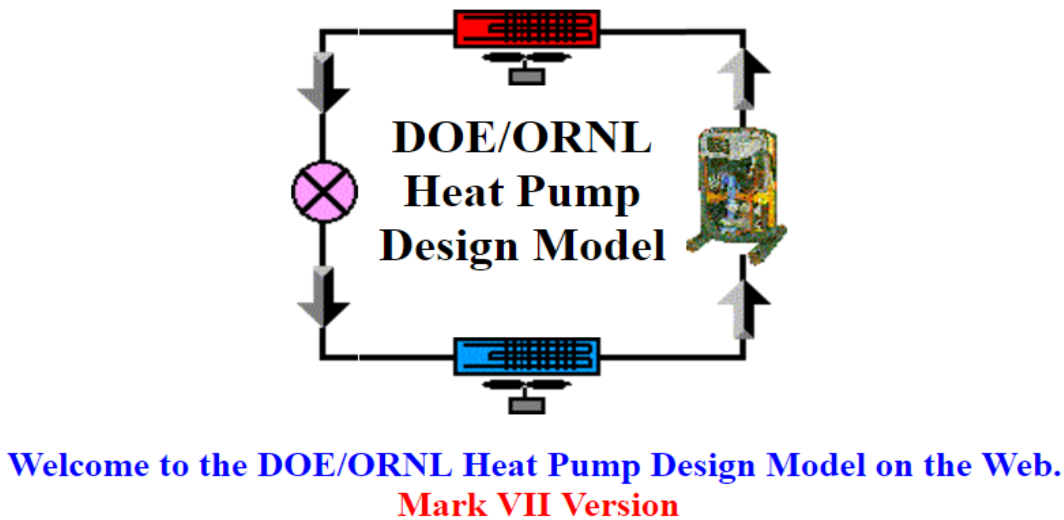

Figure 3-1 The interface for the DOE/ORNL Heat Pump Design Model

In the real world, the HPDM is a useful and effective program that has already been employed in several great essential projects. One example is Advanced variable speed air-source integrated heat pump (AS-IHP)(Baxter, 2014). This project is funded by U.S. Department of Energy with a total budget $2,120,000. Researchers utilized the Heat Pump Design Model to develop the prototype design and lab prototype test system proposal. Calibrated by HPDM, researchers developed test results. Another application of this software is for Cold Climate Heat Pump (CCHP) research projects, also funded by the Department of Energy. Oak Ridge National Laboratory (ORNL) and Emerson Climate Technologies worked together for a Cooperative Research and Development Agreement (CRADA) to investigate a Cold Climate Heat Pump for the residential market in the U.S between 2011 and 2015 (Bouza, 2016). Creating the urgency to develop more about this, there are lots of states in cold or very cold zones, shown in Figure 3-2. This figure is obtained from “High-Performance Home Technologies Guide to Determining Climate Regions by County” at Pacific Northwest National Laboratory (PNNL) and ORNL in August 2010.

In the CCHP projects, researchers achieve building energy models by utilizing the HPDM. In theory, the Heat Pump Design Model is a physical model, which means that models are provided by physical or engineering principles and the most accurate estimators of output can be obtained when models are operated accurately (Katipamula & Brambley, 2005). Also, the HPDM has already achieved several rooftop units (RTU)’s projects for the U.S. DOE. Therefore, the Heat Pump Design Model is an accurate and reliable software both in reality and theory for RTUs.

Figure 1. Seven of the eight US climate zones recognized by Building America occur in the continental United States. The sub-arctic U.S. climate zone, not shown on the map, appears only in Alaska.

Figure 3-2 Climate distributed zones in the U.S.

3.2. Methodology of Heat Pump Design Model

In order to utilize the HPDM proficiently, researchers are required to identify exactly the real methodology for this software application for every input parameter. Additionally, the Heat Pump Design Model is a physical model so operators need to specify a large number of details which will be discussed in this section. More specific parameters which will have to be input will be exposed and analyzed. The Heat Pump Design Model inputs should be identified as 5 different parameters.

- General System Descriptions
- System Refrigerant-Side Balancing
- Compressor Characteristics
- Fin-and-Tube Heat Exchanger Parameters and Configurations
- System Operating Conditions

Since the input parameters for the HPDM are not simple values, their meanings and applications are described in particular in the following five sections.

3.2.1 General System Descriptions

Users need to specify whether they are using air conditioners or heat pump systems. In addition, the refrigerant for the system is required to be confirmed by operators. Typically, for the mini split heat pump system, manufacturers would prefer R22 as the system refrigerant in the past, but now they prefer to apply R410A for system refrigerant, since R410a is more environmentally friendly, not contributing to ozone depletion, and absorbing and releasing more heat than R22 (Thien, 2012). Figure 3-3 shows the interface of general system descriptions when users are specifying the system and the refrigerant. Users select cooling mode or heating mode and the refrigerant they are utilizing.

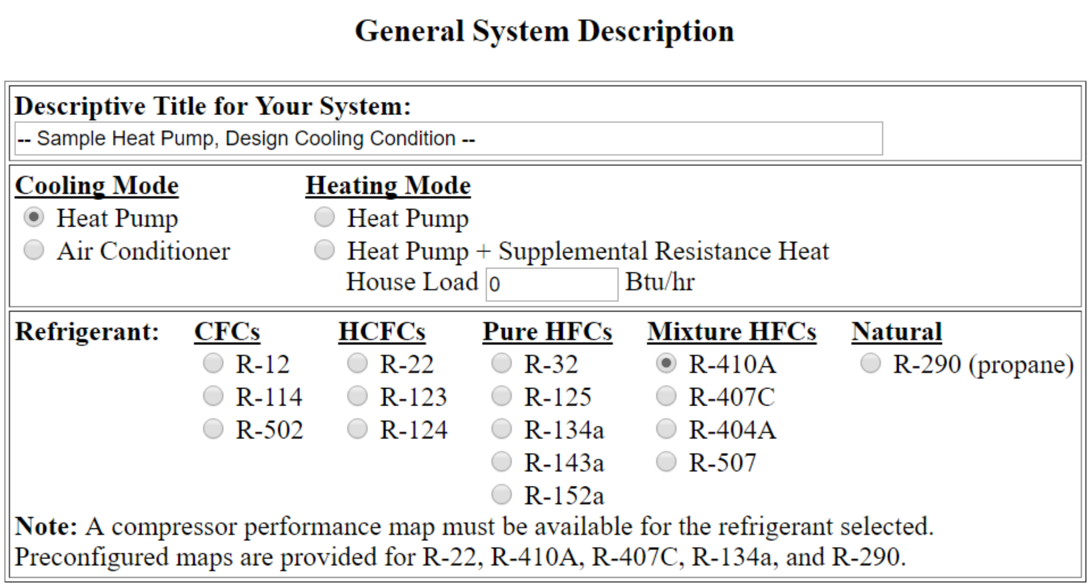

Figure 3-3 The interface of General System Description

3.2.2 System Refrigerant-Side Balancing

The next item users need to indicate is the system refrigerant-side balancing. There are three important parameters that users need to recognize: system refrigerant charge, superheat temperature and subcooling temperature. If a user specifies any two of these three parameters and estimates the third, they can achieve the system refrigerant-side balancing target. As shown in Figure 3-4, there are three combination arrangements for these three parameters. SH means the compressor inlet superheat temperature, while SC means the condenser exit subcooling temperature or flow control devices. If the user has the ability to identify the flow control equipment details, like capillary tubes, short-tube orifices or thermostatic expansion valves, the SC input can be satisfied. Otherwise, users need to specify the condenser exit subcooling temperature. Mass is the abbreviation of the system refrigerant charge. Therefore, as long as individuals specify any two of these three parameters and guess estimate the third, the system will make the iteration computations.

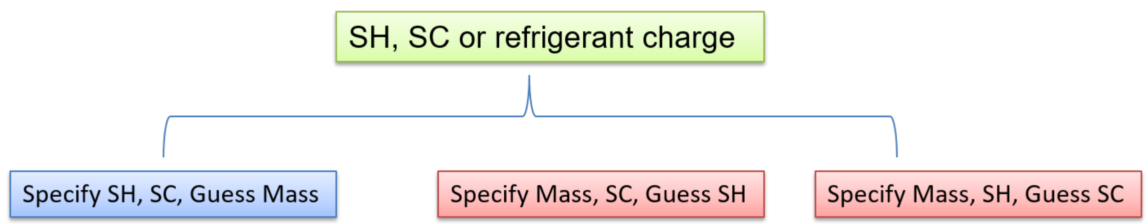

Figure 3-4 Three different ways to achieve system refrigerant-side balancing

For the purpose of investigating how the system operates, the three methodologies will be explored using three logic diagrams in the next succeeding pages. With great contributions to the Heat Pump Design Model by the research group, these three logical schemes were obtained and are represented here.

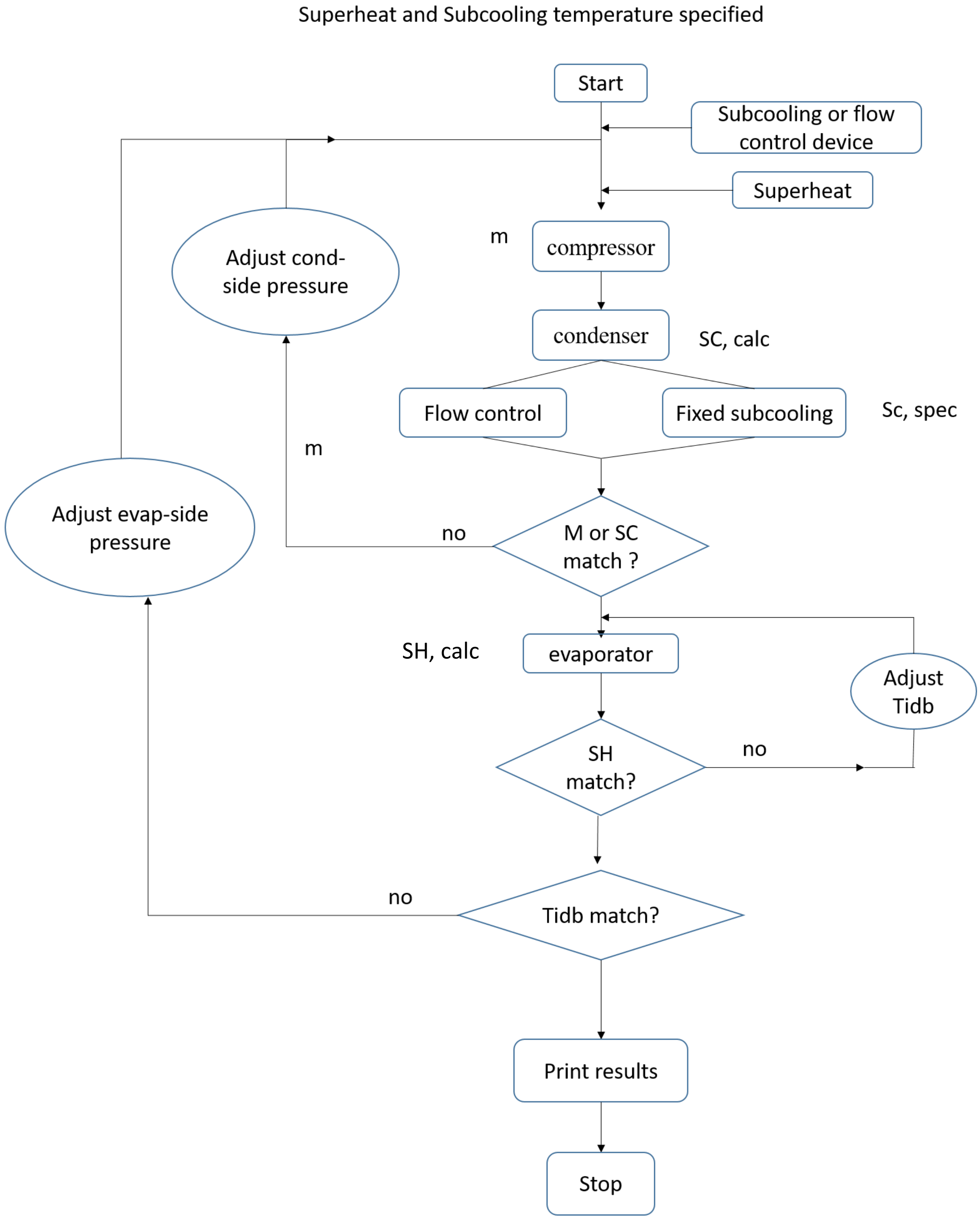

Figure 3-5 the logic diagram for specifying SH and SC

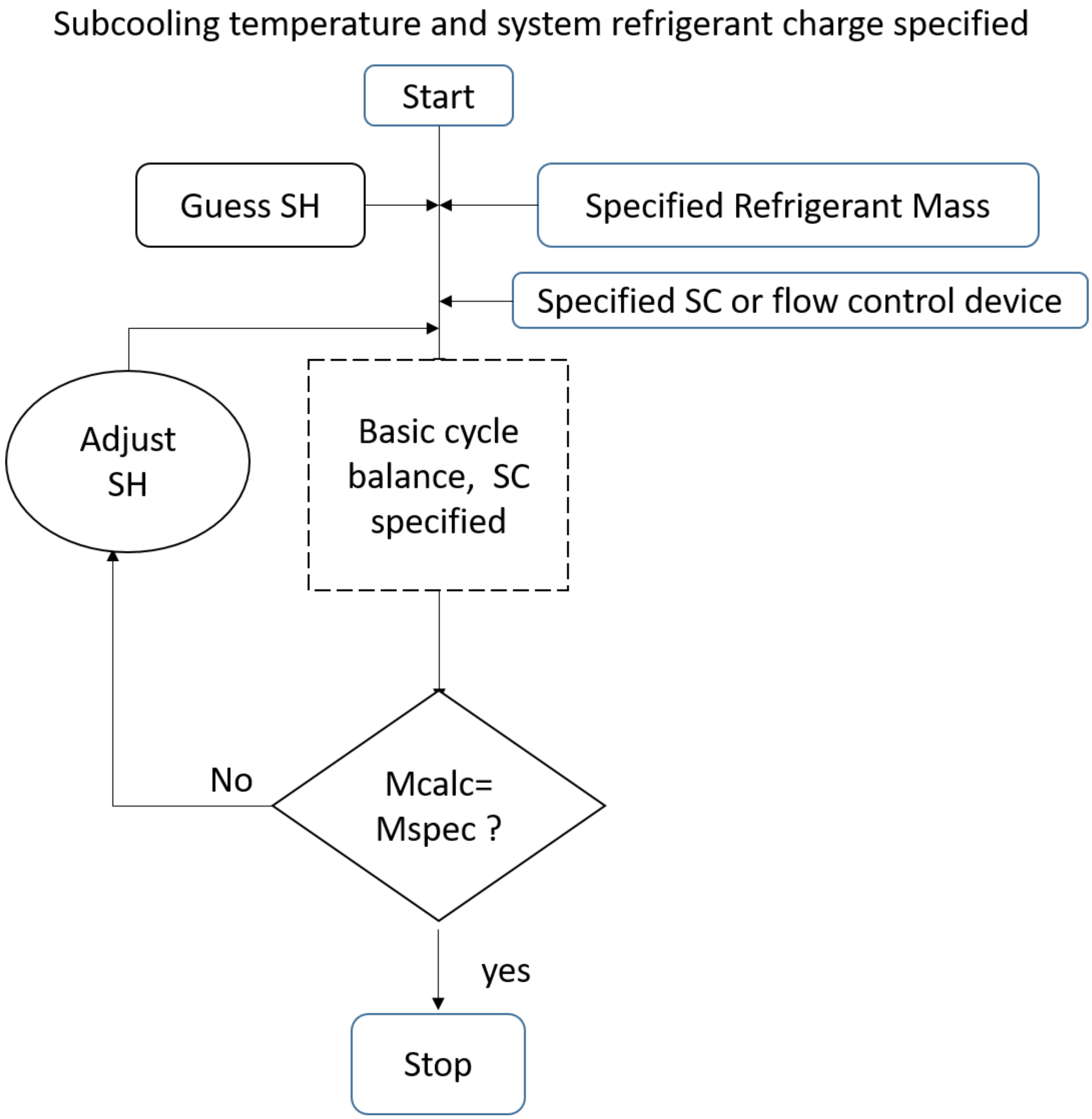

Figure 3-6 the logic diagram for specifying SC and system refrigerant charge

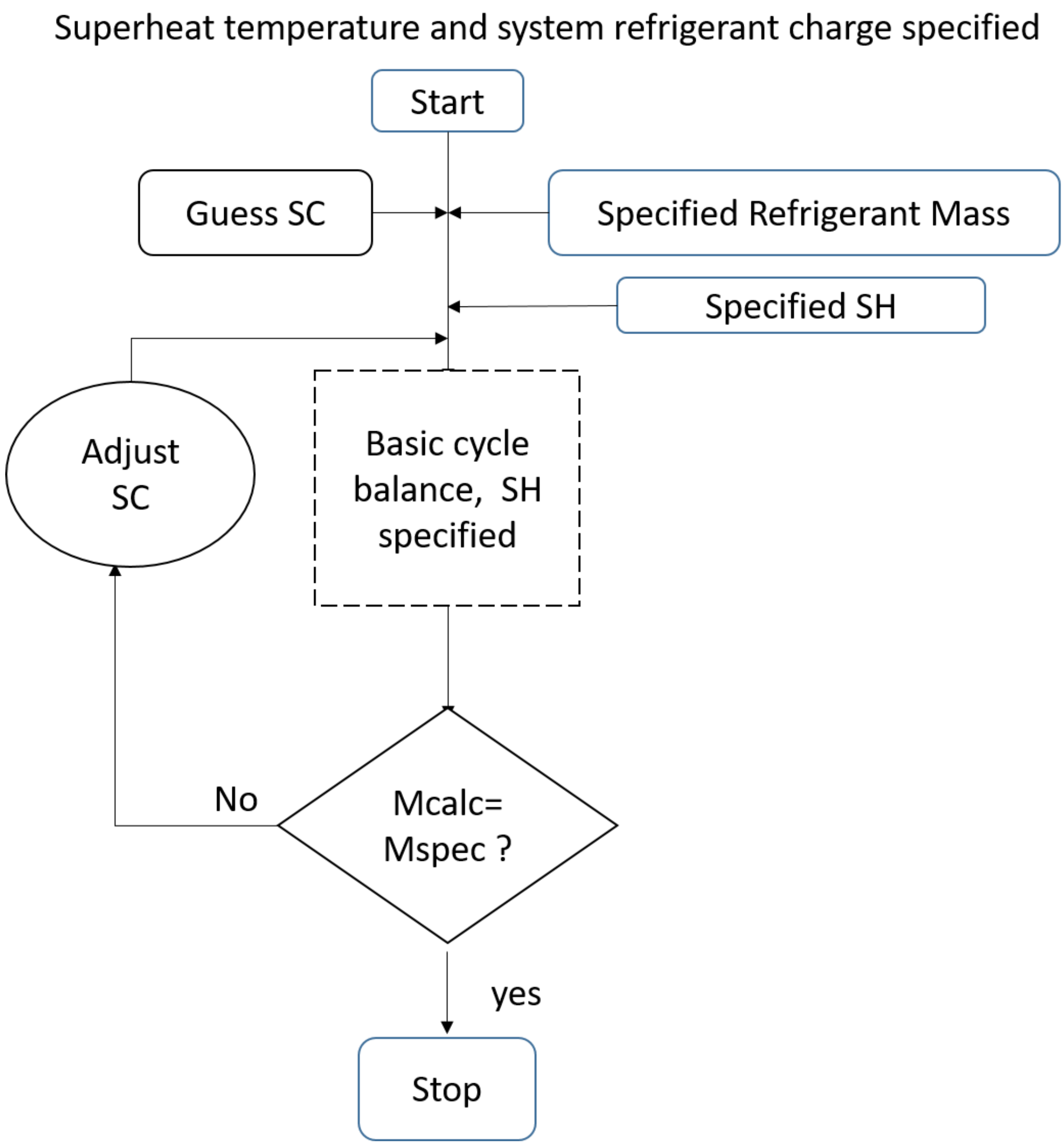

Figure 3-7 the logic diagram for specifying SH and system refrigerant charge

The Figures 3-5, 3-6, and 3-7 are different logic diagrams in three various specified conditions. Shown in Figure 3-5, if users have already discerned the superheat temperature and subcooling temperature for the operation system, they input these two parameters into the Heat Pump Design Model. After several steps' computations, the HPDM will calculate the subcooling temperature by itself. Comparing calculated subcooling temperature and specified subcooling temperature, if the absolute difference value of these two parameters is smaller than the setting value, whose default value is 0.2 F, the HPDM will continue computations to evaporator superheat temperature calculations. Otherwise, the HPDM will need to change the condenser side pressure to satisfy the requirement of the previous conditional statement. As the similar condition, if the absolute difference value between calculated superheat temperature and the specified superheat temperature is less than the setting value, whose default value is 0.5 F, the HPDM will compare the calculated indoor dry-bulb temperature with the T_{idb}, which is specified in the operation condition. Moreover, the default absolute difference of indoor dry-bulb temperature is 0.1 F, which means that the calculation result should be less than the setting value. If the superheat temperature calculation result does not satisfy the setting condition, the Heat Pump Design Model will change T_{idb} to allow system convergence. If the T_{idb} does not match with the setting value, the HPDM will change the evaporator side pressure in order to get a good result. The first logic diagram will print the results by filling the content with all of the conditional statements mentioned above.

The second and the third conditions operate on the same principle but change with specified superheat temperature or subcooling temperature. For example, for the second condition, Figure 3-6, users need to identify subcooling temperature or flow control device and the amount of refrigerant mass. In addition, it is necessary to estimate superheat temperature in order to achieve the operation objective. After a basic cycle balance calculation, the HPDM will calculate the system refrigerant charge. If the calculated one is close enough to the specified one, the system loop will be terminated. Otherwise, the Heat Pump Design Model will need to change the estimated superheat temperature to satisfy the conditional statement. The third situation shares the same principle with the second condition, which is explained in Figure 3-7.

3.2.3 Compressor characteristics

The compressor is the most important part of the whole refrigeration system and the importance could be compared to the heart of a man. Therefore, there is an indispensable need for researchers to investigate more about compressor characteristics. In this section, there are three pieces of information that need to be known: compressor selection, compressor data, and compressor calibration, as shown in Figure 3-8.

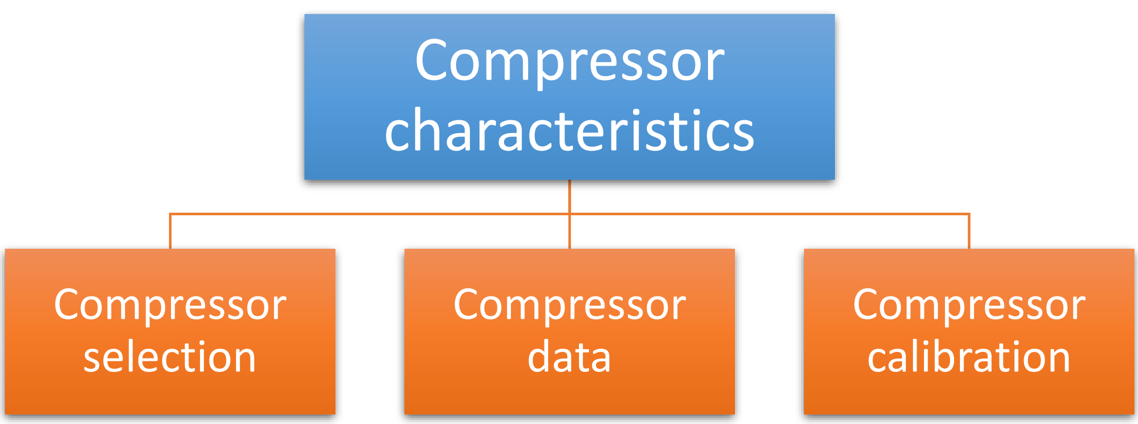

Figure 3-8 The components of compressor characteristics

3.2.3.1 Compressor selections

Users can select preconfigured compressors or input their own compressor characteristics. Figure 3-9 shows the interface for compressor selection that people could specify their own compressors or just select default ones.

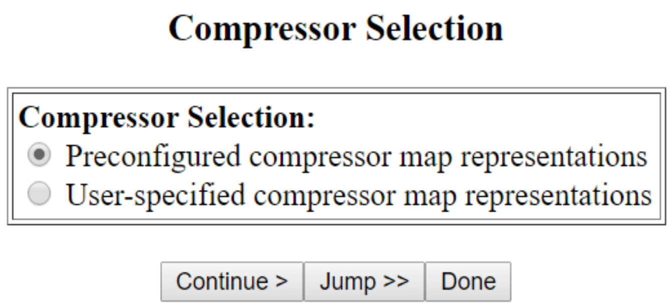

Figure 3-9 the interface for compressor selection

3.2.3.2 Compressor data

After selecting the compressor, users need to specify some detailed compressor data, represented in Figure 3-10. These details are rated EER, rated cooling capacity, rated inlet condition (superheat/return gas temperature) and compressor map equations. Moreover, if users recognize the total displacement, motor size, nominal speed and nominal voltage, they can also input these optional parameters. Otherwise, the HPDM will generate them itself.

Compressor Data

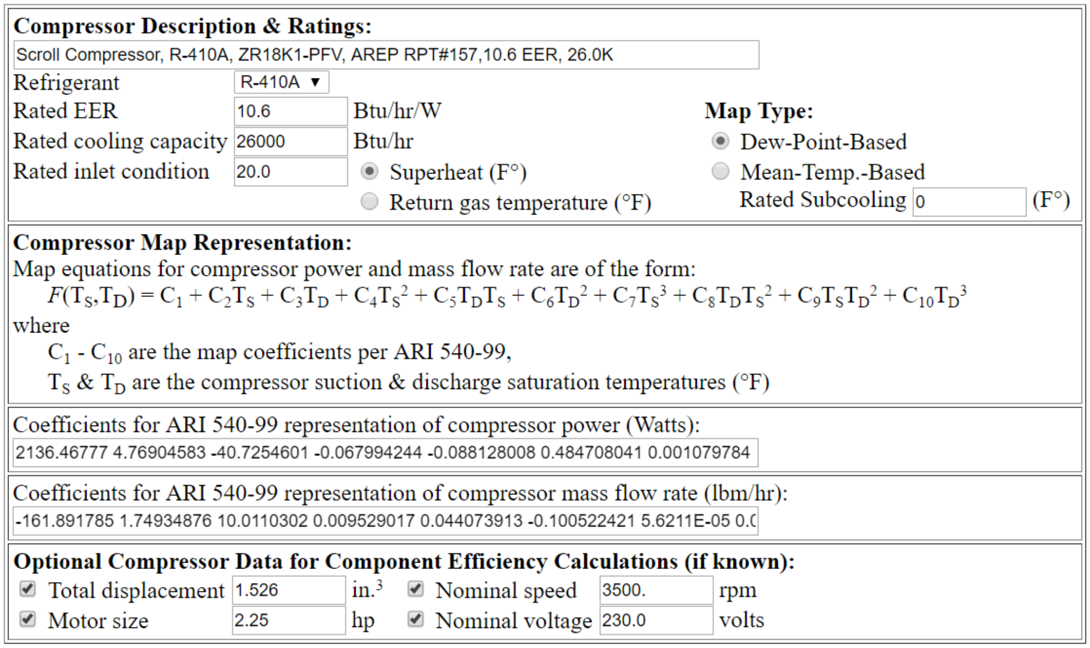

Figure 3-10 the interface of compressor data

A compressor map is an essential part of the compressor data. Based on the compressor map, the Heat Pump Design Model can generate the results of compressor power consumption and compressor mass flow rate. Equation 3.1 represents the system compressor map.

$$F(T_s, T_D) = C_1 + C_2T_s + C_3T_D + C_4T_s^2 + C_5T_DT_s + C_6T_D^2 + C_7T_s^3 + C_8T_DT_s^2 + C_9T_sT_D^2 + C_{10}T_D^3 \qquad (3.1)$$

where

$C_1 - C_{10}$ are the coefficients found by the linear regression method.

T_S and T_D are the compressor suction and discharge saturation temperature, respectively.

Typically, the rated compressor parameters can be found in the manufacturer brochures. However, the compressor map cannot be obtained very easily for some large companies. If a user cannot specify the compressor map for the refrigeration system, the system loop cannot be simulated, therefore, there is no doubt that users need to find a method which can generate a compressor map that is very similar to the manufacturers.

In industry, companies would like to scale compressor maps in order to get a new compressor map which is standard for those manufacturers. Thus, a scaling method to achieve this goal is introduced here.

Scaling method:

$$\text{Adjustment Factor(AF)} = \frac{Predicted\ Rated\ Capacity}{Based\ Rated\ Capacity} \tag{3.2}$$

$$B\_Mass\ Flow\ Rate \times \frac{Predicted\ Rated\ Capacity}{Based\ Rated\ Capacity} = P\_Mass\ Flow\ Rate \tag{3.3}$$

$$B\_Power\ Consumption \times \frac{Predicted\ Rated\ Capacity}{Based\ Rated\ Capacity} = P\_Power\ Consumption \tag{3.4}$$

Where

$B\_Mass\ Flow\ Rate$ is the mass flow rate of the based compressor map.

$P\_Mass\ Flow\ Rate$ is the mass flow rate of the predicted compressor map.

$B\_Power\ Consumption$ is the compressor power consumption of the based one.

$P\_Power\ Consumption$ is the compressor power consumption of the predicted one.

For this scaling method, the Adjustment Factor (AF) should be explained first. A baseline compressor map could be found with no trouble since there are large amounts of compressor maps provided by U.S. companies. In addition, the predicted rated cooling capacity and the base rated cooling capacity could be recognized conveniently from the manufactural data. Therefore, the AF can be computed from Equation 3.2.

The second step is to get every point of mass flow rate and power consumption within the different evaporating temperature and condensing temperature. According to Equations 3.3 and 3.4, the predicted mass flow rate and the predicted power consumption is not complicated to calculate. Here are some examples to illustrate this claim.

Typically, for the mini-split heat pump system, manufacturers would like to prefer the rotary compressor or scroll compressor since their operation range is more suitable for residential applications. From Figure 3-11, the cooling capacities of rotary compressors are around 1 ton, while the cooling capacities of scroll compressors are from 1 to 10 tons (Mitchell & Braun, 2011). Therefore, the examples focus more on scaling the rotary compressors and scroll compressors.

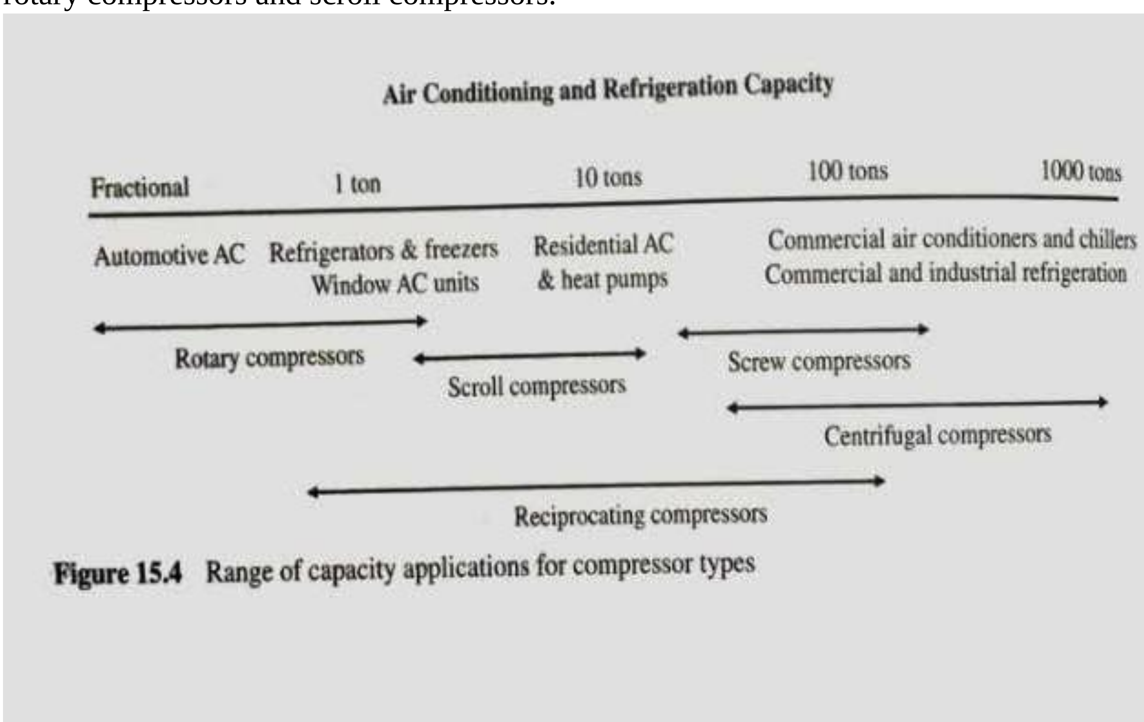

Figure 3-11 Ranges of capacity applications for different compressor types

Three compressor maps from the TECUMECH company are used for the rotary compressors. The rated cooling capacities are 5300 Btu/hr, 7125 Btu/hr and 10150 Btu/hr, respectively. Important to note is that all three of these compressors have the same voltage, frequency, refrigerant, phase, and application. Setting the rated cooling capacity of 7125 Btu/hr as the base rated cooling capacity, the rated cooling capacity of 5300 Btu/hr and 10150 Btu/ hr can be set as predicted rated cooling capacities. Meanwhile, according to Equation 3.1, the Adjustment Factor could be computed without any difficulties.

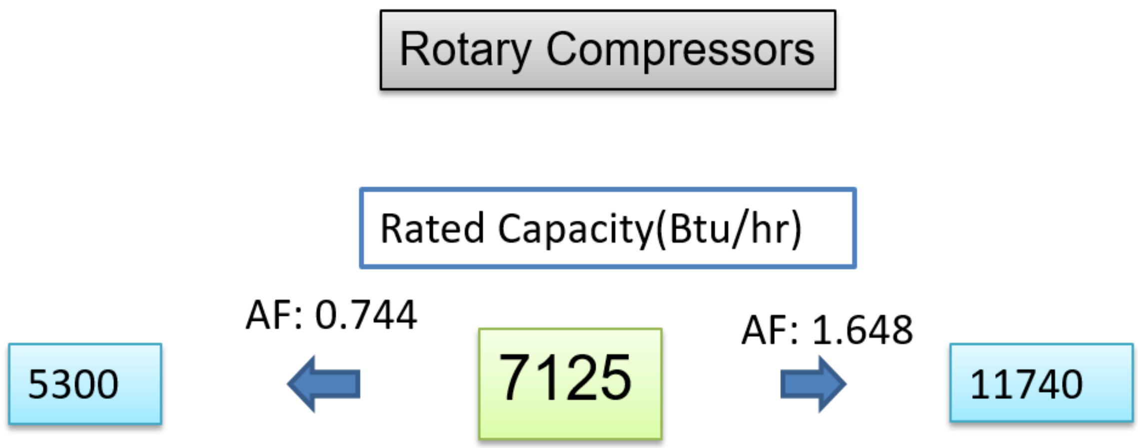

Figure 3-12 the AF for the rotary compressors examples

When utilizing the cooling capacity of the 7125 Btu/hr compressor map to predict the cooling capacity of the 11740 Btu/hr compressor map, the Adjustment factor is 1.648. The relative error calculation is shown in Equation 3.5.

$$\text{Relative error} = \frac{(\text{Predicted parameters} - \text{Based parameters})}{\text{Based parameters}} \tag{3.5}$$

For example, researchers can predict the mass flow rate and power consumption as the procedures as below.

Based on Appendix I, the rated cooling capacity of the compressor map for 7125 Btu/hr, the mass flow rate is 93.2 lb/hr and power consumption is 660 w, when the condensing temperature is 40 F and evaporating temperature is 120 F. Based on Appendix II, the rated cooling capacity of the compressor map for 11740 Btu/hr, the mass flow rate is 156 lb/hr and power consumption is 1110 w, when the condensing temperature is 40 F and evaporating temperature is 120 F.

Predicted mass flow rate

= Based mass flow rate * AF

$= 93.2\ \text{lbm/hr} * 1.648$

$= 153.6\ \text{lbm/hr}$

Predicted power consumption

= Based power consumption * AF

$= 660\ \text{w} * 1.648$

$= 1087.7\ \text{w}$

After obtaining the predicted mass flow rate and power consumption at the specified point, researchers can compute the relative errors for these two parameters.

Relative error of the mass flow rate

$$= \frac{(Predicted\ mass\ flow\ rate - Based\ mass\ flow\ rate)}{Based\ mass\ flow\ rate}$$

$$= \frac{153.6\ \mathrm{lbm/hr} - 156\ lbm/hr}{156\ lbm/hr}$$

= -1.56%

Relative error of power consumption

$$= \frac{(Predicted\ power\ consumption - Based\ power\ consumption)}{Based\ power\ consumption}$$

$$= \frac{1087.7\ \mathrm{w} - 1110\ w}{1110\ w}$$

= -2.03%

The example point is marked as red in Table 3-1 and Table 3-2. Moreover, this is an example of a specified point, but researchers need to generate all points in every condensing temperature and evaporating temperature. Microsoft Excel can be the utilization tool to achieve these. After computing relative errors for mass flow rates and power consumptions, the prediction from rated cooling capacity 7125 Btu/hr to rated cooling capacity 11740 Btu/hr, relative errors can be calculated. The relative errors of power consumptions and mass flow rates are shown in Tables 3-1 and 3-2.

From these two tables, researchers can easily deduct that the maximum relative error for power consumption is - 4.73%, which is marked as blue in Table 3-1, when the condensing temperature is -15 F and the evaporating temperature is 80 F. The average of relative errors for every condition is -2.67% and the standard deviation is determined to be 0.80% after computing in Excel. Simultaneously, the maximum relative error for mass flow rate is -5.75% which can be observed in a blue font in Table 3-2, when the condensing temperature is -15 F and evaporating temperature is 80 F. The average of relative errors for every situation is -1.39% and the standard deviation is 0.99%.

Table 3-1 Relative errors for power consumption prediction from 7125 Btu/hr to 11740 Btu/hr

Relative Error	Tevap(F)	Power consumption prediction					
Tcond(F)	80	90	100	110	120	130	140
-15	-4.73%	-4.72%					
-10	-4.19%	-3.76%	-3.49%				
-5	-3.50%	-3.23%	-2.90%	-2.61%			
0	-3.25%	-2.89%	-2.70%	-2.34%	-2.03%		
5	-3.14%	-2.82%	-2.44%	-2.03%	-1.77%		
10	-3.09%	-2.68%	-2.34%	-2.04%	-2.11%	-1.89%	-0.71%
15	-3.09%	-2.59%	-2.36%	-1.92%	-1.61%	-1.73%	-0.86%
20	-3.23%	-2.74%	-2.53%	-2.18%	-1.45%	-1.14%	-1.40%
25	-3.43%	-3.03%	-2.52%	-2.20%	-1.75%	-1.70%	-1.53%
30	-3.65%	-3.04%	-2.73%	-2.13%	-2.34%	-1.69%	-1.78%
35	-3.81%	-3.27%	-2.78%	-2.93%	-2.47%	-2.09%	-1.64%
40	-3.74%	-3.34%	-3.04%	-3.09%	-2.03%	-1.95%	-1.76%
45	-4.09%	-3.65%	-3.03%	-2.62%	-2.76%	-2.07%	-2.24%
50	-3.97%	-3.55%	-3.25%	-2.93%	-3.05%	-2.60%	-2.23%
55	-3.86%	-3.46%	-3.19%	-2.99%	-2.77%	-2.73%	-2.58%

Table 3-2 Relative errors for mass flow rate prediction from 7125 Btu/hr to 10150 Btu/hr

Relative Error	Tevap(F)	Mass flow rate prediction					
Tcond(F)	80	90	100	110	120	130	140
-15	-5.75%	-4.82%					
-10	-4.18%	-3.38%	-2.65%				
-5	-3.00%	-2.48%	-1.91%	-0.96%			
0	-2.28%	-1.99%	-1.43%	-0.78%	-0.02%		
5	-1.91%	-1.60%	-1.27%	-0.87%	-0.15%		
10	-1.60%	-1.47%	-1.21%	-0.82%	-0.48%	0.16%	0.76%
15	-1.62%	-1.27%	-1.17%	-0.96%	-0.81%	-0.41%	0.13%
20	-1.28%	-1.43%	-0.83%	-1.14%	-1.46%	-0.87%	-0.54%
25	-1.01%	-1.53%	-1.14%	-1.70%	-1.43%	-1.29%	-0.98%
30	-0.90%	-0.78%	-1.50%	-1.51%	-1.53%	-1.54%	-1.55%
35	-0.72%	-0.92%	-1.14%	-1.25%	-1.49%	-1.86%	-2.13%
40	-0.95%	-0.55%	-1.44%	-1.14%	-1.56%	-2.11%	-1.92%
45	-0.27%	-0.25%	-0.77%	-1.32%	-1.33%	-1.73%	-2.35%
50	0.30%	-0.97%	-0.97%	-1.48%	-1.49%	-2.39%	-2.43%
55	-0.12%	-0.09%	-0.83%	-1.29%	-2.09%	-2.12%	-2.48%

Most of the time, the manufacturer will use rotary compressors for mini-split heat pump systems in residential applications. Therefore, one more example of compressor map prediction is provided as follows.

With the same principles, utilizing rated cooling capacity of the 7125 Btu/hr compressor map to predict rated cooling capacity for the 5300 Btu/hr compressor map, the results for relative errors are shown in Tables 3-3 and 3-4. The Adjustment Factor of this prediction is 0.744.

Table 3-3 Relative errors for power consumption prediction from 7125 Btu/hr to 5300 Btu/hr

Relative Error	Tevap(F)	Power consumption prediction					
Tcond(F)	80	90	100	110	120	130	140
-15	-2.35%	-2.73%					
-10	-2.54%	-2.37%	-2.49%				
-5	-2.05%	-2.40%	-2.23%	-2.33%			
0	-1.96%	-2.01%	-2.06%	-2.16%	-2.25%		
5	-1.95%	-1.99%	-2.03%	-1.89%	-1.98%		
10	-2.09%	-1.84%	-1.89%	-1.92%	-1.79%	-1.68%	-1.62%
15	-1.79%	-1.83%	-1.81%	-1.67%	-1.51%	-1.41%	-1.37%
20	-1.87%	-1.63%	-1.62%	-1.66%	-1.34%	-1.40%	-1.18%
25	-1.81%	-1.77%	-1.55%	-1.38%	-1.23%	-1.29%	-1.21%
30	-2.14%	-1.57%	-1.55%	-1.37%	-1.22%	-1.10%	-0.86%
35	-2.01%	-1.65%	-1.43%	-1.21%	-1.07%	-0.96%	-0.73%
40	-2.05%	-1.67%	-1.38%	-1.15%	-1.02%	-0.91%	-0.69%
45	-2.43%	-1.98%	-1.45%	-1.21%	-0.87%	-0.77%	-0.53%
50	-2.42%	-1.94%	-1.59%	-1.32%	-0.97%	-0.68%	-0.45%
55	-2.88%	-1.98%	-1.61%	-1.33%	-0.77%	-0.64%	-0.57%

Table 3-4 Relative errors for mass flow rate prediction from 7125 Btu/hr to 5300 Btu/hr

Relative Error	Tevap(F)	Mass flow rate prediction					
Tcond(F)	80	90	100	110	120	130	140
-15	-3.74%	-1.17%					
-10	-2.79%	-0.71%	1.26%				
-5	-2.12%	-0.82%	0.99%	3.00%			
0	-1.66%	-0.66%	1.02%	2.61%	4.08%		
5	-1.63%	-0.68%	0.65%	1.88%	3.56%		
10	-1.19%	-0.26%	0.33%	1.22%	2.00%	3.12%	4.64%
15	-1.24%	-0.51%	0.55%	1.69%	2.73%	4.19%	5.89%
20	-0.87%	-0.32%	0.11%	0.78%	1.74%	2.37%	3.56%
25	-0.69%	-0.42%	0.19%	0.71%	1.26%	1.91%	2.78%
30	-0.59%	-0.30%	0.05%	0.42%	0.82%	1.44%	2.10%
35	-0.47%	-0.24%	0.00%	0.37%	0.65%	1.00%	1.49%
40	-0.56%	0.36%	-0.11%	0.30%	0.47%	0.71%	1.07%
45	0.17%	0.26%	0.35%	0.44%	0.68%	0.53%	0.78%
50	0.35%	-0.27%	-0.09%	0.22%	0.55%	-0.02%	0.32%
55	-0.09%	0.38%	-0.10%	0.29%	-0.11%	0.30%	-0.01%

Based on these two tables listed above, the maximum relative error for power consumption is -2.88% which is marked as blue in Table 3-1, when the condensing temperature is 55 F and evaporating temperature is 80 F. The average of relative errors for every condition is -1.61% and the standard deviation is 0.54%. Meanwhile, the maximum relative error for mass flow rate is 5.89% which can be seen in a blue font in Table 3-2, when the condensing temperature is 15 F and evaporating temperature is 140 F. The average of relative errors for every situation is 0.60% and the standard deviation is 1.55%.

From these two examples analyzed above, the maximum relative error for mass flow rate prediction and power consumption prediction is around 5% and average relative error is below 5%. Thus, the compressor map prediction method is convincing for rotary compressors.

For the scroll compressors, two compressor maps from the TECUMECH company are provided afterward. The rated cooling capacities are 28,999 Btu/hr and 56,898 Btu/hr, respectively. In addition, the Adjustment Factor is 1.962, presented in Figure 3-13.

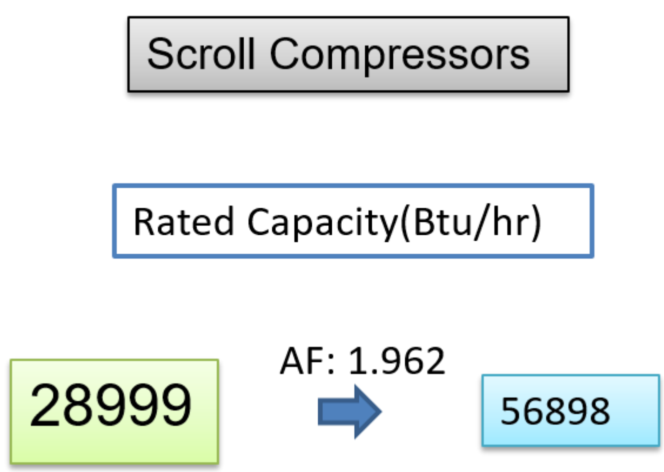

Figure 3-13 The AF for the scroll compressors example

Scroll compressors are also widely utilized in residential applications and the rated cooling capacity is around 1 to 10 tons, based on Figure 3-11. In order to test scaling method feasibility for scroll compressors, one example would like to be provided as follows.

In this example, a compressor map of the rated cooling capacity for 28,999 Btu/hr is used to predict a compressor map of the rated cooling capacity for 56,898 Btu/hr. Also, mass flow rate and power consumption are the parameters that need to be predicted. The scaling principles are the same as the rotary compressors prediction.

Table 3-5 Relative errors for power consumption prediction from 28999 Btu/hr to 56898 Btu/hr

Relative Error	Tevap(F)	Power consumption prediction					
Tcond(F)	80	90	100	110	120	130	140
-15	1.93%	2.50%					
-10	2.20%	2.14%	3.05%				
-5	2.80%	2.39%	3.00%	2.67%			
0	2.48%	2.37%	2.70%	3.10%	3.50%		
5	2.18%	2.37%	2.94%	3.08%	3.08%		
10	2.20%	2.37%	2.94%	3.08%	3.08%	3.34%	3.59%
15	2.56%	2.39%	2.70%	2.62%	3.29%	3.36%	3.77%
20	2.62%	2.72%	2.46%	3.10%	3.31%	3.38%	3.80%
25	2.37%	2.47%	2.75%	3.12%	3.33%	3.58%	3.53%
30	2.46%	2.52%	2.53%	2.93%	3.16%	3.44%	3.73%
35	2.55%	2.29%	2.57%	2.73%	2.99%	3.47%	3.61%
40	2.67%	2.37%	2.62%	2.77%	3.03%	3.51%	3.82%
45	2.43%	2.45%	2.42%	2.82%	3.07%	3.38%	3.70%
50	1.82%	2.53%	2.75%	2.87%	3.11%	3.42%	3.76%
55	2.34%	1.98%	2.54%	2.91%	3.16%	3.66%	3.64%

Table 3-6 Relative errors for mass flow rate prediction from 28999 Btu/hr to 56898 Btu/hr

Relative Error	Tevap(F)	Mass flow rate prediction					
Tcond(F)	80	90	100	110	120	130	140
-15	-0.11%	0.09%					
-10	-0.05%	0.11%	-0.60%				
-5	0.00%	0.12%	-0.47%	0.01%			
0	-0.24%	0.13%	-0.39%	0.01%	0.16%		
5	-0.43%	-0.38%	-0.32%	0.02%	0.13%		
10	-0.37%	-0.33%	-0.51%	0.01%	-0.40%	-0.30%	0.18%
15	-0.13%	-0.09%	-0.46%	-0.21%	0.07%	0.18%	0.12%
20	-0.31%	-0.09%	-0.06%	-0.21%	-0.16%	0.13%	0.28%
25	-0.29%	-0.27%	-0.08%	-0.05%	-0.17%	-0.10%	0.21%
30	-0.44%	-0.12%	-0.10%	-0.07%	-0.03%	-0.13%	0.31%
35	-0.31%	-0.42%	-0.13%	-0.25%	-0.08%	0.12%	0.07%
40	-0.45%	-0.31%	-0.29%	-0.15%	0.01%	0.06%	0.14%
45	-0.24%	-0.34%	-0.21%	-0.08%	0.06%	-0.01%	0.06%
50	-0.38%	-0.27%	-0.36%	-0.24%	-0.11%	0.14%	0.10%
55	0.14%	0.16%	-0.59%	-0.39%	-0.56%	-0.54%	0.11%

After calculated by Excel, the relative errors for the power consumption prediction and mass flow rate prediction are listed in Tables 3-5 and 3-6. All of the absolute value results are below 5%. The average relative error for power consumption is 2.89% and the maximum is 3.82%, which is marked as a blue font when the condensing temperature is 40 F and evaporating temperature is 140 F. Additionally, Standard deviation is an important method to quantify the amount of variation (Bland & Altman, 1996). The lower the standard deviation is; the more data points are close to the mean value (WIKIPEDIA, 2016b). Otherwise, the higher the standard deviation is; the fewer data points are close to the average value. The standard deviation of relative errors for power consumption is 0.49%, which is very small. For another, the mean value of relative error for mass flow rate is -0.13%. The absolute value for a maximum of relative error for mass flow rate prediction is 0.6% and the standard deviation is 0.22%, which is also very small. Based on these values mentioned above, a conclusion can be obtained by researchers that the scaling method is also suitable for scroll compressor performance prediction, since the average value and maximum value for the relative error are both lower than 5% and the standard deviations are relatively small.

In this section, rotary compressor map prediction and scroll compressor map prediction for mass flow rate and power consumption were explored. Based on performance values, the relative error for these two parameters is relatively small. Therefore, the scaling method for these two compressor maps is reasonable. In addition, these two kinds of compressors are widely utilized for the mini-split heat pump system. Therefore, this method can be used in the mini-split heat pump system's compressor map prediction.

3.2.3.3 Compressor calibration

There are two functions for compressor calibration: scaling compressor performance and scaling system performance. When users want to scale the compressor performance, they can change the value for EER, capacity, and voltage by inputting the exact value or by inputting a multiplier. For the scaling system performance part, researchers can scale the system performance by inputting a scaling system capacity within a specific range. Figures 3-14 and 3-15 show the interfaces for scaling compressor performance and scaling system performance, respectively. Typically, users will not utilize the compressor and system performance scaling since the Heat Pump Design Model will get a result by simulation. However, non-convergence will happen in the HPDM simulation and it will show a system server error on the computer. In this time, users will need to scale the system capacity within a certain range.

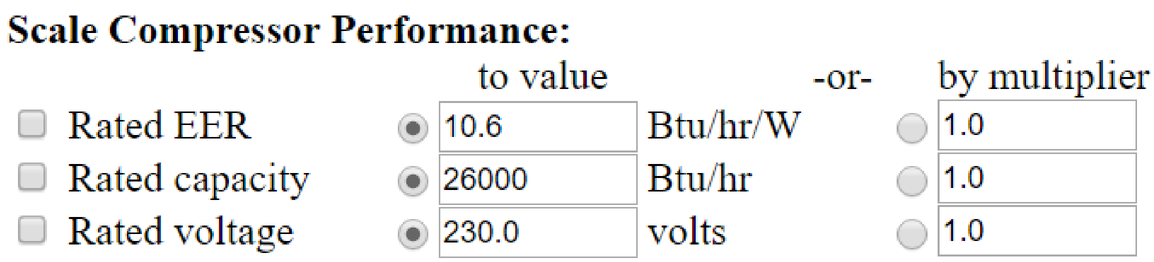

Figure 3-14 the interface for scaling compressor performance

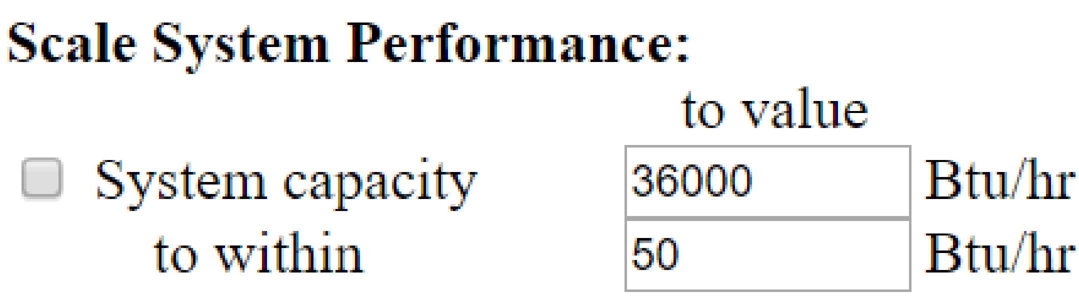

Figure 3-15 the interfaces for scaling system performance

3.2.4 Fin-and-Tube Heat Exchanger Parameters and Configurations

In this section, heat exchanger parameters and configurations will be discussed specifically for indoor units and outdoor units. Figures 3-16 and 3-17 are the interfaces for the indoor unit configurations and outdoor unit geometries, separately. In addition, the Heat Pump Design Model requires the same heat exchanger configurations for both indoor units and outdoor units in cooling mode and heating mode. Fortunately, almost all of the configurations parameters can be found on the manufacturer brochures. If researchers can possess the units or units can be donated from manufacturers to researchers, the configurations parameters can be measured by researchers without difficulties.

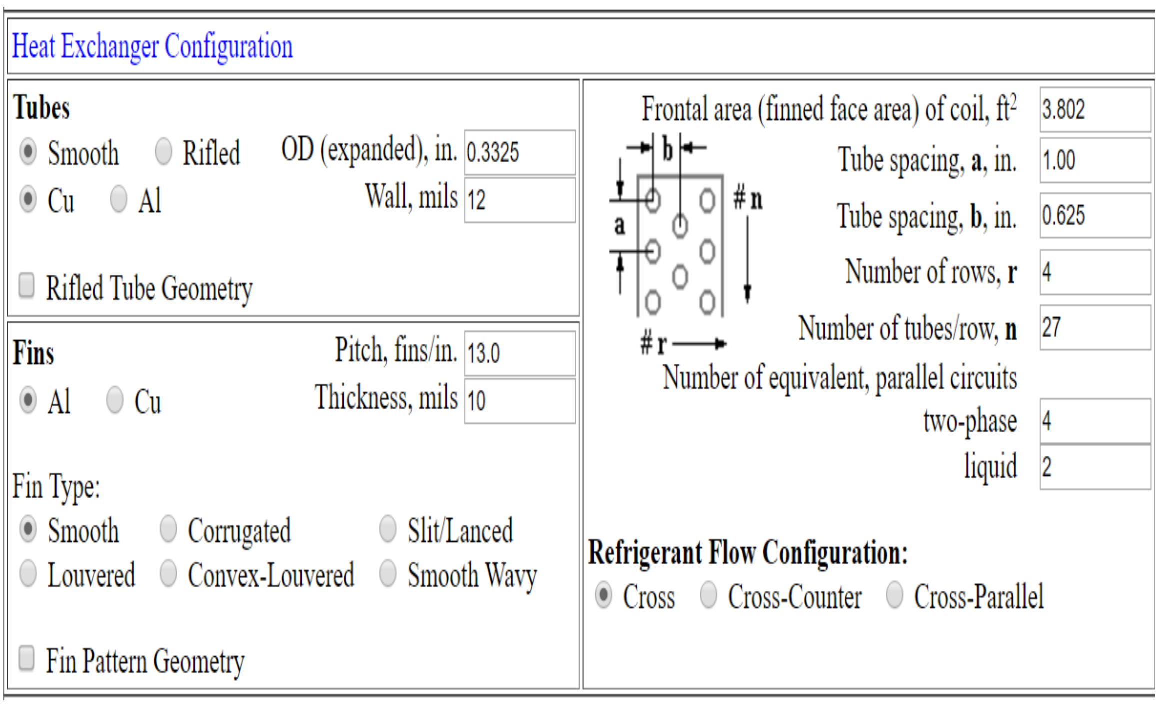

Figure 3-16 the interface of indoor unit heat exchanger configuration

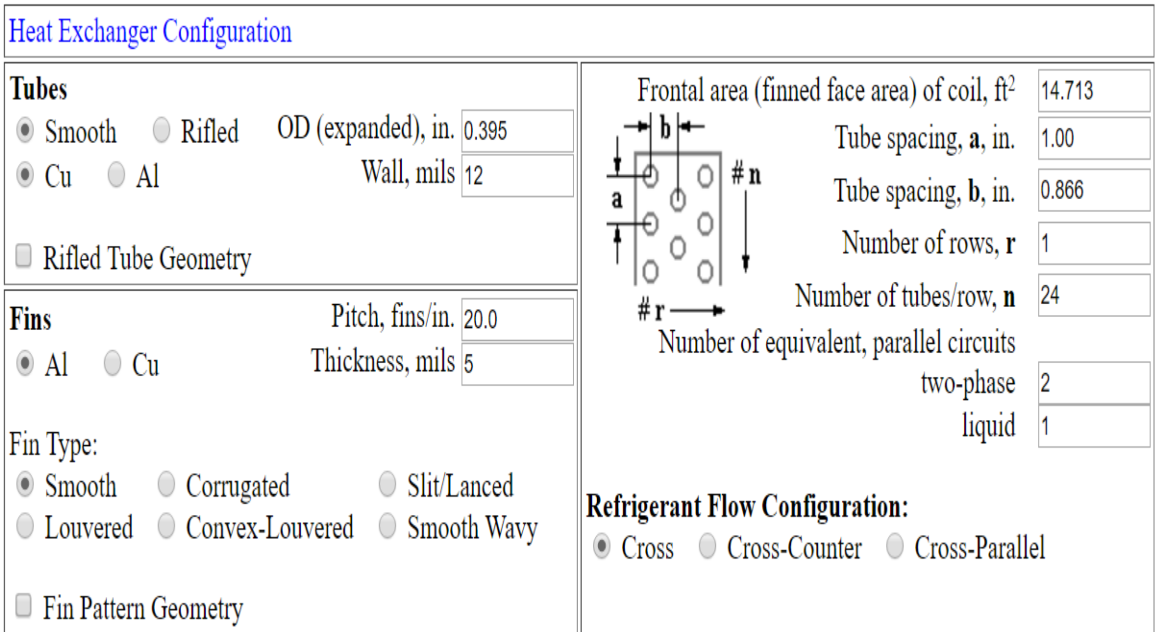

Figure 3-17 the interface of outdoor unit heat exchanger configuration

There are several heat exchanger configuration parameters that should be specified.

- Tube parameters
- Fin parameters
- Tube spacing and rows

First, for tube parameters, tube type (smooth or rifled) and material (copper or aluminum) need to be determined. Moreover, the outer diameter and the wall thickness of the tube should be input into the Heat Pump Design Model.

Second, in order to achieve the fin parameters, the material and type should also be determined by users. The material may also be either copper or aluminum. Furthermore, the fin pitch and thickness need to be discerned. Fin pitch is the spacing between the adjacent fins (Shawn, 2016) and the fin thickness is the thickness of a single fin. In order to explain unambiguously, the tube outer diameter, the wall thickness of a tube, fin pitch and fin thickness are presented in Figure 3-18.

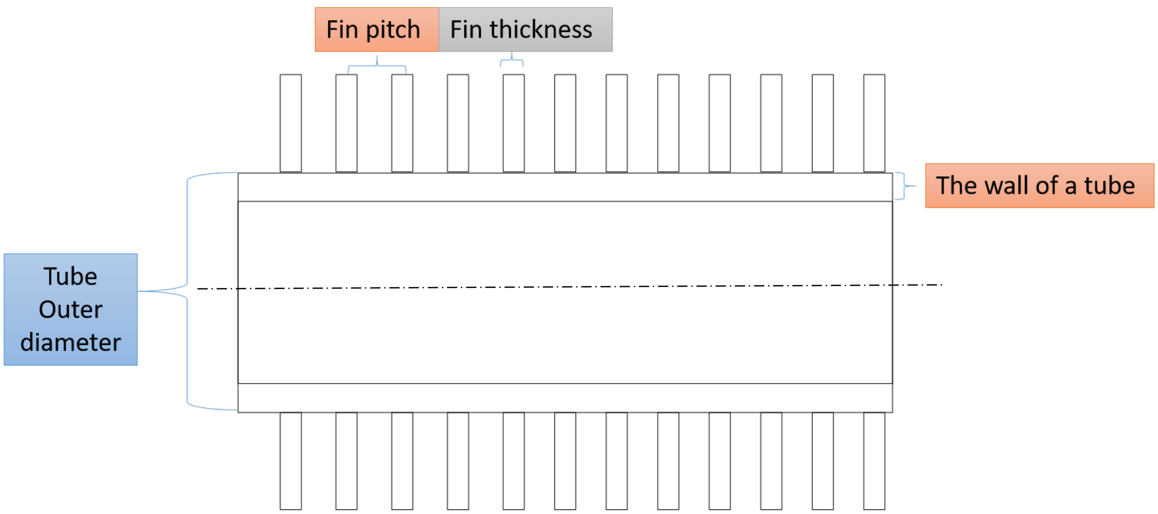

Figure 3-18 the geometry diagram of a tube

Finally, for the tube spacing and rows, there are also some geometry parameters needed to be specified, which include finned face area of a coil, tube spacing in longitude and transversal, the number of rows, the number of tubes in each row and the number of equivalent, parallel circuits for two-phase and liquid phase.

For the Heat Pump Design Model, the following parameters are able to be explained clearly. “P” is the finned face area of a coil. “a” is the longitudinal center-to-center distance between tubes and “b” is the transverse center-to-center distance between tubes. “h” is the height for the heat exchanger, and “d” is the depth for this heat exchanger. “#r” is the number of tubes in each row and “#n” is the number of rows. These are shown in Figure 3-19.

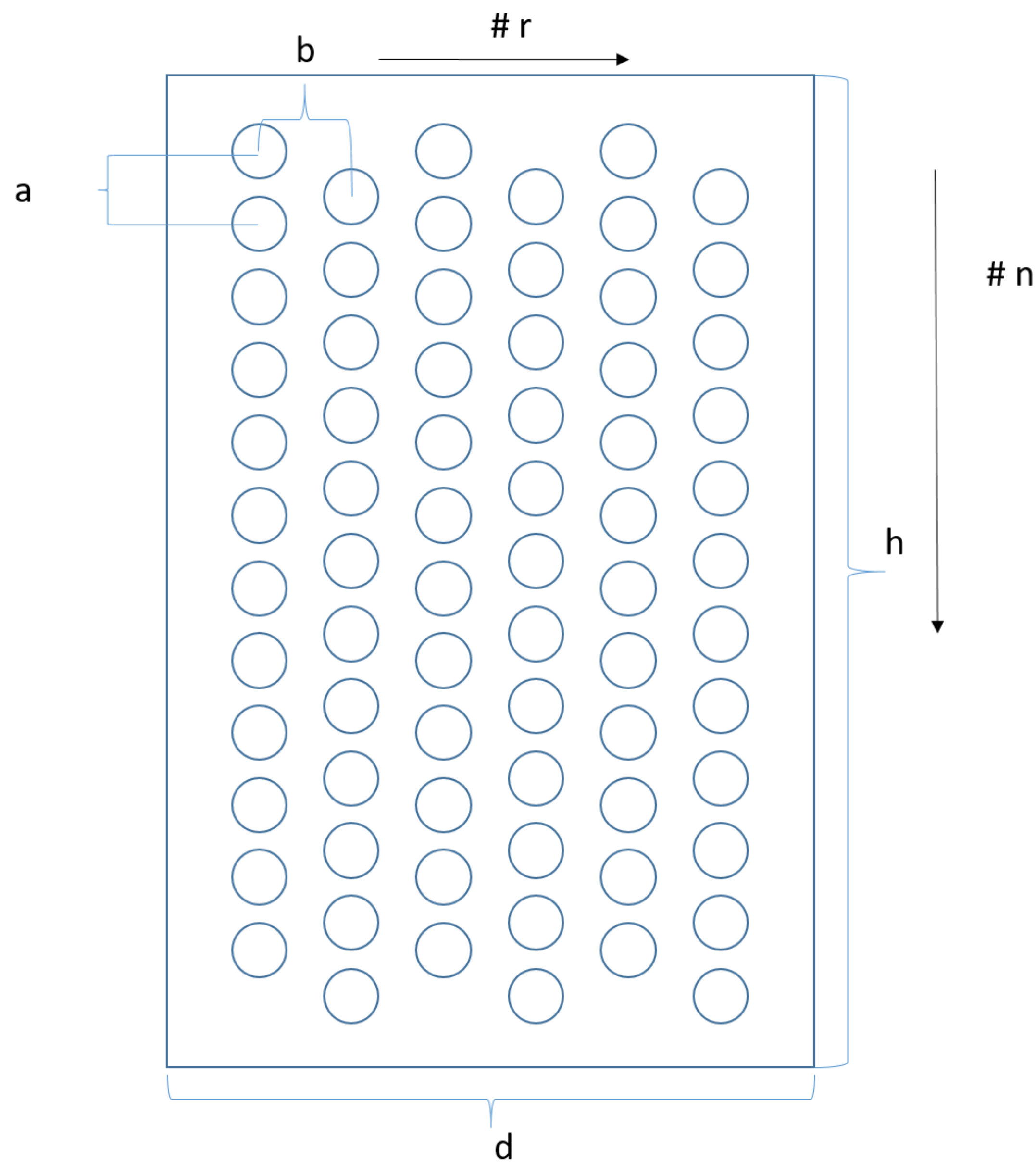

Figure 3-19 a transverse figure for a heat exchanger

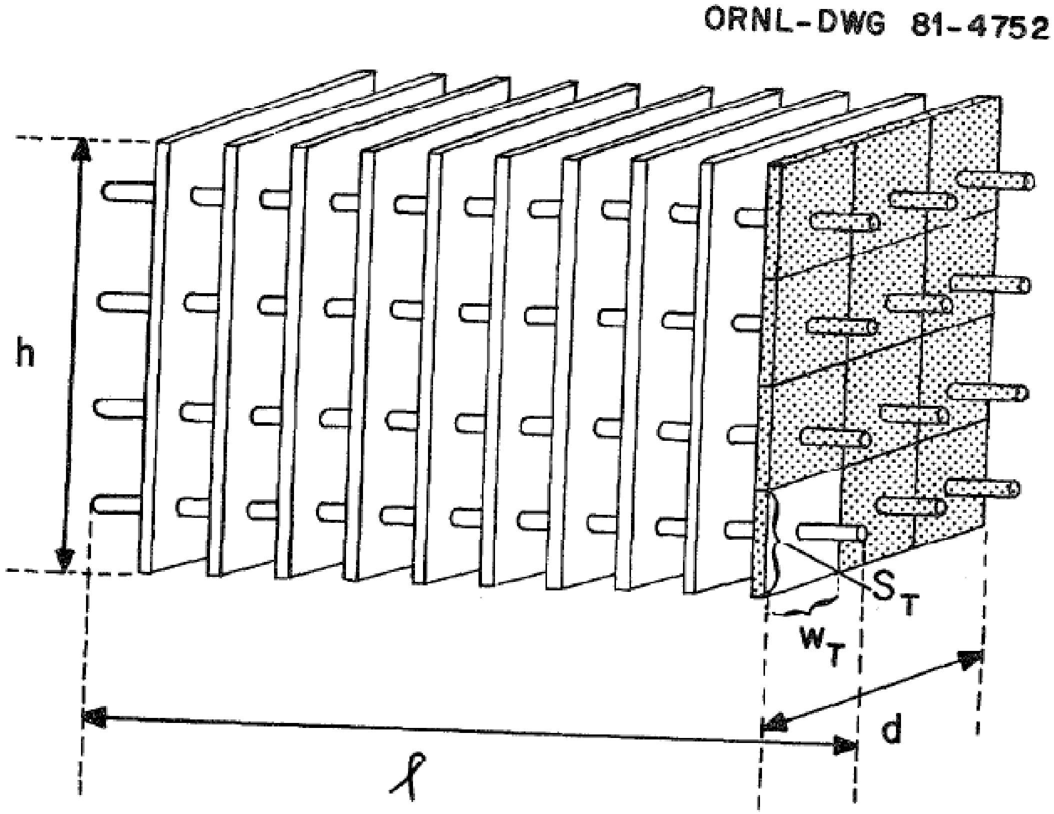

Figure 3-20 a 3D figure for a heat exchanger

The Figure 3-19 is a schematic diagram of a transverse figure for a heat exchanger. According to this figure, researchers can discern most of the parameters in this section. The finned face area of a coil, "P", can be defined as the product of the length of the heat exchanger and the width of the heat exchanger, which can be defined as the equation 3.6. Other detailed parameters can be found in Figures 3-19 and 3-20 easily, except for the number of equivalent, parallel circuits for two-phase and liquid phase.

$$P = d * h \tag{3.6}$$

The Figure 3-20 is a 3D figure for a heat exchanger and it can be understood much deeper than the Figure 3-19. This figure shows some basic geometry parameters (Fischer & Rice, 1983) obtained by the Oak Ridge National Laboratory.

In the last part of this section, the terms "two-phase" and "liquid" regions of the number of equivalent and parallel circuits for indoor units and outdoor units are explained. The Heat Pump Design Model divides the heat exchanger into two sections: two-phase region and liquid region. For an evaporator, the type of equivalent and parallel circuits can be separated as a two-phase region and superheat region, which is shown in Figure 3-21. For a condenser, the type of equivalent and parallel circuits can be separated as a two-phase region and subcooling region (Hahn, 1992), which is presented in Figure 3-22. These two figures are sketches for a typical heat exchanger, and researchers can understand this parameter more deeply.

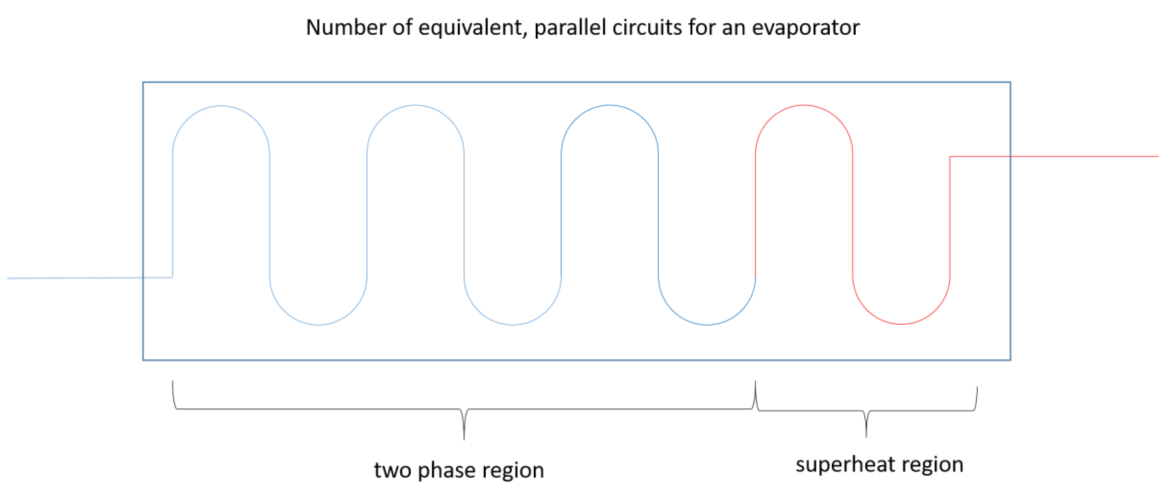

Figure 3-21 the diagram for number of equivalent, parallel circuits for an evaporator

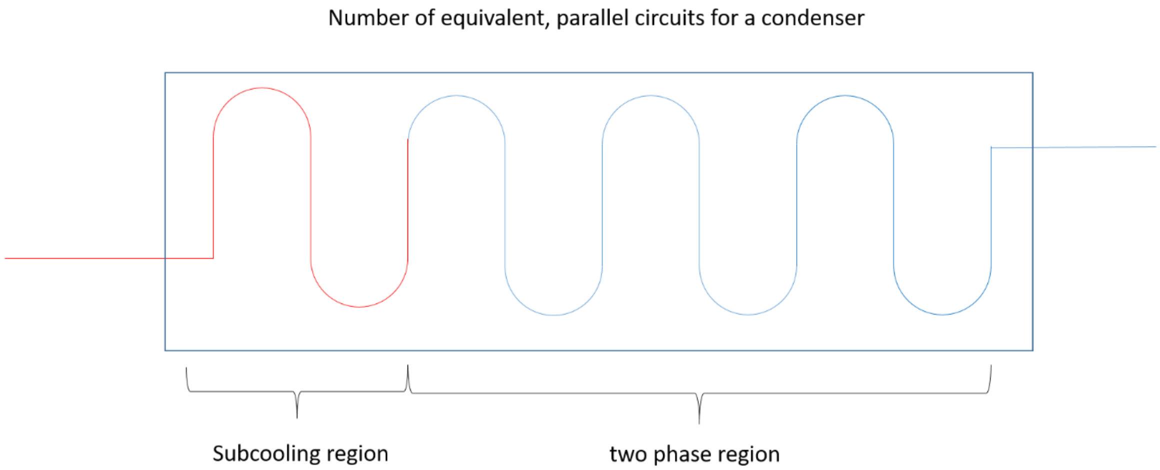

Figure 3-22 the diagram for number of equivalent, parallel circuits for an evaporator

3.2.5 System Operating Conditions

In this part, the Heat Pump Design Model asks the users the specify several key parameters which are listed below.

- Indoor and outdoor dry-bulb and wet-bulb temperature
- Indoor and outdoor air flow rate
- Indoor and outdoor blower power consumptions
- Static pressure
- Vapor lines and liquid lines outer diameters

For these five kinds of parameters, users can input any values into the HPDM to get results. In addition, whether manufacturers or lab testers, the first four parameters will be provided. But the last parameter can be discerned from the manufacturer brochures without difficulties.

3.2.6 Simulation results

After inputting all of the parameters, the Heat Pump Design Model will perform the simulation and output every key point performance result for users. Moreover, system performance values will also be provided by the HPDM. The system capacity, power consumption, mass flow rate, sensible heat ratio and system EER or COP are included in the output results. Typically, there are some key points for a refrigeration cycle, which are presented as Point 1 through 7 in Figure 3-23. Point 1 and Point 2 represent the inlet and outlet conditions for a compressor. Additionally, Point 3 and Point 4 show the inlet and outlet situations for a condenser. Simultaneously, point 5 presents the inlet condition for an expansion valve. Finally, point 6 and point 7 indicate the inlet and outlet circumstances for an evaporator. These seven key points are significant for evaluating a vapor compression cycle. Therefore, the Heat Pump Design Model will generate performance values for every key point, with respect to pressure, dry-bulb temperature, saturation temperature, and enthalpy. For Points 1, 2 and 4, the superheat temperature will be provided in the results interface. For points 5 and 6, the subcooling temperature is shown in the printed results interface, too.

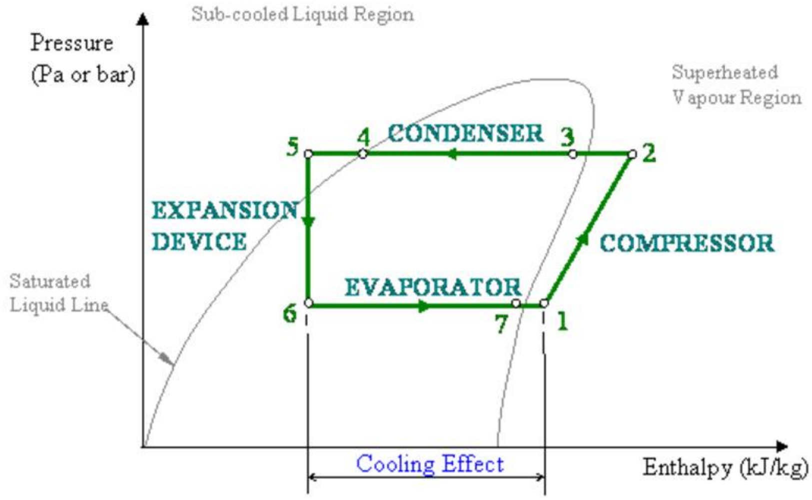

Figure 3-23 Key points for a vapor compression cycle in a P-h diagram
Source:(website owner, 2016)

Moreover, the component sizing and system charge will also be provided in the results interface. The system charge for the simulation results should be equal or similar to the provided laboratory data or manufactural data, and system charge comparison can be a good evaluation method for simulation results. Sometimes, researchers cannot find all of the geometry data, so they need to estimate some values for configuration data based on similarly rated cooling capacity heat pump systems. If the relative error of the system charge is less than 5%, then the estimated values are within a reasonable range. Component sizing was not considered in this thesis, but there is still a large potential to utilize it in other research objectives.

The printed results are shown in Figures 3-24 and 3-25, which include system operating conditions, component sizing, charge and performance data. The system operating condition and the performance data are in the middle of Figure 3-24. There are seven large numbers in Figure 3-24, corresponding to the seven numbers in Figure 3-23.

Since the Heat Pump Design Model lists the seven key point performance data in the printed results, researchers can know exactly the refrigeration cycle performance simulation conditions. The component sizing and system charge are listed in Figure 3-25, but this project only focused on the system charge in both cooling mode and heating mode.

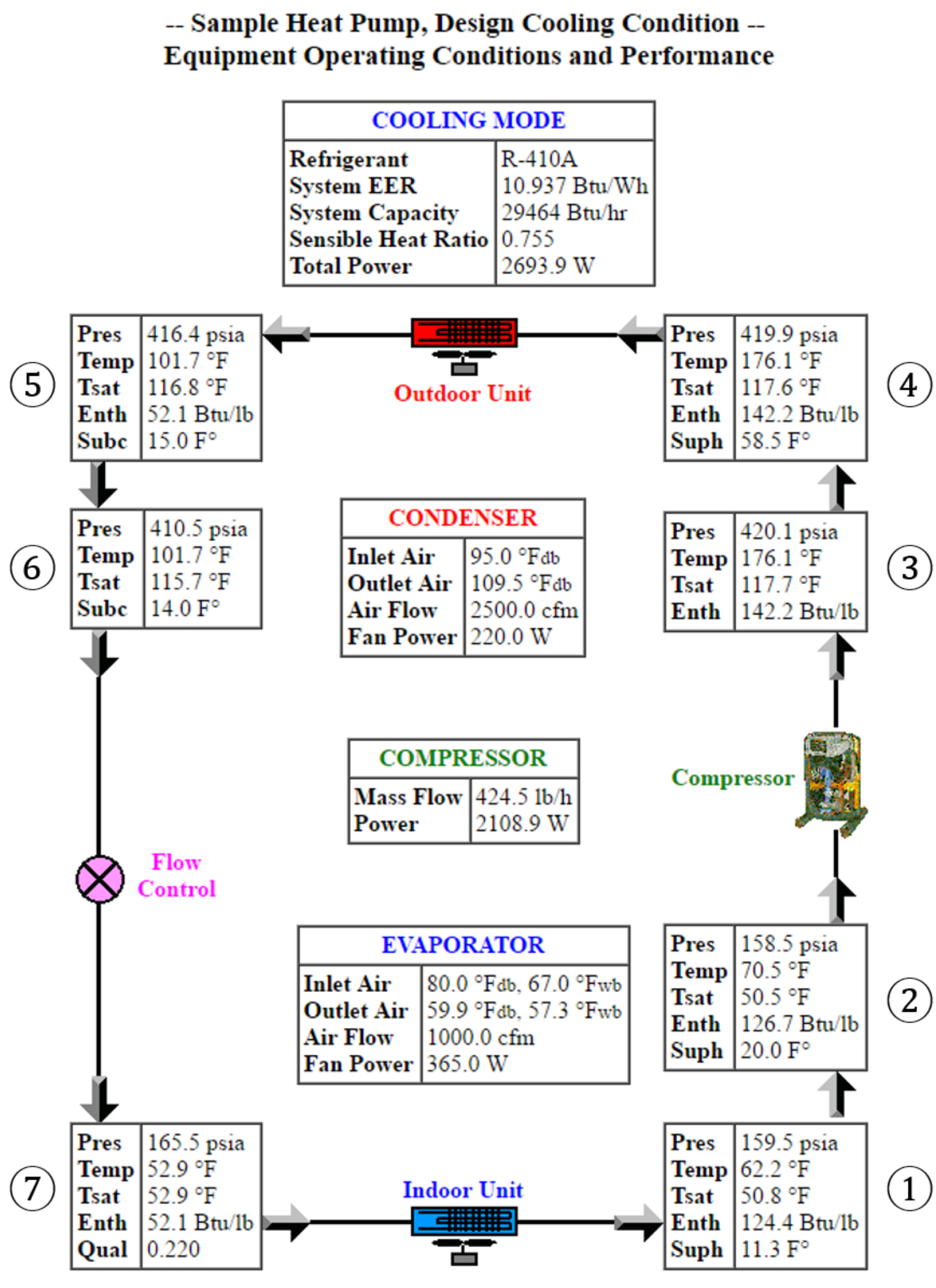

Figure 3-24 Printed results for system operating conditions and performance

-- Sample Heat Pump, Design Cooling Condition --
Component Sizing, Charge, and Performance

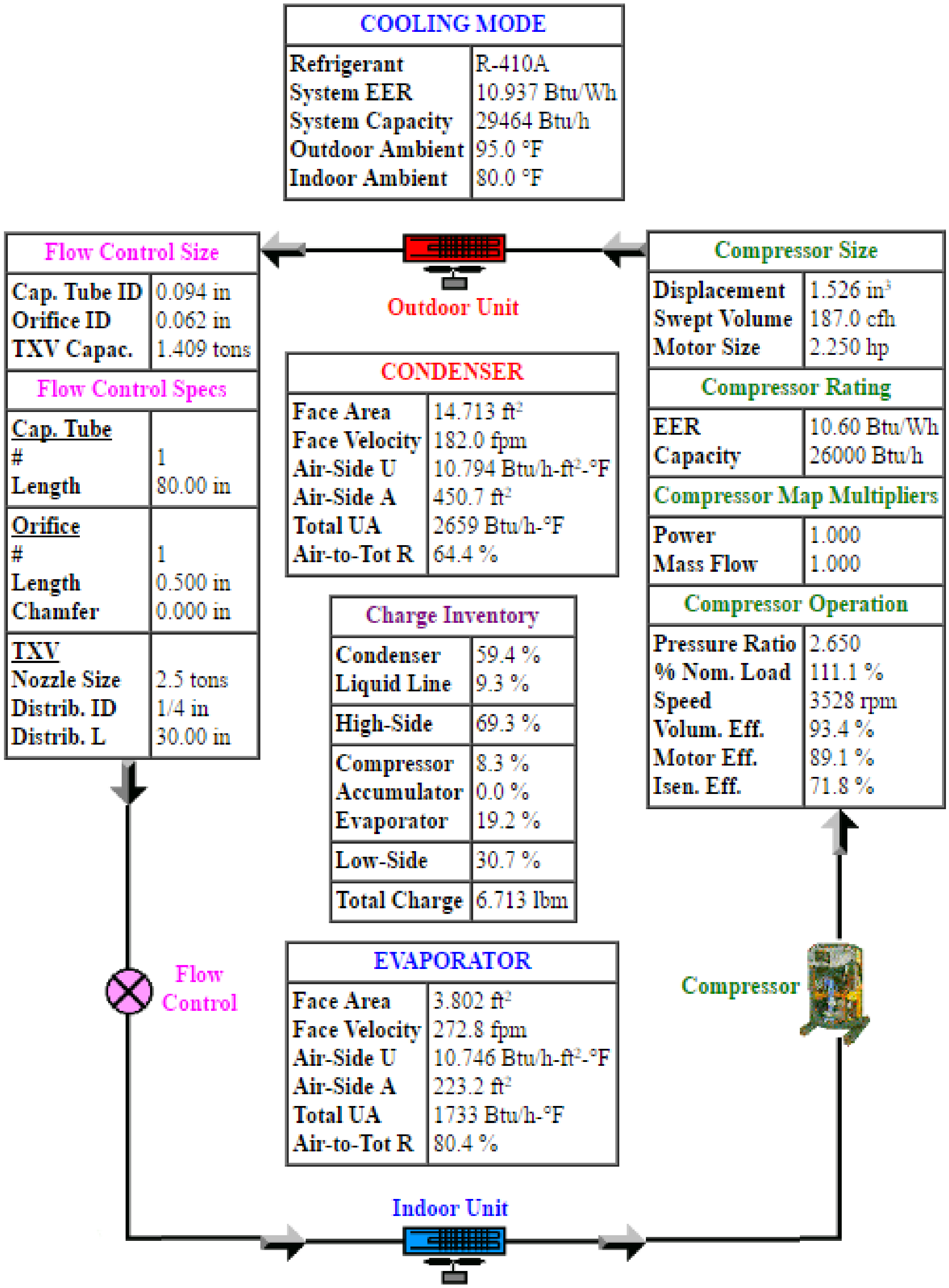

Figure 3-25 Printed results for component sizing and system charge

After specifying all the input parameters, the Heat Pump Design Model will do the calculation to generate the outputs. A simple schematic diagram is shown in Figure 3-26 which summarizes all of the input values and output parameters that users should identify. Therefore, users can understand the HPDM's computation logic easily enough.

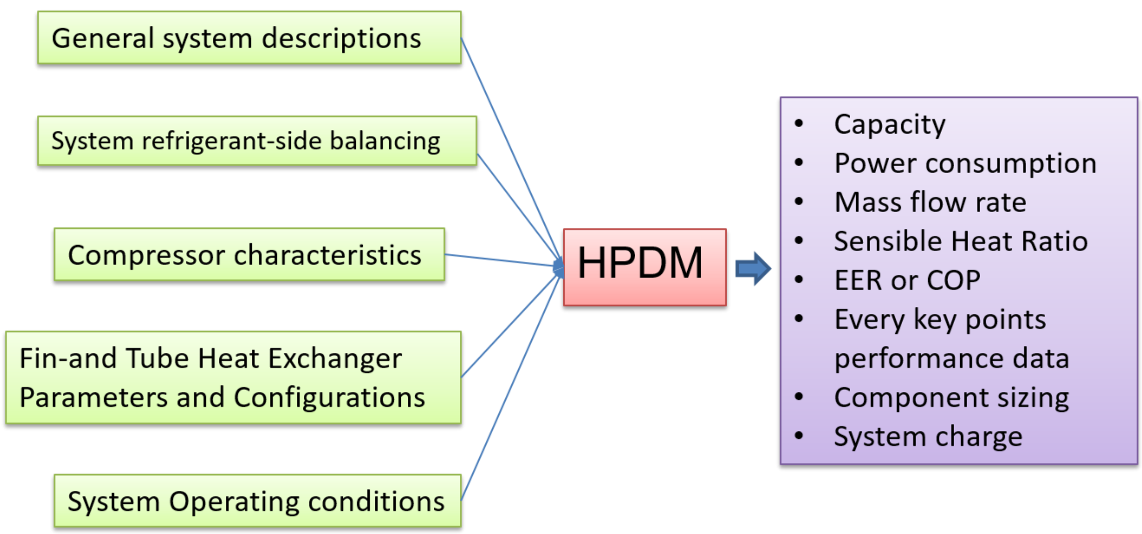

Figure 3-26 Detailed inputs and outputs for the HPDM

In this section, input parameters and output performance values are discussed. In order to operate the Heat Pump Design Model proficiently, explanations of every input value are provided and output interpretations are offered. In addition, the logic flow for utilizing the HPDM is also provided in this chapter, thus, users can understand the processing methodology of the HPDM conveniently.

CHAPTER 4. A GENERIC BENCHMARK FOR MINI-SPLIT HEAT PUMP SYSTEM

4.1. Overview

In this chapter two case studies which include laboratory data and manufactural data simulation for both cooling mode and heating mode are described. The five different kinds of lab data input parameters for the LG LA096HV will be discussed in section 4.2 and the manufactural data inputs are presented in section 4.3. The simulation results are described in section 4.4 and detail tables are represented in the appendix. Finally, the discussions about those simulation results for lab data and manufactural data in both cooling mode and heating mode are investigated in section 4.5.

4.2. Case study 1: HPDM inputs for the lab data

In the last chapter, the methodology of the Heat Pump Design Model has been illustrated clearly. Therefore, in this section, several lab data inputs will be examined to validate the HPDM's feasibility or effectiveness within a certain temperature range. The simulated testing unit the author simulating is the LG 096HV, which is a mini-split heat pump system with a rated cooling capacity of 9000 Btu/hr and rated heating capacity 11,700 Btu/hr.

Since there are total five kinds of the input values, researchers need to determine each one by one. The first input value should be "General system description". It is very easy to decide the first input for this unit. Cooling mode and heating mode are both applied in this LG unit and the refrigerant is the R410A. Therefore, the required first input can be figured out without any difficulties. Other four kinds of inputs are also required to be identified. However, the lab data includes the system refrigerant balancing data (the second input) which consists of superheat temperature, subcooling temperature and system refrigerant charge. In addition, system operating condition parameters (the fifth input) are also included in the laboratory data. The compressor characteristics (the third input) and the detailed geometry configuration values (the fourth input) can be found online by asking the specified manufacturers or using the methodology provided in Chapter 3 of this thesis.

The five kinds of detailed inputs for this LG unit will be displayed in the following paragraphs.

General system description:

- Both in cooling mode and heating mode
- Refrigerant R410A

System refrigerant side balancing:

- System refrigerant charge: 2.3 lbs
- Superheat temperature and subcooling temperature will be shown combined with system operation conditions

The system refrigerant side balancing inputs combined with system operating condition inputs are list one table for clarity.

System operating condition:

As discussed in Chapter 3, the contents of system operating conditions can be found without difficulties. Therefore, the detailed values for the inputs of system refrigerant side balancing and the inputs of system operating conditions will be presented in Appendix VII for the cooling mode, while these two inputs will be presented in Appendix X for the heating mode.

The static pressure is zero in this testing procedure. Also, the outer diameter of the liquid line is 1/4 in and the vapor line is 3/8 in for this LG unit. In the next section, the inputs for compressor characteristics are provided, which is the most important part to influence system performance.

Compressor characteristics:

This compressor's rated cooling capacity is 9163 Btu/hr and the rated EER for the compressor is 9.2. In addition, the rated return gas temperature is 65 F. Most important for the compressor performance simulation is the compressor map. Unfortunately, the manufacturer does not provide the compressor map for users, therefore, the scaling compressor map method should be implemented in order to obtain this specific compressor map. By utilizing the compressor map of rated cooling capacity 11,740 Btu/hr to predict the compressor map of rated cooling capacity 9173 Btu/hr, the Adjustment Factor should be specified.

$$\text{Adjustment Factor(AF)} = \frac{Predicted\ Rated\ Capacity}{Based\ Rated\ Capacity} = \frac{9173\ Btu/hr}{11740\ Btu/hr} = 0.7813$$

Multiplying by the AF, the compressor map can be specified reasonable and logical. Appendix IV shows the compressor map for this compressor. In order to generate the compressor map equations, the software Easy Equation Solver (EES) will be utilized.

EES has a linear regression function for inputting the data values. Since the compressor map representations are two 10-coefficient equations for mass flow rate and power consumption, EES can find the equations immediately and accurately. Tables 4-1 and 4-2 list the coefficients of representations for the compressor map.

Table 4-1 Power consumption coefficients

Power consumption coefficients	
C1	245.93753
C2	-6.48637635
C3	3.84440165
C4	-0.061561913
C5	0.098089265
C6	0.003499407
C7	-0.000205588
C8	0.000173891
C9	-1.77976E-05
C10	-6.06368E-06

Table 4-2 Mass flow rate coefficients

Mass flow rate coefficients	
C1	87.6772433
C2	1.63236463
C3	-0.668987207
C4	0.015336757
C5	-0.003167124
C6	0.004620506
C7	-1.82561E-05
C8	4.90809E-06
C9	-1.13937E-06
C10	-1.48437E-05

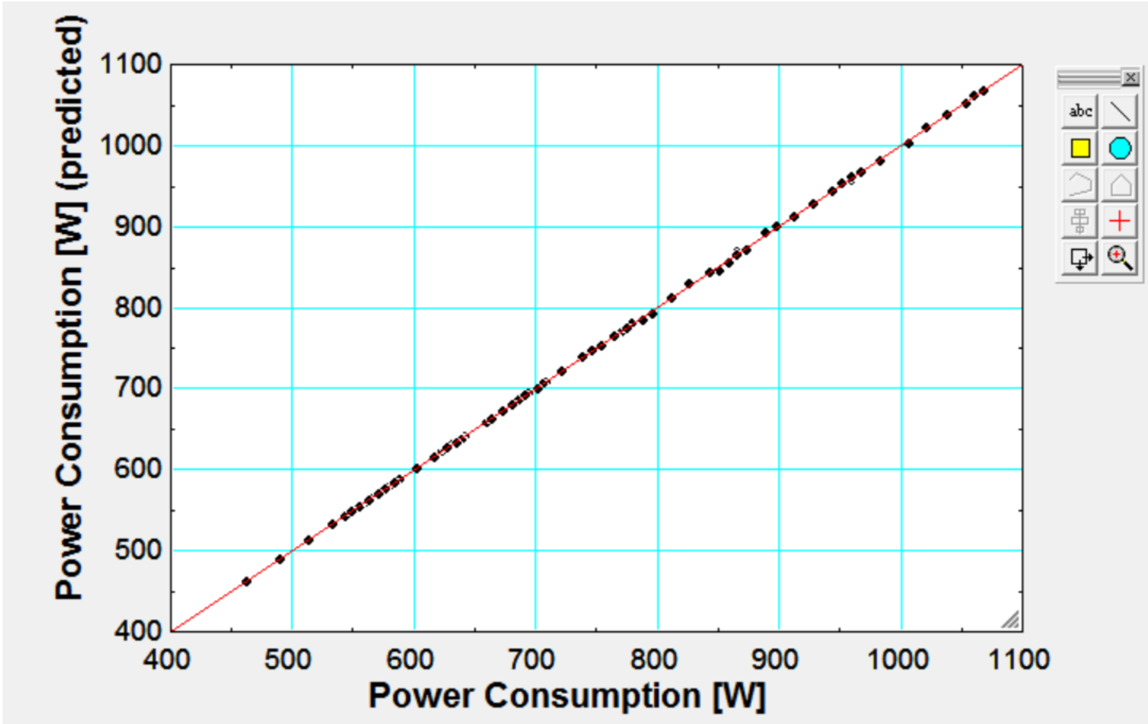

Figure 4-1 The diagram power consumption linear regression

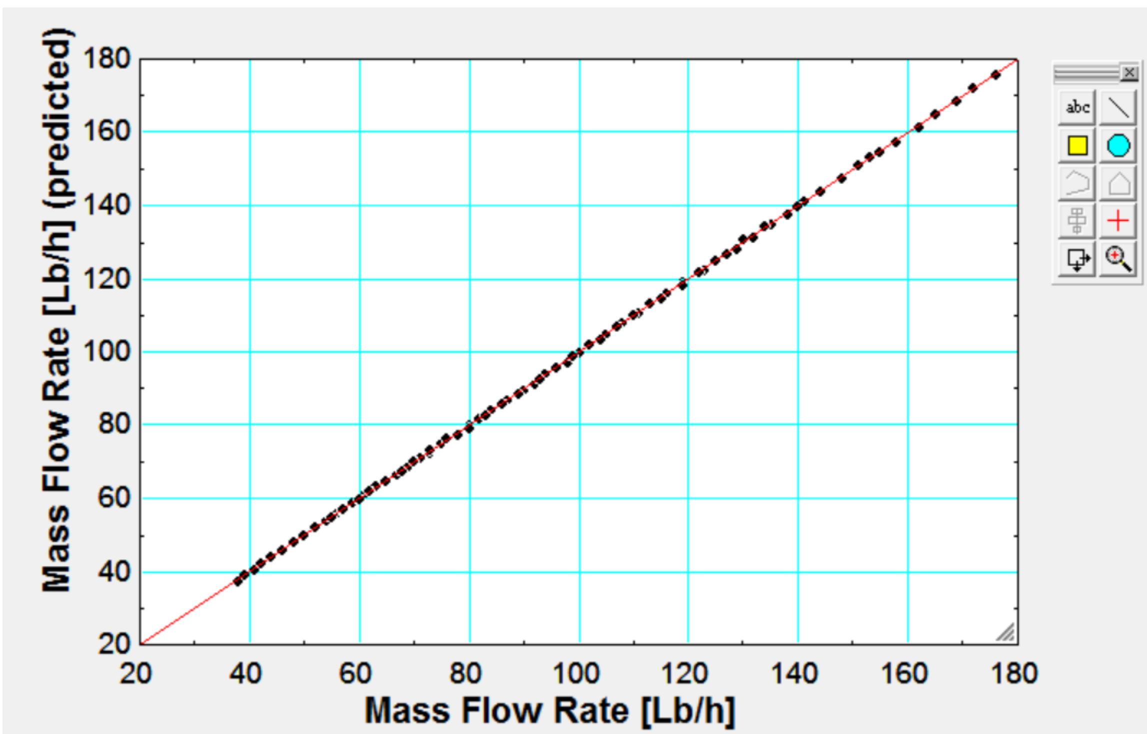

Figure 4-2 The diagram for mass flow rate linear regression

Figures 4-1 and 4-2 illustrate the degree of fit for compressor maps for mass flow rate and power consumption equations, utilizing the linear regression method in EES. Based on these two figures, the prediction compressor maps demonstrate a good fit, since all of the points are on the linear fit lines for both the compressor maps for mass flow rate and power consumption. To determine the compressor map to input, some optional parameters might be ascertained, illustrated in the following paragraph.

There are four optional parameters used to determine the Heat Pump Design Model –compressor displacement, compressor motor size, the nominal speed and the nominal voltage for the compressor. For this LG unit, the compressor displacement is 0.8 in^3/rev and the compressor motor size is 0.88 hp. In addition, the nominal speed and the nominal voltage for this compressor are 3500 rpm and 230 v, respectively.

Fin and tube heat exchanger parameters and configurations:

The most difficult information to determine is the heat exchanger geometry parameters, since manufacturers do not generally provide all of these parameters. However, if researchers can find them online or by asking professional workers, they can specify those parameters accurately. For this LG unit, thanks to Dr. Howard Cheung, most of the geometry configurations are provided by his measurement.

Table 4-3 heat exchanger parameters for the indoor unit

Indoor unit			Fin		
			Material	Al	
Tube			Type	smooth	
Material	Cu		Pitch	20	fins/in
Type	smooth		Thickness	6.05	mils
OD	0.289	in	Number of rows	2	
Wall	30	mils	Number of tubes/row(n)	15	
Frontal area	2.217	ft2	Number of equivalent, parallel circuits		
Tube spacing (a)	0.827	in	two-phase	1	
Tube spacing (b)	0.489	in	liquid phase	1	

Table 4-4 heat exchanger parameters for the outdoor unit

Outdoor unit			Fin		
			Material	Al	
Tube			Type	smooth	
Material	Cu		Pitch	17	fins/in
Type	smooth		Thickness	6.89	mils
OD	0.282	in	Number of rows	2	
Wall	30	mils	Number of tubes/row(n)	24	
Frontal area	4.83	ft2	Number of equivalent, parallel circuits		
Tube spacing (a)	0.798	in	two-phase	2	
Tube spacing (b)	0.72	in	liquid phase	1	

Tables 4-3 and 4-4 list the heat exchanger parameters for this LG indoor unit and outdoor unit, separately. The yellow region is for tube values, while orange is for fin parameters. The blue region is other parameters that should be specified for the HPDM.

4.3 Case study 2: HPDM inputs for the manufacturing data

Based on Figure 3-26, there are in total five different kinds of inputs that should be specified for the Heat Pump Design Model. Since we are utilizing the same LG indoor and outdoor unit, the inputs of general system descriptions, compressor characteristics and fin-and-tube heat exchanger parameters and configurations are the same as the lab data inputs. The only differences for the manufactural data inputs are system refrigerant side balancing, which should specify superheat temperature and subcooling temperature used in the system, and the system operating conditions, which should include indoor and outdoor dry-bulb temperature and wet-bulb temperature and air flow rates. Typically, the manufacturers set the superheat temperature at a range of 8-12 F, while the subcooling temperature range within a certain amount is 10-15 F. Those temperature inputs for system operating conditions are shown in Tables 2-1 and 2-2 in section 2.3. The air flow rates are 371 cfm for the indoor unit and 954 cfm for the outdoor unit.

4.4 Case study 1&2 results for laboratory and manufactural data by using HPDM

After required values are inputted into the Heat Pump Design Model, users can obtain the simulation results and system performance data. In order to validate the test for the HPDM in the mini-split heat pump system, performance output values are required to be obtained for comparison. Therefore, the simulation results will include system cooling capacity and heating capacity, EER, SHR, mass flow rate and power consumption for both the cooling and heating modes. Appendix IX shows the simulation lab data results for cooling mode and Appendix XII shows the simulation laboratory data results for heating mode.

There are a total of 19 data sets for the laboratory testing data. Most of the simulation procedures are reliable without system scaling in the cooling mode, but system scaling was used when the outdoor dry-bulb temperature was 67 F because the HPDM did not get a convergent result and showed a system internal error. As for the heating mode, in order to get more accurate results, the scaling system for all of the 27 combinations of temperature conditions was applied and the results are presented in Appendix XII. The HPDM manufactural data simulation results are presented in Appendix XIII and Appendix XIV.

Since the manufacturer only provides some outputs related to the system capacity and power consumption, the focus will be for outputs based more on these two performance parameters. Therefore, two kinds of performance data are specified for this LG unit regarding the simulation results of manufactural data within different temperature inputs – system cooling or heating capacity and system power consumption.

After obtaining the simulation results for laboratory data and manufactural data for both cooling mode and heating mode, detailed discussions are provided in the next section.

4.5 Case studies discussions for simulation results for lab data and manufactural data

In this section, the results of two case studies are compared. The first case study is the laboratory data comparison for cooling mode and heating mode. The second case study is the manufactural data comparison for cooling mode and heating mode. The Heat Pump Design Model cannot do all the time range speculations because there are large relative errors for some temperature ranges, but the HPDM can be a generic benchmark for most of the temperature conditions.

4.5.1 Case study 1: lab performance data comparison

In this case study, the cooling mode and the heating mode results have already been provided to compare with the lab performance data which has been considered in the last section. Cooling capacity, EER, SHR, mass flow rate and power consumption are compared in Table 4-5.

Based on shown values of Table 4-5, the relative errors for capacities are in the reasonable range and the absolute maximum relative error is 8.62%. However, the scaling system capacity option was applied in the Heat Pump Design Model when simulating the outdoor dry-bulb of 67 F, since the HPDM cannot converge at this temperature which is typically too low for cooling mode. In the scaling system option, the cooling capacity was set, as the lab data provided, within the capacity of 50 Btu/hr when the outdoor dry bulb temperature was 67 F. The simulation results of SHR are very close to the actual values and also for the mass flow rates, since the relative errors for these two parameters are relatively small enough. The largest absolute relative error value is 5.11% for the SHR, while 5.39% is the largest absolute relative error value for mass flow rate. The power consumption prediction is considerably good in most of the temperature ranges except for the outdoor dry-bulb temperature of 67 F, where relative errors for these are all over 10%. Also, because of Equation 4.1, the EER is the ratio of the capacity and power consumption. Therefore, the relative errors of the EER are much larger when the T_{odb} is 65 F. The reason for this situation is the temperature range of the compressor map, since the temperature point of 65 F for the outdoor dry-bulb temperature is out of prediction range. The compressor map is trustworthy for interpolation and defective for extrapolation, thus the power consumption prediction is not accurate when the T_{odb} is 65 F. Among those inaccurate simulation results for EER, only one point of EER is not accurate, and that is when the outdoor dry-bulb temperature is 105 F. Here, the relative error is 48.17%. This is because of data mistakenly recorded by the program. When the relative errors are more than 10%, they are represented in red font.

$$\text{EER} = \frac{Capacity\ (Btu/hr)}{Power\ (Watts)} \tag{4.1}$$

After Table 4-5 showing the relative errors for lab data outputs in cooling mode, five figures are presented as follows. These figures are a graphic representation for the relative errors of outputs for cooling capacity, EER, SHR, mass flow rate and power consumption. In addition, the system charges and discharge pressures for every point of the cooling mode are both in the reasonable range, while the suction pressures are in the acceptable ranges except T $_{odb}$ is 65 F, shown in Table 4-6.

Table 4-5 the relative errors for lab data outputs in cooling mode

Intended Outdoor Dry-bulb	Intended Indoor Dry-bulb	Intended Indoor Wet-bulb	Capacity	EER	Sensible Heat Ratio	Mass Flow rate	Power
[F]	[F]	[F]	Btu/hr	[Btu/W-h]		[lbm/hr]	W
87	74	62	5.35%	-0.68%	-2.58%	0.76%	3.40%
95	80	67	8.62%	3.60%	-3.65%	3.77%	2.44%
105	80	67	5.92%	5.00%	-3.91%	0.10%	-0.95%
105	80	56	1.28%	1.90%	-3.90%	-5.39%	-1.17%
105	80	56	2.18%	-0.36%	-3.20%	-3.73%	-2.12%
95	74	66	5.66%	-0.50%	-0.63%	3.37%	2.32%
87	74	66	6.21%	-3.61%	-0.20%	3.10%	4.62%
115	80	67	7.06%	-3.40%	-6.60%	-3.05%	-1.70%
87	74	66	6.18%	-5.90%	0.40%	3.57%	3.93%
95	74	66	7.98%	-5.17%	-0.50%	5.76%	2.18%
105	80	67	4.20%	-0.20%	-1.13%	-1.25%	-1.34%
105	80	67	2.58%	0.54%	-0.45%	-3.68%	-1.23%
67	80	67	-0.09%	27.30%	-1.20%	-0.78%	-19.01%
67	74	62	-0.07%	128.90%	-2.70%	-1.58%	-11.38%
105	74	62	3.18%	-48.17%	-3.90%	-5.82%	-0.13%
67	80	56	0.52%	14.77%	0.00%	-1.78%	-10.35%
67	80	67	-1.22%	-22.53%	0.10%	-4.30%	26.18%
67	74	66	-0.01%	4.64%	1.10%	-1.01%	-11.19%
67	74	66	-0.55%	13.45%	-0.50%	-0.46%	-13.29%

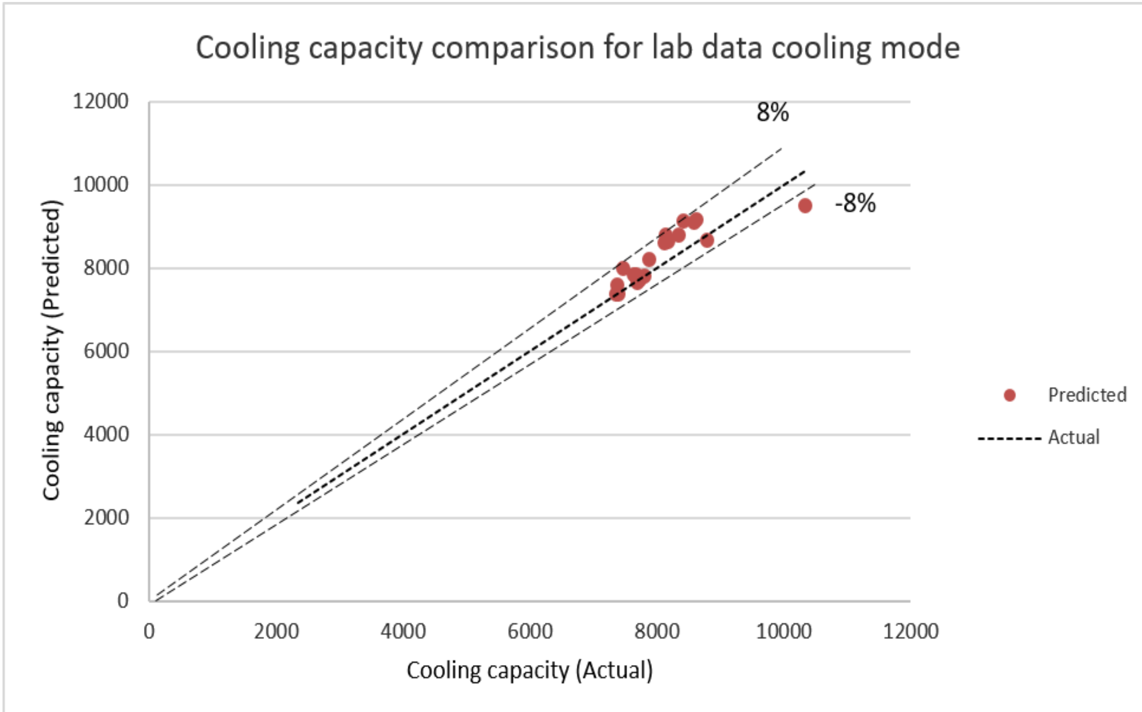

Figure 4-3 Cooling capacity comparison for lab data cooling mode

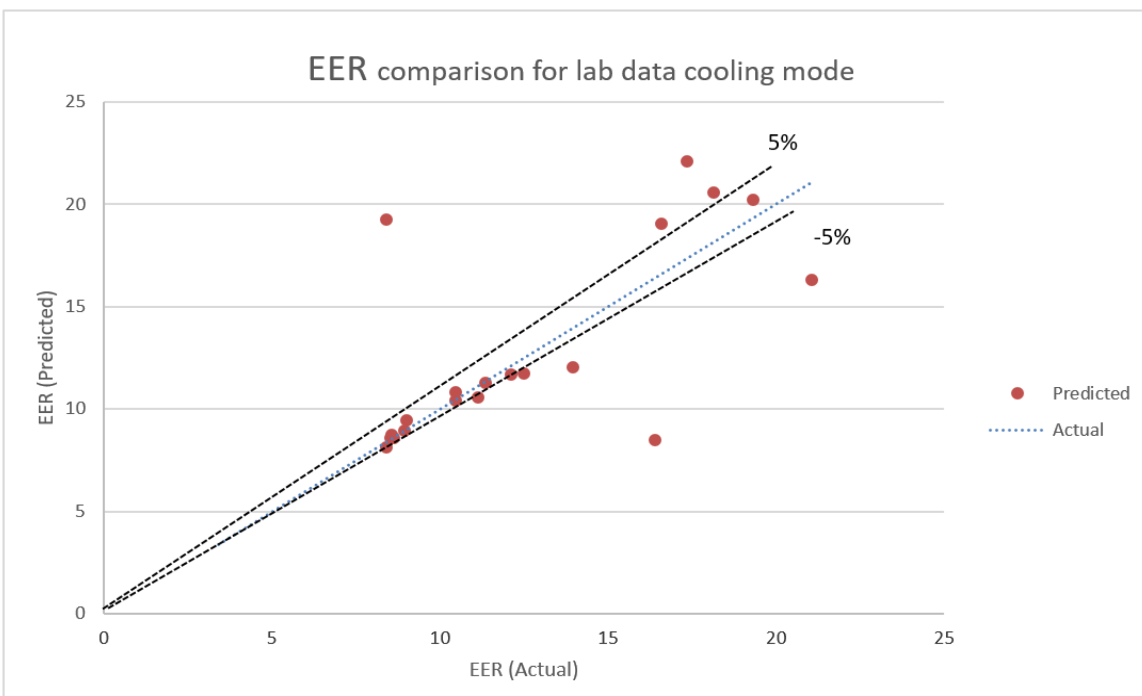

Figure 4-4 EER comparison for lab data cooling mode

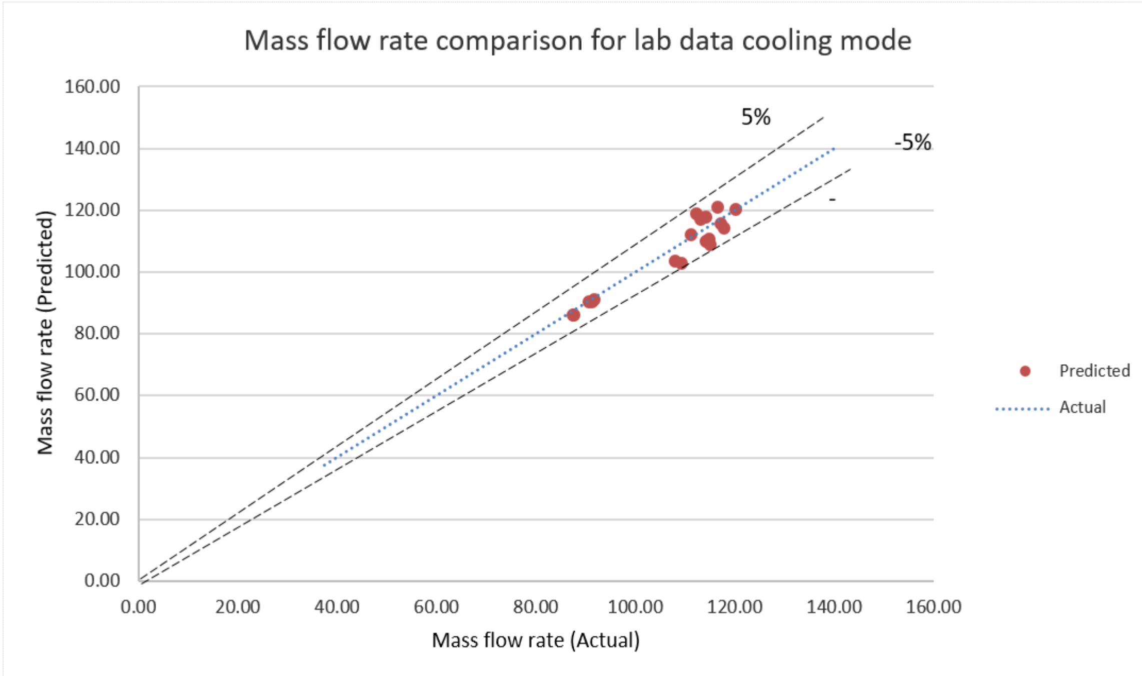

Figure 4-5 Mass flow rate comparison for lab data cooling mode

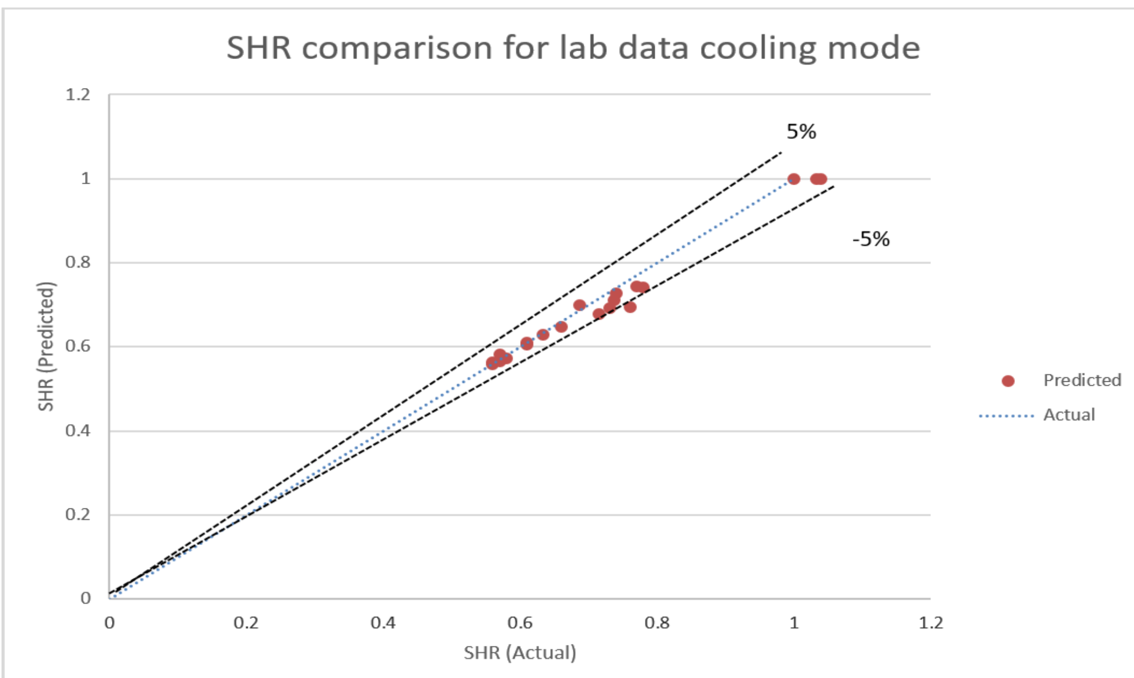

Figure 4-6 SHR comparison for lab data cooling mode

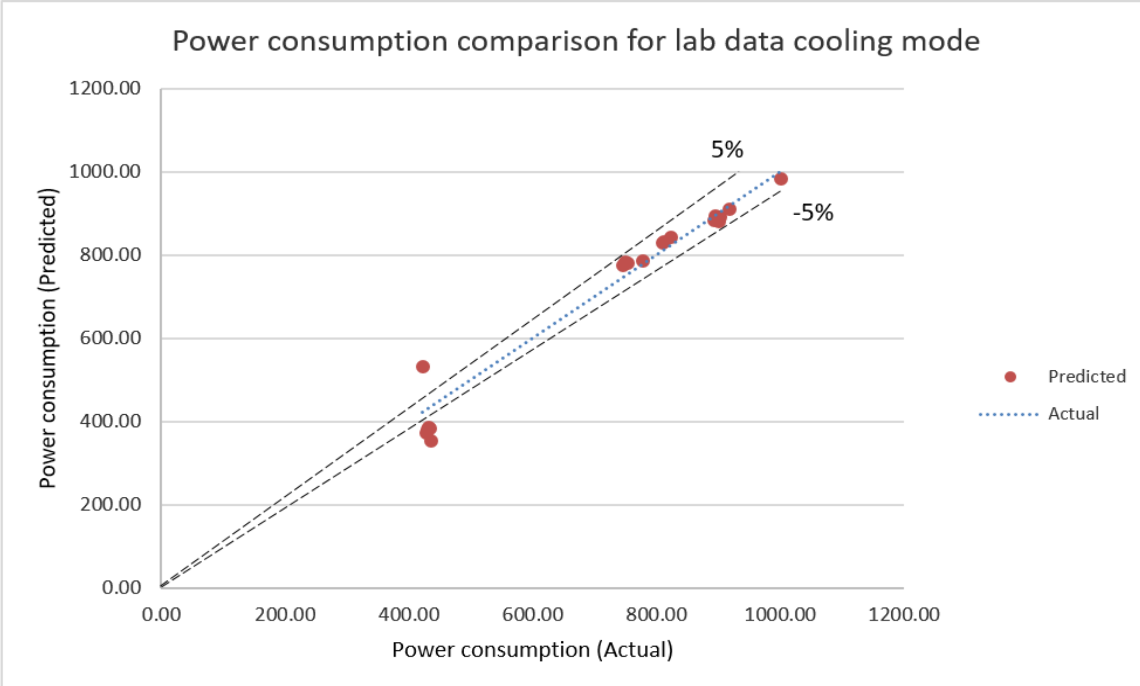

Figure 4-7 Power consumption comparison for lab data cooling mode

Figure 4-3 shows the cooling capacity comparison for lab data in the cooling mode, while Figure 4-4 presents the EER comparison. In addition, the Mass flow rate and SHR relative errors are illustrated in Figure 4-5 and 4-6, respectively. Finally, Figure 4-7 shows the power consumption comparison for the lab data in the cooling mode. Based on the representatives of these five figures, only when the outdoor temperature is 65 F, the Heat Pump Design Model did not obtain good simulation results, which are shown outside of the limitation line in these figures.

Table 4-6 Compressor suction, discharge pressure, and system charge

Intended Outdoor Dry-bulb	Intended Indoor Dry-bulb	Intended Indoor Wet-bulb	suction pressure	discharge pressure	system charge
87	74	62	2.61%	-2.56%	-4.07%
95	80	67	5.70%	-1.37%	-1.86%
105	80	67	3.07%	-0.25%	-2.77%
105	80	56	-2.55%	-0.81%	-6.32%
105	80	56	-1.62%	-0.67%	-6.10%
95	74	66	5.76%	-1.34%	-2.68%
87	74	66	4.86%	-1.95%	-2.29%
115	80	67	3.21%	2.31%	2.16%
87	74	66	5.17%	-1.90%	-2.03%
95	74	66	5.76%	-1.34%	-2.68%
105	80	67	0.68%	-0.50%	-4.20%
105	80	67	-1.15%	-0.62%	-5.37%
67	80	67	7.24%	0.02%	-4.24%
67	74	62	-1.54%	-0.68%	-7.19%
105	74	62	11.46%	1.12%	-2.77%
67	80	56	-6.06%	-1.05%	-8.14%
67	80	67	-19.05%	-4.38%	-8.27%
67	74	66	21.68%	-5.35%	4.85%
67	74	66	20.83%	-4.62%	4.55%

Analysis of the cooling mode for lab data, the comparison of the laboratory data in the heating mode is provided in the next step. The scaling system method was utilized for all twenty-seven situations since the Heat Pump Design Model cannot create good results for the heating capacity simulation. The relative error results for the lab data outputs for the heating mode are shown in Table 4-7. Additionally, compressor suction, discharge pressure, and system charge are all in Table 4-8.

Table 4-7 the relative error for lab data outputs in heating mode

Intended Indoor Dry-bulb	Indoor coil Inlet air Wet-bulb	Intended Outdoor Dry-bulb	Intended Outdoor Wet-bulb	capacity	cop	mass flow rate	power
[F]	[F]	[F]	[F]	Btu/hr		lbm/hr	w
69.76	53.83	46.96	38.51	-0.05%	-1.13%	10.39%	4.08%
63.93	50.88	62.17	49.63	0.88%	-0.25%	5.22%	4.32%
76.03	58.15	62.02	49.26	-1.30%	3.97%	2.10%	-2.54%
64.04	51.31	68.07	53.08	-0.04%	4.36%	-1.02%	-1.61%
70.00	53.87	41.91	34.80	0.13%	-0.77%	8.21%	3.29%
69.89	54.97	42.00	37.14	-0.35%	-0.63%	7.65%	2.52%
63.93	51.98	34.09	29.23	-0.94%	-1.02%	1.69%	2.12%
75.94	58.91	35.04	30.52	0.78%	4.63%	8.07%	-0.43%
70.00	54.99	61.90	51.21	0.91%	0.50%	10.38%	3.88%
63.93	48.03	61.88	46.59	1.46%	0.91%	2.79%	4.45%
76.08	53.91	61.97	47.85	-1.37%	6.51%	-1.25%	-3.51%
76.17	55.49	67.78	51.39	-0.24%	8.60%	-1.76%	-4.07%
69.93	52.56	67.91	51.22	-0.77%	10.85%	-4.77%	-6.10%
70.03	53.38	62.04	48.37	-0.11%	5.64%	-1.55%	10.31%
69.98	51.71	46.92	37.42	-1.14%	-1.99%	8.83%	6.16%
70.03	51.75	47.03	37.42	-1.04%	6.23%	6.76%	-0.45%
70.18	53.24	41.97	40.50	-0.74%	-1.66%	7.75%	4.75%
70.07	54.90	34.99	30.28	-0.57%	0.46%	6.26%	4.54%
69.91	51.31	26.92	23.10	-0.83%	-0.27%	3.91%	2.88%
70.00	49.62	17.05	15.57	-0.09%	4.87%	-2.43%	0.04%
69.94	49.43	7.03	6.33	-0.60%	8.02%	-9.10%	-2.61%
69.94	49.80	34.93	27.69	-1.56%	0.32%	5.58%	4.53%
69.96	49.95	-2.92	-2.97	-1.86%	24.07%	-27.54%	-15.25%
69.98	51.64	46.84	39.09	-1.67%	6.51%	3.82%	-5.27%
70.05	51.62	34.98	29.89	-1.11%	4.67%	3.88%	-2.02%

Table 4-8 Compressor suction, discharge pressure, and system charge

Intended Indoor Dry-bulb [F]	Intended Indoor Wet-bulb [F]	Intended Outdoor Dry-bulb [F]	Intended Outdoor Wet-bulb F	suction pressure	discharge pressure	system charge
69.76	53.834	47.0	38.5088	-3.82%	-0.94%	-5.43%
63.93	50.882	62.2	49.6256	-2.04%	0.28%	0.00%
76.03	58.154	62.0	49.262	-2.19%	-0.44%	-0.35%
64.04	51.314	68.1	53.078	1.85%	0.17%	-0.04%
70.00	53.87	41.9	34.8008	-5.70%	-0.68%	-4.35%
69.89	54.968	42.0	37.139	-8.97%	-1.48%	-10.83%
63.93	51.98	34.1	29.2262	-13.60%	-1.50%	-16.87%
75.94	58.91	35.0	30.5163	-5.54%	-1.63%	-14.17%
70.00	54.986	61.9	51.206	-0.87%	0.38%	-1.35%
63.93	48.0254	61.9	46.5872	-1.97%	-0.89%	-0.22%
76.08	53.906	62.0	47.8454	-1.90%	-0.97%	0.00%
76.17	55.49	67.8	51.386	2.94%	-2.57%	0.00%
69.93	52.556	67.9	51.224	4.59%	-2.39%	0.00%
70.03	53.384	62.0	48.3692	-3.30%	3.11%	0.00%
69.91	54.032	62.0	48.6608	10.53%	-4.51%	0.00%
69.98	51.71	46.9	37.4234	-5.51%	-1.15%	0.00%
70.03	51.746	47.0	37.4162	-5.39%	-2.50%	-7.22%
70.18	53.24	42.0	40.496	-6.18%	-2.24%	-8.70%
70.07	54.896	35.0	30.2833	-8.92%	-3.06%	-13.04%
69.91	51.314	26.9	23.1044	-14.27%	-3.89%	-8.70%
70.05	51.026	27.1	23.9162	-14.40%	6.91%	-13.04%
70.00	49.6166	17.0	15.5696	-11.90%	-11.71%	-17.39%
69.94	49.4312	7.0	6.332	-14.23%	-11.39%	-17.39%
69.94	49.8038	34.9	27.6872	-8.01%	-2.43%	-11.57%
69.98	51.638	46.8	39.092	-3.17%	-1.99%	-4.35%
70.05	51.62	35.0	29.894	-6.70%	-3.84%	-4.35%

However, achieving this option in the HPDM, the simulation results for heating capacity are all very similar to the actual ones and the relative errors for the heating capacities are all in a reasonable range and they are all around 1% relative errors. There is a data set where the prediction was not very accurate when the outdoor dry-bulb and wet-bulb temperature is around -3 F. The HPDM cannot get good simulation results in the heating mode when the outdoor temperature is very low. The simulation results include the mass flow rate, power consumption, and COP prediction. The main reason for the inaccurate simulation results is the limitation for a temperature range of the compressor map which is directly related to the mass flow rate and power consumption. In addition, the COP is the ratio of the heating capacity and the power consumption in the heating mode, which is illustrated in Equation 4.2. If any of these two parameters is inaccurate, the COP prediction will also be an inaccurate one. Therefore, the HPDM cannot get a good simulation result when the outdoor dry-bulb and wet-bulb temperature is around -3 F. The Heat Pump Design Model is demonstrated to be good results for the simulating heating mode in other temperature ranges for the high compressor speed. Since there are limited data sets for low compressor speed and medium compressor speed, this study focuses more on the high compressor speed data values.

$$\mathrm{COP} = \frac{Capacity\ (Watts)}{Power\ (Watts)} \tag{4.2}$$

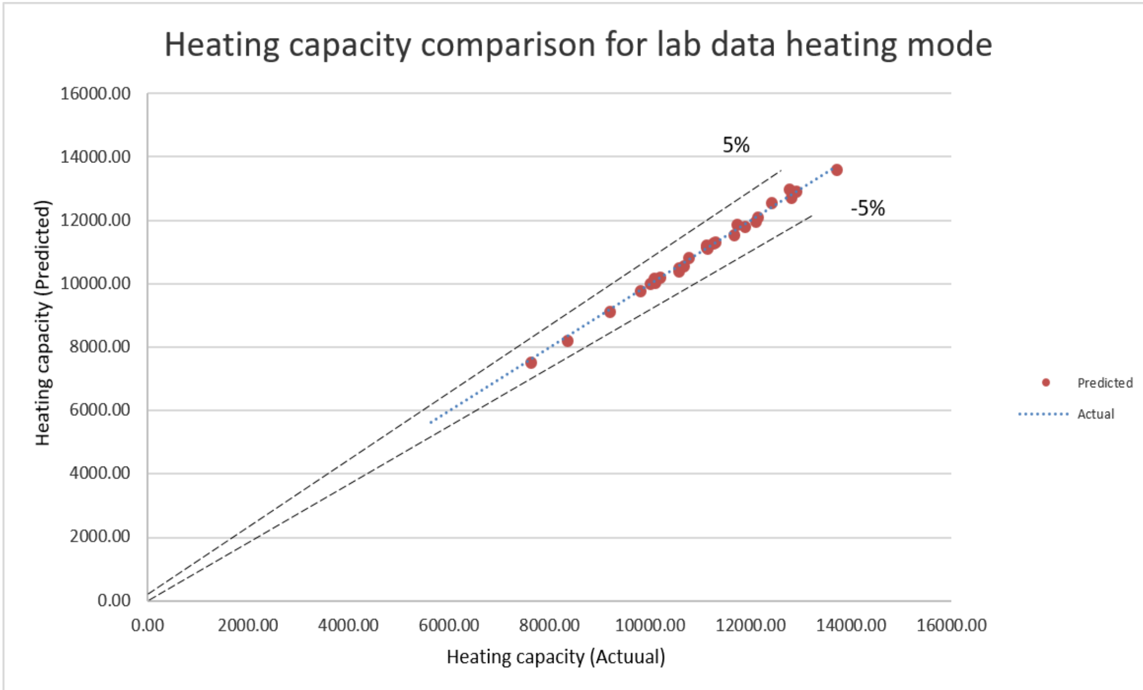

Figure 4-8 Heating capacity comparison for lab data heating mode

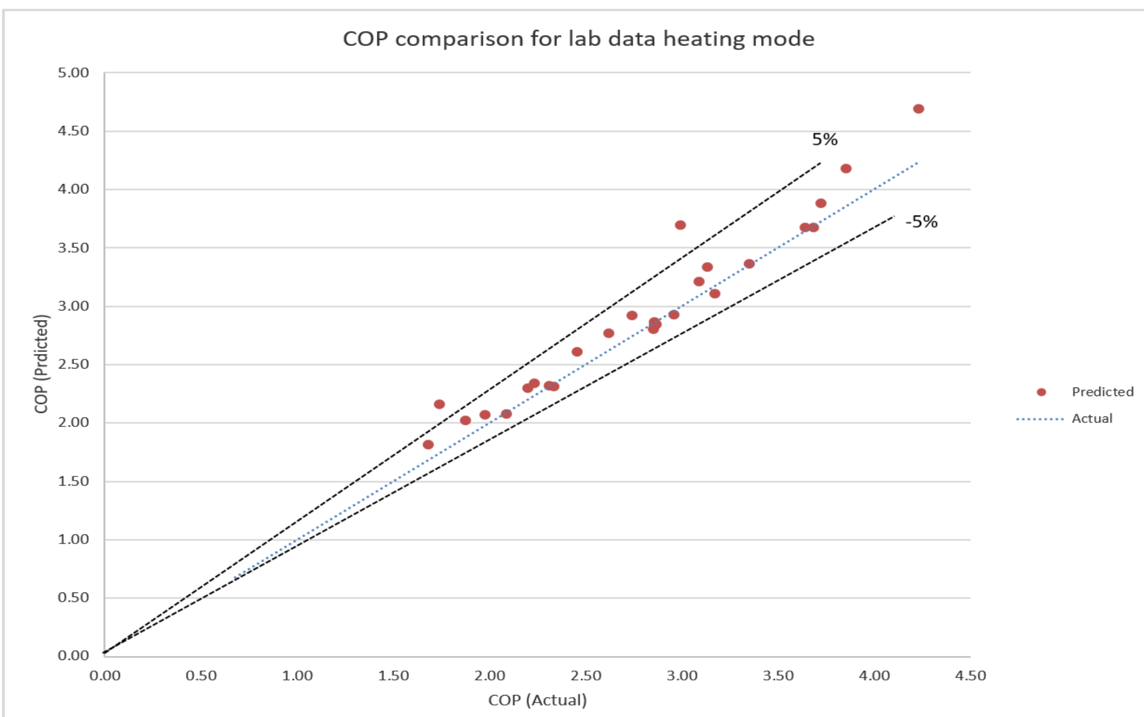

Figure 4-9 COP comparison for lab data heating mode

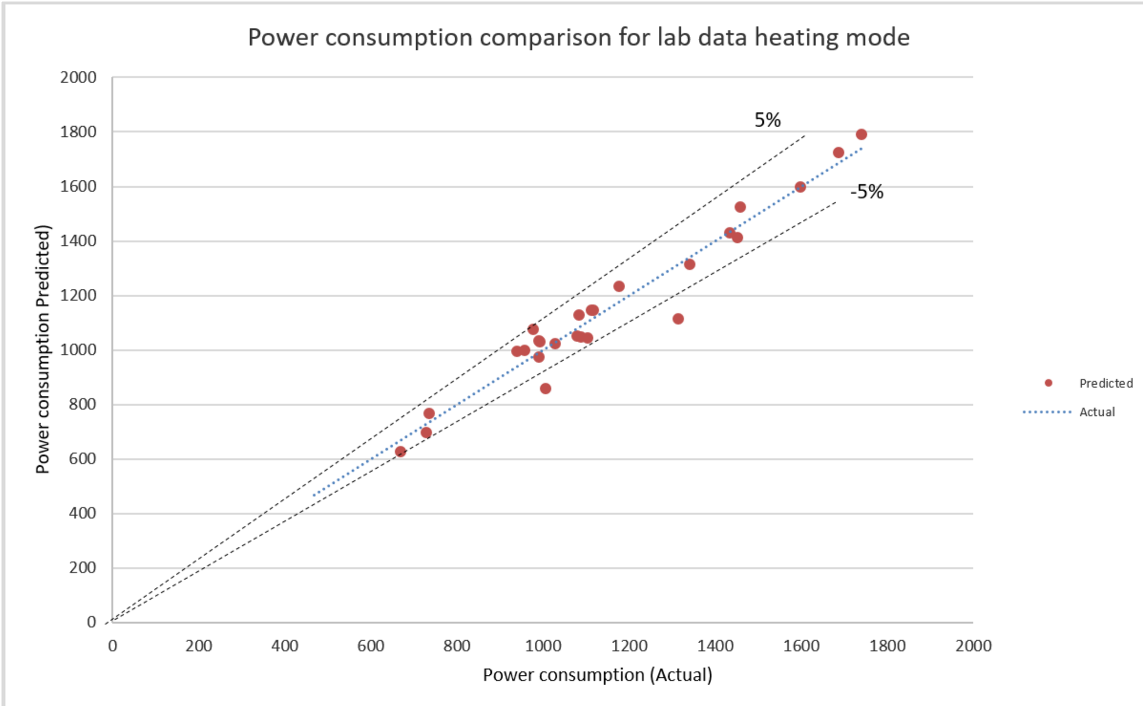

Figure 4-10 Power consumption comparison for lab data heating mode

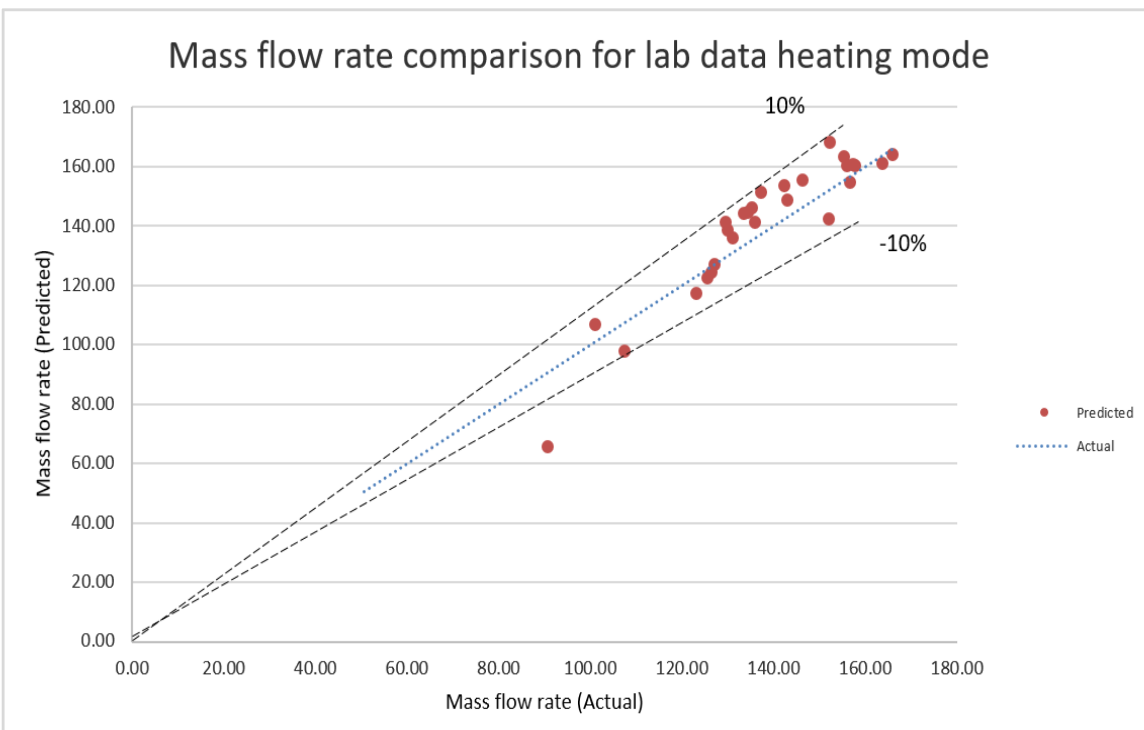

Figure 4-11 Mass flow rate comparison for lab data heating mode

There are four figures shown in 4-8, 4-9, 4-10 and 4-11, for the capacity comparison, COP comparison, mass flow rate comparison and the power consumption comparison in the heating mode simulation, respectively. They are the illustrations for these four key outputs. Readers can determine deviation points for the heating mode lab data simulation directly by looking at these figures.

4.5.2 Case study 2: manufactural performance data comparison

Comparisons between manufactural performance data and the simulation outputs for the Heat Pump Design Model are also provided. In the beginning, the comparison will concentrate on the cooling mode. In the next step, the heating mode comparison will be offered in this section, too.

First, the cooling mode comparison is provided for the first step. The cooling capacity and power consumption comparisons are the major topic to be discussed.

Table 4-9 Capacity relative error for manufacture data in the cooling mode

Outdoor	Capacity relative error					
dry-bulb	Indoor dry-bulb temperature (F)/ wet-bulb temperature (F)					
temperature (F)	68/57.2	71.6/60.8	77/64.4	80.6/66.2	86/71.6	89.6/75.2
68	0.50%	0.45%	0.04%	0.07%	-0.99%	-2.96%
77	-0.12%	0.40%	0.38%	0.44%	-0.01%	-1.60%
89.6	-0.97%	-1.16%	-0.64%	-0.53%	-1.09%	-1.89%
95	-1.60%	-2.15%	-1.84%	-0.69%	-1.40%	-2.26%
104	-3.26%	-1.78%	-4.51%	-4.28%	-3.66%	-4.07%
109.4	-4.35%	-5.26%	-6.08%	-6.46%	-5.67%	-5.98%
114.8	-5.55%	-6.73%	-7.94%	-8.13%	-7.99%	-7.98%

Table 4-10 Power consumption error for manufacture data in the cooling mode

Outdoor dry-bulb temperature (F)	Power consumption relative error Indoor dry-bulb temperature (F)/ wet-bulb temperature (F)					
	68/57.2	71.6/60.8	77/64.4	80.6/66.2	86/71.6	89.6/75.2
68	40.5%	3.2%	-4.3%	-6.1%	-6.4%	-6.5%
77	45.9%	10.9%	3.2%	1.3%	-0.6%	-0.8%
89.6	22.2%	3.7%	-0.5%	-1.9%	-3.2%	-5.9%
95	15.5%	1.3%	-1.3%	0.2%	-3.5%	-6.0%
104	11.4%	2.4%	2.3%	2.6%	1.6%	-0.8%
109.4	18.2%	14.0%	14.9%	15.3%	14.4%	12.9%
114.8	38.1%	37.0%	40.5%	43.6%	44.6%	42.5%

Base on Table 4-9, the relative errors for cooling capacity are relatively small. The maximum absolute relative error is 8.13% and another relative error is smaller than this value. Illustrated on Table 4-10, the power consumption prediction for the manufactural data describes good agreement in a certain temperature range, otherwise, the power consumption does not have great simulation results. The optimal temperature ranges were from 71.6 F to 89.6 F for the indoor dry-bulb temperature, 60.8 F to 75.2 F for the indoor wet-bulb temperature, and 68 F to 104 F for the outdoor temperature. The HPDM will give bad power consumption simulation results in other temperature conditions, which are marked by the red font in Table 4-10. As discussed in the last case study, the reason for this issue is the limited temperature range for the compressor map. In addition, figures 4-12 and 4-13 present the relative error bounds for the cooling capacity and the power consumption for manufacturer data in the cooling mode within the different temperature range inputs.

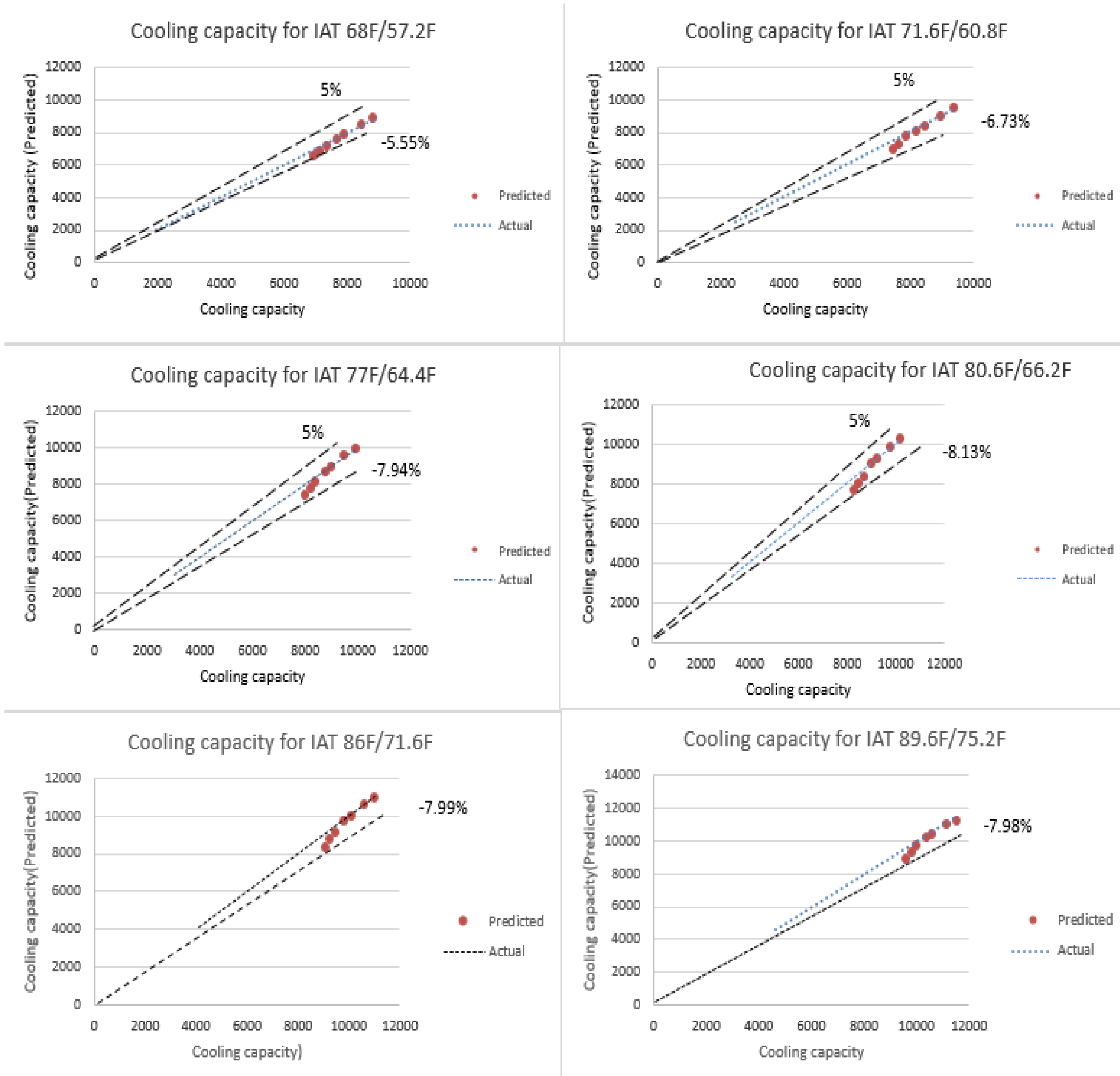

Figure 4-12 Cooling capacity comparison for manufactural data in the cooling mode

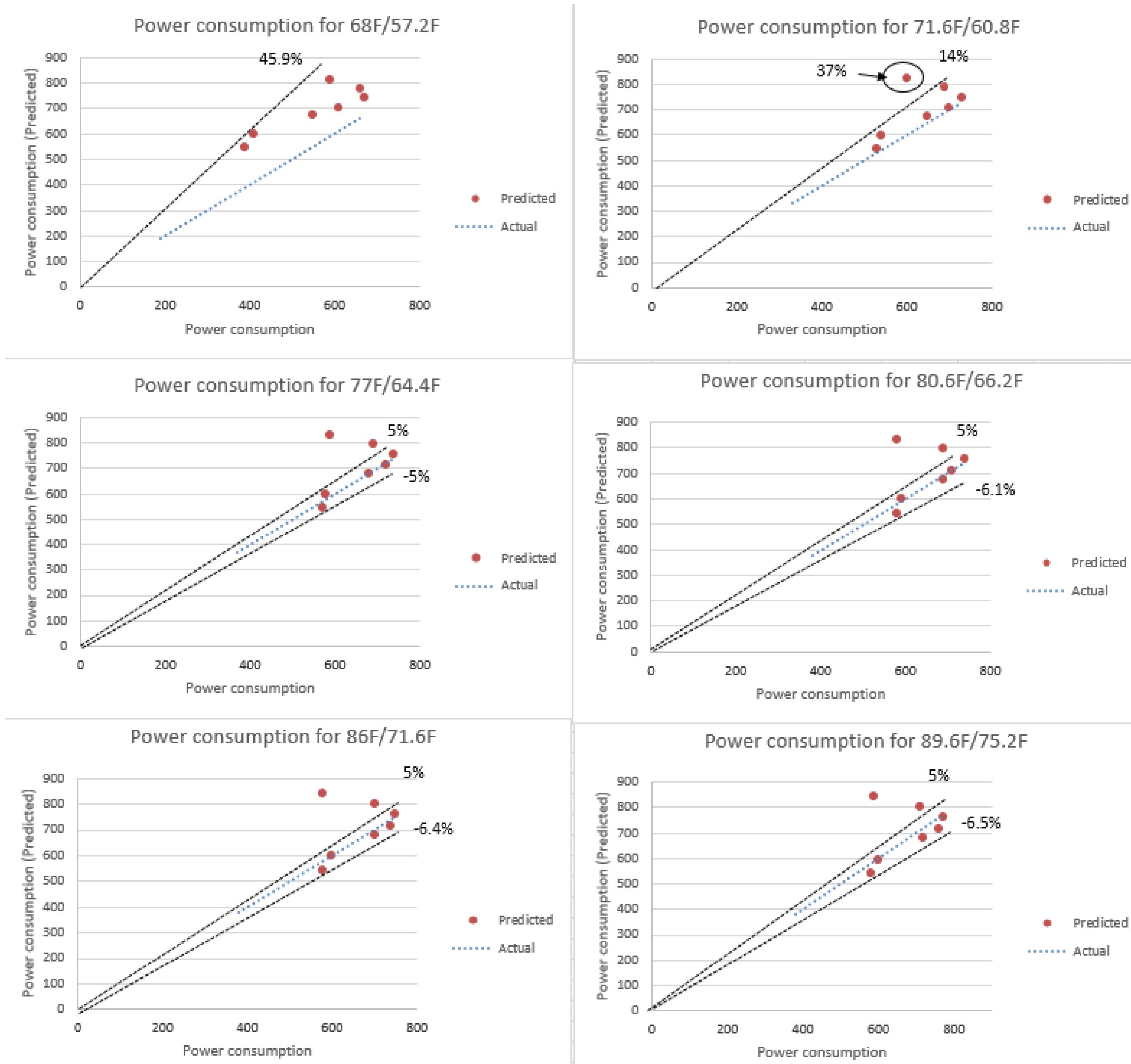

Figure 4-13 Power consumption comparison for manufactural data in the cooling mode

The heating mode comparisons are provided as below. According to Table 4-11, the relative errors for the heating capacity is pretty small, while the power consumption prediction can be large within a certain temperature range, except for the condition of outdoor wet bulb temperatures being 5 F, 14 F, and 59 F. The reason for heating capacity simulating well is that the scaling system capacity option is also utilized for all of the heating modes in the manufactural data conditions. Simultaneously, the power consumption prediction can work well when the outdoor wet-bulb temperature is from 23 F to 50 F and the indoor dry-bulb temperature is between 60.8 F and 75.2 F. Figures 4-14 and 4-15 show the heating capacity and power consumption error bounds for different temperature conditions which are illustrated in the Table 4-11.

Table 4-11 Capacity and power consumption relative error for manufacture data in the heating mode

Heating Capacity relative error							
Indoor unit	outdoor unit wet bulb temperature (F)						
dry bulb	5	14	23	32	42.8	50	59
60.8	0.3%	0.3%	-0.5%	-3.0%	0.7%	0.1%	0.0%
64.4	0.5%	0.3%	0.2%	-3.5%	-0.7%	0.1%	-0.1%
68	0.5%	0.0%	-0.8%	-3.8%	0.0%	-0.9%	-0.2%
69.8	0.6%	-0.1%	0.3%	-3.7%	0.0%	-0.8%	-0.1%
71.6	0.5%	0.0%	0.5%	-4.1%	-0.5%	-0.8%	0.1%
75.2	1.7%	0.5%	0.7%	-3.7%	-0.2%	0.0%	-0.3%
Power consumption relative error							
Indoor unit	outdoor unit wet bulb temperature (F)						
dry bulb (F)	5	14	23	32	42.8	50	59
60.8	-24.7%	-16.5%	-11.5%	-1.7%	4.1%	7.3%	16.6%
64.4	-23.2%	-14.6%	-7.6%	-2.6%	2.1%	8.1%	18.8%
68	-20.9%	-13.0%	-6.7%	-2.3%	5.7%	7.6%	22.8%
69.8	-19.7%	-10.8%	-3.2%	-2.1%	6.0%	9.6%	24.1%
71.6	-18.8%	-10.2%	-2.6%	-1.9%	5.2%	10.4%	24.3%
75.2	-19.0%	-9.6%	-2.1%	-1.6%	7.1%	10.3%	24.5%

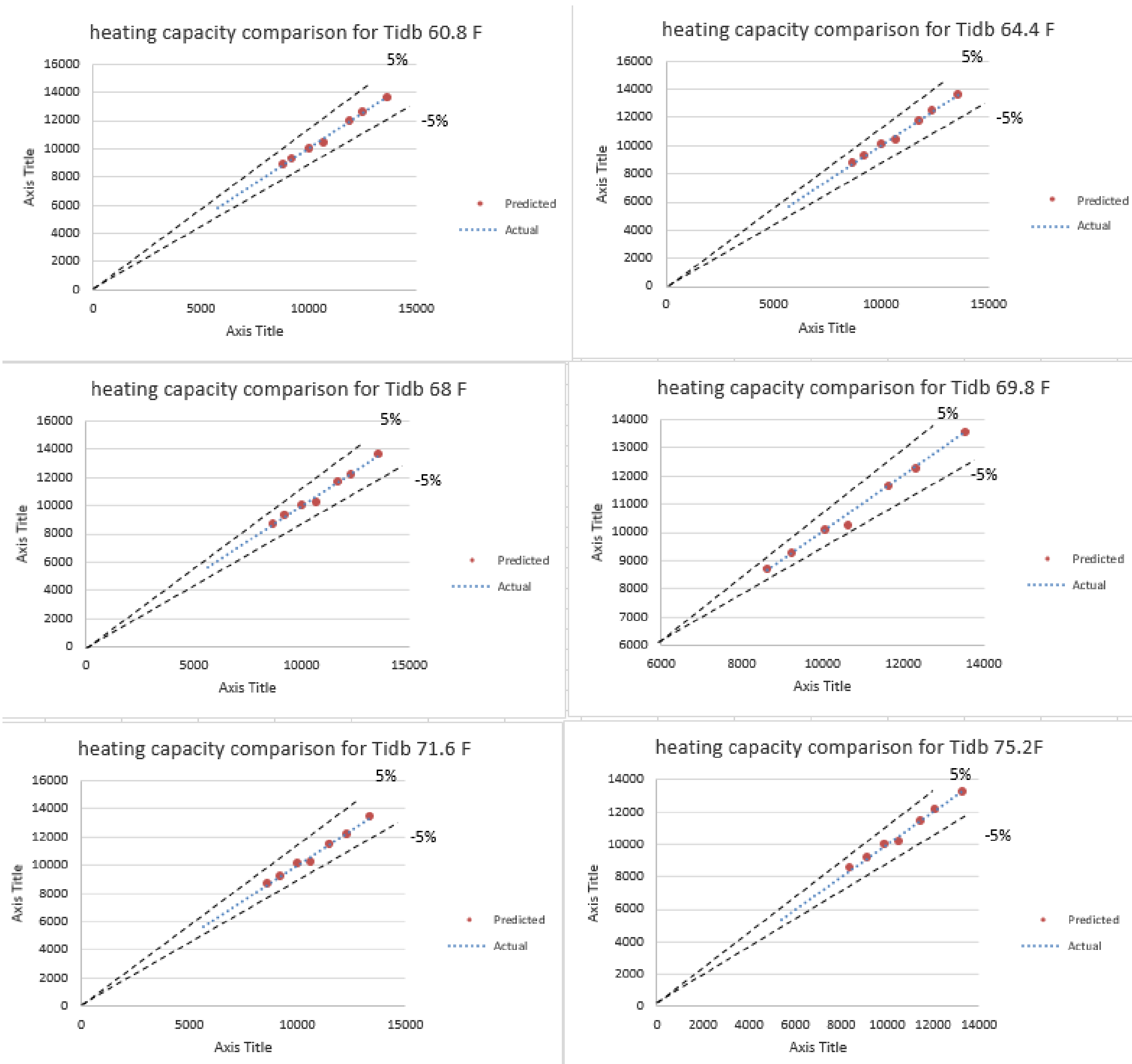

Figure 4-14 Heating capacity comparison for manufactural data in the heating mode

The error bounds of the heating capacity comparisons are 5% for each temperature situation. The horizontal axis represents actual heating capacity, while the vertical axis is the predicted heating capacity. Every simulation result is within the 5% error bound for heating capacity simulation.

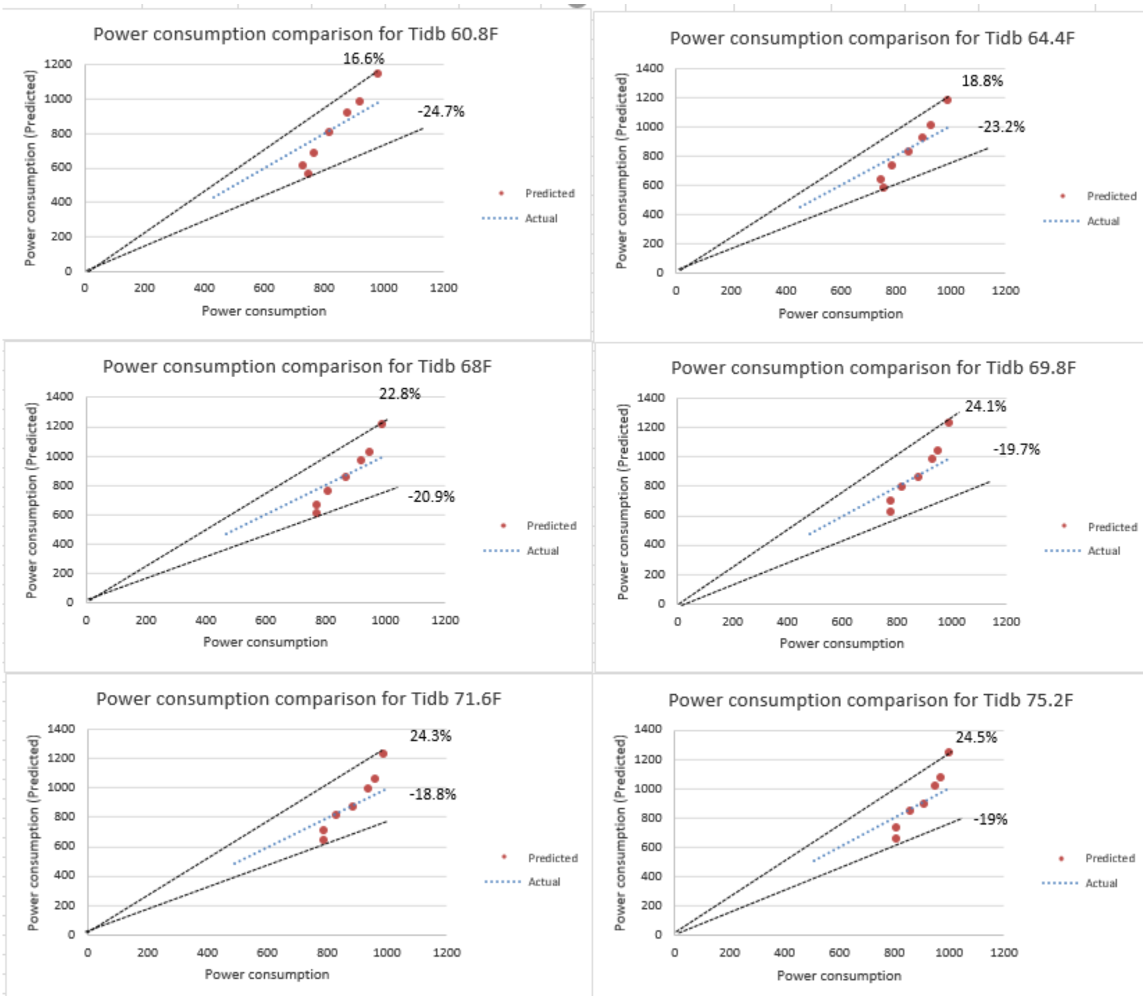

Figure 4-15 Power consumption comparison for manufactural data in the heating mode

In the meantime, the power consumption prediction is not as good as the heating capacity prediction. The points for the outdoor wet-bulb temperature being 5 F, 14 F, and 59 F are out of the error bounds, which are shown in Figure 4-15.

CHAPTER 5. CONCLUSIONS AND FUTURE WORK

The Heat Pump Design Model is a useful web-based and free of charge software. In theory, a physical model is the core of the HPDM, thus it can provide accurate simulation results for both cooling and heating modes. In practice, there are several large projects utilized in the split system funded by the U.S Department of Energy. In addition, there is limited research about mini-split heat pump system. However, the physical model requires complex input parameters and sometimes these parameters are not easily to obtained. When users can figure out these input parameters, the HPDM can be a good model. Therefore, it is possible to investigate the HPDM in order to generalize it for a mini-split heat pump system.

There are two kinds of data sources offered in this study: the laboratory data and the manufactural data. Further, these two data sets are provided for both cooling mode and heating mode. The lab data is provided by studies from the Herrick lab at Purdue University.

The methodology of the HPDM should be acknowledged by the user. There are five kinds of input values which are general system descriptions, system refrigerant-side balancing input, compressor characteristics, fin-and-tube heat exchanger parameters and, configurations inputs and system operating conditions inputs. After inputting these parameters mentioned above and finishing the simulating process, the Heat Pump Design Model can generate several key outputs which are system capacity, power consumption, mass flow rate, sensible heat ratio, system EER or COP, system charge, every key point performance data and component sizing.

In order to be a generic benchmark for the mini-split heat pump system, the HPDM needs to test several case studies. There are two case studies provided in this thesis. The first one implements the lab data inputs for both cooling mode and heating mode. By comparison between the lab performance data and simulation results in the cooling capacity, EER, SHR, mass flow rate and system power consumption, the HPDM produces good results for the cooling mode lab data, except for the condition of an outdoor dry-bulb temperature of 65 F. In addition, the heating capacity, COP, mass flow rate and power consumption parameter comparisons are illustrated in this thesis. These parameters have good simulation results in the lab data heating mode, except for T_{odb} and T_{owb} are -2.92 F and -2.97 F, respectively. The second case study compares manufactural performance output and simulation results in system capacity and power consumption for both cooling mode and heating mode. Capacity prediction is great for the temperature range provided in Case study 2. However, the relative errors for power consumption are relatively small when T_{owb} is between 23 F and 50 F, and the T_{idb} is from 60.8 F to 75.2 F for the heating mode. In addition, the HPDM works well in the combined temperature range of 71.6 F/ 60.8 F~89.6 F/ 75.2 F for T_{idb} /T_{iwb} and 68 F ~104 F for T_{odb}. Since the study of the temperature range of the manufactural data is included in the temperature range of the lab data, by combining these two case studies, the HPDM can be used as a generic benchmark in the temperature range which is shown for the manufactural data for cooling mode and heating mode.

Future studies may focus on power consumption predictions in low outdoor dry-bulb temperatures below 68 F in the cooling mode. For the heating mode, the power consumption should be studied in low T_{owb} temperature ranges (< 23 F) and high T_{owb} temperature ranges (>50 F).

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Appendix I Rotary Compressor Map with Rated Capacity as 7125 Btu/Hr

	Tcond(F)							
Tevap(F)		80	90	100	110	120	130	140
-15	Btu/h	2190	2150					
	Watts	403	425					
	Amps	1.91	2.02					
	Lb/h	28.6	27.9					
-10	Btu/h	2580	2490	2400				
	Watts	414	441	468				
	Amps	1.97	2.1	2.22				
	Lb/h	32.8	31.9	30.9				
-5	Btu/h	3020	2880	2740	2600			
	Watts	424	454	485	516			
	Amps	2.02	2.16	2.3	2.45			
	Lb/h	37.5	36.4	35.3	34.2			
0	Btu/h	3510	3320	3130	2940	2750		
	Watts	431	465	499	534	569		
	Amps	2.06	2.21	2.37	2.53	2.7		
	Lb/h	42.7	41.4	40.2	38.9	37.5		
5	Btu/h	4040	3800	3560	3320	3080		
	Watts	435	473	511	550	589		
	Amps	2.08	2.25	2.42	2.6	2.78		
	Lb/h	48.4	47	45.6	44.1	42.6		
10	Btu/h	4620	4330	4040	3750	3450	3160	2870
	Watts	437	479	521	563	606	649	693
	Amps	2.09	2.28	2.46	2.66	2.85	3.05	3.25
	Lb/h	54.7	53.1	51.5	49.9	48.2	46.5	44.7
15	Btu/h	5250	4910	4560	4220	3880	3530	3190
	Watts	437	483	528	575	621	668	716
	Amps	2.09	2.29	2.49	2.7	2.91	3.12	3.34
	Lb/h	61.5	59.8	58	56.2	54.3	52.4	50.5
20	Btu/h	5930	5540	5140	4740	4350	3950	3560
	Watts	434	484	533	583	634	684	736
	Amps	2.08	2.29	2.51	2.73	2.96	3.18	3.41

	Lb/h	68.9	67	65	63	61	58.9	56.8
25	Btu/h	6670	6220	5770	5320	4870	4420	3970
	Watts	429	482	536	590	644	698	753
	Amps	2.06	2.29	2.52	2.76	2.99	3.23	3.48
	Lb/h	76.9	74.7	72.6	70.4	68.2	65.9	63.7
30	Btu/h	7460	6950	6450	5940	5440	4940	4440
	Watts	421	479	536	594	652	710	769
	Amps	2.03	2.27	2.52	2.77	3.02	3.27	3.53
	Lb/h	85.4	83.1	80.7	78.3	75.9	73.5	71.1
35	Btu/h	8300	7740	7180	6620	6070	5510	4960
	Watts	411	472	534	595	657	719	782
	Amps	1.99	2.25	2.51	2.77	3.04	3.31	3.58
	Lb/h	94.6	92	89.4	86.9	84.3	81.6	79
40	Btu/h	9200	8580	7970	7350	6740	6130	5520
	Watts	399	464	529	594	660	726	793
	Amps	1.95	2.22	2.49	2.77	3.05	3.33	3.61
	Lb/h	104	102	98.7	96	93.2	90.3	87.5
45	Btu/h	10200	9480	8810	8140	7470	6810	6150
	Watts	383	452	522	591	661	731	801
	Amps	1.9	2.18	2.47	2.76	3.05	3.34	3.64
	Lb/h	115	112	109	106	103	99.6	96.6
50	Btu/h	11200	10400	9710	8990	8260	7540	6820
	Watts	366	439	512	585	659	733	807
	Amps	1.84	2.14	2.43	2.74	3.04	3.35	3.66
	Lb/h	126	122	119	116	113	109	106
55	Btu/h	12200	11500	10700	9890	9110	8330	7550
	Watts	346	423	500	577	655	732	810
	Amps	1.77	2.08	2.4	2.71	3.03	3.35	3.67
	Lb/h	137	134	130	127	123	120	116

Appendix II Rotary Compressor Map with Rated Capacity as 11740 Btu/hr

	Tcond(F)							
Tevap(F)		80	90	100	110	120	130	140
-15	Btu/h	3860	3740					
	Watts	697	735					
	Amps	3.35	3.5					
	Lb/h	50	48.3					
-10	Btu/h	4460	4270	4070				
	Watts	712	755	799				
	Amps	3.4	3.57	3.75				
	Lb/h	56.4	54.4	52.3				
-5	Btu/h	5150	4880	4600	4320			
	Watts	724	773	823	873			
	Amps	3.44	3.64	3.84	4.04			
	Lb/h	63.7	61.5	59.3	56.9			
0	Btu/h	5930	5580	5230	4860	4490		
	Watts	734	789	845	901	957		
	Amps	3.47	3.69	3.92	4.15	4.38		
	Lb/h	72	69.6	67.2	64.6	61.8		
5	Btu/h	6800	6370	5940	5500	5050		
	Watts	740	802	863	925	988		
	Amps	3.49	3.74	3.99	4.25	4.51		
	Lb/h	81.3	78.7	76.1	73.3	70.3		
10	Btu/h	7750	7240	6730	6220	5690	5160	4620
	Watts	743	811	879	947	1020	1090	1150
	Amps	3.49	3.77	4.05	4.34	4.62	4.91	5.2
	Lb/h	91.6	88.8	85.9	82.9	79.8	76.5	73.1
15	Btu/h	8790	8200	7610	7020	6420	5810	5190
	Watts	743	817	891	966	1040	1120	1190
	Amps	3.48	3.79	4.1	4.41	4.72	5.04	5.36
	Lb/h	103	99.8	96.7	93.5	90.2	86.7	83.1
20	Btu/h	9910	9250	8580	7910	7230	6540	5850
	Watts	739	820	901	982	1060	1140	1230
	Amps	3.46	3.8	4.14	4.47	4.82	5.16	5.51

	Lb/h	115	112	108	105	102	97.9	94.1
25	Btu/h	11100	10400	9630	8880	8120	7360	6590
	Watts	732	819	906	994	1080	1170	1260
	Amps	3.42	3.79	4.16	4.52	4.89	5.27	5.64
	Lb/h	128	125	121	118	114	110	106
30	Btu/h	12400	11600	10800	9930	9100	8260	7410
	Watts	720	814	908	1000	1100	1190	1290
	Amps	3.37	3.77	4.16	4.56	4.96	5.36	5.76
	Lb/h	142	138	135	131	127	123	119
35	Btu/h	13800	12900	12000	11100	10200	9240	8320
	Watts	704	804	905	1010	1110	1210	1310
	Amps	3.3	3.73	4.15	4.58	5.01	5.44	5.87
	Lb/h	157	153	149	145	141	137	133
40	Btu/h	15200	14200	13300	12300	11300	10300	9310
	Watts	683	791	899	1010	1110	1220	1330
	Amps	3.21	3.67	4.13	4.58	5.04	5.5	5.96
	Lb/h	173	169	165	160	156	152	147
45	Btu/h	16800	15700	14600	13600	12500	11500	10400
	Watts	658	773	887	1000	1120	1230	1350
	Amps	3.11	3.59	4.08	4.57	5.06	5.55	6.04
	Lb/h	190	185	181	177	172	167	163
50	Btu/h	18400	17200	16100	15000	13800	12700	11500
	Watts	628	750	872	993	1120	1240	1360
	Amps	2.98	3.5	4.02	4.54	5.06	5.58	6.1
	Lb/h	207	203	198	194	189	184	179
55	Btu/h	20100	18900	17600	16400	15200	14000	12800
	Watts	593	722	851	980	1110	1240	1370
	Amps	2.83	3.38	3.94	4.49	5.04	5.59	6.14
	Lb/h	226	221	216	212	207	202	196

Appendix III Rotary Compressor Map with Rated Capacity as 5300 Btu/hr

	Tcond(F)							
Tevap(F)		80	90	100	110	120	130	140
-15	Btu/h	1670	1610					
	Watts	307	325					
	Amps	1.34	1.45					
	Lb/h	22.1	21					
-10	Btu/h	1960	1870	1770				
	Watts	316	336	357				
	Amps	1.38	1.5	1.62				
	Lb/h	25.1	23.9	22.7				
-5	Btu/h	2280	2160	2030	1890			
	Watts	322	346	369	393			
	Amps	1.42	1.55	1.67	1.8			
	Lb/h	28.5	27.3	26	24.7			
0	Btu/h	2640	2480	2310	2150	1980		
	Watts	327	353	379	406	433		
	Amps	1.45	1.58	1.72	1.86	1.99		
	Lb/h	32.3	31	29.6	28.2	26.8		
5	Btu/h	3030	2830	2630	2430	2230		
	Watts	330	359	388	417	447		
	Amps	1.47	1.61	1.76	1.9	2.05		
	Lb/h	36.6	35.2	33.7	32.2	30.6		
10	Btu/h	3470	3230	2990	2750	2510	2270	2030
	Watts	332	363	395	427	459	491	524
	Amps	1.49	1.64	1.79	1.94	2.1	2.26	2.41
	Lb/h	41.2	39.7	38.1	36.5	34.9	33.2	31.4
15	Btu/h	3930	3660	3380	3110	2830	2560	2280
	Watts	331	366	400	435	469	504	540
	Amps	1.49	1.65	1.81	1.97	2.14	2.3	2.47
	Lb/h	46.3	44.6	43	41.3	39.6	37.8	35.9
20	Btu/h	4440	4130	3820	3500	3190	2880	2560
	Watts	329	366	403	441	478	516	554
	Amps	1.49	1.66	1.83	2	2.17	2.35	2.52

	Lb/h	51.7	50	48.3	46.5	44.6	42.8	40.8
25	Btu/h	4980	4630	4280	3930	3580	3230	2880
	Watts	325	365	405	445	485	526	567
	Amps	1.48	1.66	1.84	2.02	2.2	2.38	2.56
	Lb/h	57.6	55.8	53.9	52	50.1	48.1	46.1
30	Btu/h	5570	5180	4790	4400	4020	3630	3240
	Watts	320	362	405	448	491	534	577
	Amps	1.46	1.65	1.84	2.03	2.21	2.41	2.6
	Lb/h	63.9	62	60	58	56	53.9	51.8
35	Btu/h	6190	5760	5340	4910	4490	4060	3630
	Watts	312	357	403	448	494	540	586
	Amps	1.44	1.64	1.83	2.03	2.23	2.43	2.63
	Lb/h	70.7	68.6	66.5	64.4	62.3	60.1	57.9
40	Btu/h	6850	6390	5920	5460	4990	4530	4070
	Watts	303	351	399	447	496	545	594
	Amps	1.41	1.62	1.82	2.02	2.23	2.44	2.65
	Lb/h	77.8	75.6	73.5	71.2	69	66.7	64.4
45	Btu/h	7560	7050	6550	6050	5540	5040	4540
	Watts	292	343	394	445	496	548	599
	Amps	1.38	1.59	1.8	2.01	2.23	2.44	2.66
	Lb/h	85.4	83.1	80.8	78.5	76.1	73.7	71.3
50	Btu/h	8310	7760	7220	6680	6130	5590	5050
	Watts	279	333	387	441	495	549	603
	Amps	1.33	1.55	1.78	2	2.22	2.44	2.67
	Lb/h	93.4	91	88.6	86.1	83.6	81.1	78.6
55	Btu/h	9100	8510	7930	7350	6760	6180	5600
	Watts	265	321	378	435	491	548	606
	Amps	1.29	1.51	1.74	1.97	2.21	2.44	2.67
	Lb/h	102	99.3	96.8	94.2	91.6	89	86.3

Appendix IV Rotary Compressor Map with Rated Capacity as 9163 Btu/hr

	Tcond(F)							
Tevap(F)		80	90	100	110	120	130	140
-15	Btu/h	3013	2919	0	0	0	0	0
	Watts	544	574	0	0	0	0	0
	Amps	3	3	0	0	0	0	0
	Lb/h	39	38	0	0	0	0	0
-10	Btu/h	3481	3333	3177	0	0	0	0
	Watts	556	589	624	0	0	0	0
	Amps	3	3	3	0	0	0	0
	Lb/h	44	42	41	0	0	0	0
-5	Btu/h	4020	3809	3590	3372	0	0	0
	Watts	565	603	642	681	0	0	0
	Amps	3	3	3	3	0	0	0
	Lb/h	50	48	46	44	0	0	0
0	Btu/h	4628	4355	4082	3793	3504	0	0
	Watts	573	616	660	703	747	0	0
	Amps	3	3	3	3	3	0	0
	Lb/h	56	54	52	50	48	0	0
5	Btu/h	5307	4972	4636	4293	3941	0	0
	Watts	578	626	674	722	771	0	0
	Amps	3	3	3	3	4	0	0
	Lb/h	63	61	59	57	55	0	0
10	Btu/h	6049	5651	5253	4855	4441	4027	3606
	Watts	580	633	686	739	796	851	898
	Amps	3	3	3	3	4	4	4
	Lb/h	71	69	67	65	62	60	57
15	Btu/h	6861	6400	5940	5479	5011	4535	4051
	Watts	580	638	695	754	812	874	929
	Amps	3	3	3	3	4	4	4
	Lb/h	80	78	75	73	70	68	65
20	Btu/h	7735	7220	6697	6174	5643	5104	4566
	Watts	577	640	703	766	827	890	960
	Amps	3	3	3	3	4	4	4
	Lb/h	90	87	84	82	80	76	73
25	Btu/h	8663	8117	7516	6931	6338	5744	5143
	Watts	571	639	707	776	843	913	983
	Amps	3	3	3	4	4	4	4
	Lb/h	100	98	94	92	89	86	83

30	Btu/h	9678	9054	8429	7750	7102	6447	5783
	Watts	562	635	709	780	859	929	1007
	Amps	3	3	3	4	4	4	4
	Lb/h	111	108	105	102	99	96	93
35	Btu/h	10771	10068	9366	8663	7961	7212	6494
	Watts	549	628	706	788	866	944	1022
	Amps	3	3	3	4	4	4	5
	Lb/h	123	119	116	113	110	107	104
40	Btu/h	11864	11083	10381	9600	8820	8039	7266
	Watts	533	617	702	788	866	952	1038
	Amps	3	3	3	4	4	4	5
	Lb/h	135	132	129	125	122	119	115
45	Btu/h	13112	12254	11395	10615	9756	8976	8117
	Watts	514	603	692	780	874	960	1054
	Amps	2	3	3	4	4	4	5
	Lb/h	148	144	141	138	134	130	127
50	Btu/h	14361	13424	12566	11707	10771	9912	8976
	Watts	490	585	681	775	874	968	1061
	Amps	2	3	3	4	4	4	5
	Lb/h	162	158	155	151	148	144	140
55	Btu/h	15688	14751	13737	12800	11864	10927	9990
	Watts	463	564	664	765	866	968	1069
	Amps	2	3	3	4	4	4	5
	Lb/h	176	172	169	165	162	158	153

Appendix V Scroll Compressor Map with Rated Capacity as 28999 Btu/hr

	Tcond(F)							
Tevap(F)		80	90	100	110	120	130	140
-15	Btu/h	9230	7930					
	Watts	1600	1750					
	Amps	7.44	8.46					
	Lb/h	112	101					
-10	Btu/h	11200	9900	8670				
	Watts	1620	1770	1980				
	Amps	7.43	8.44	9.59				
	Lb/h	135	125	115				
-5	Btu/h	13200	11900	10600	9380			
	Watts	1640	1790	2000	2250			
	Amps	7.42	8.41	9.54	10.9			
	Lb/h	158	149	140	131			
0	Btu/h	15200	13900	12600	11300	9960		
	Watts	1640	1800	2010	2270	2590		
	Amps	7.4	8.38	9.49	10.8	12.3		
	Lb/h	181	173	165	157	146		
5	Btu/h	17400	16000	14700	13400	11900		
	Watts	1630	1800	2020	2280	2590		
	Amps	7.38	8.35	9.44	10.7	12.2		
	Lb/h	204	197	190	183	173		
10	Btu/h	19600	18200	16800	15400	13900	12200	10200
	Watts	1620	1800	2020	2280	2590	2960	3400
	Amps	7.34	8.3	9.39	10.6	12.1	13.9	15.9
	Lb/h	229	222	216	210	200	187	169
15	Btu/h	21900	20500	19000	17600	16000	14200	12200
	Watts	1610	1790	2010	2270	2590	2950	3390
	Amps	7.29	8.26	9.33	10.6	12	13.8	15.8
	Lb/h	255	249	243	237	229	217	199
20	Btu/h	24400	22900	21400	19900	18200	16300	14100
	Watts	1590	1780	2000	2270	2580	2940	3370
	Amps	7.24	8.2	9.27	10.5	12	13.7	15.7

	Lb/h	282	277	272	266	258	247	230
25	Btu/h	27100	25500	23900	22300	20500	18500	16200
	Watts	1560	1760	1990	2260	2570	2930	3340
	Amps	7.17	8.14	9.21	10.4	11.9	13.6	15.6
	Lb/h	311	306	302	297	289	278	262
30	Btu/h	30000	28300	26600	24900	23000	20900	18500
	Watts	1530	1740	1970	2240	2550	2910	3320
	Amps	7.08	8.06	9.14	10.4	11.8	13.5	15.4
	Lb/h	342	338	334	329	322	311	296
35	Btu/h	33100	31300	29500	27600	25600	23400	20800
	Watts	1500	1710	1950	2220	2530	2890	3290
	Amps	6.99	7.98	9.07	10.3	11.7	13.4	15.3
	Lb/h	376	371	368	363	357	347	331
40	Btu/h	36500	34600	32600	30600	28500	26100	23400
	Watts	1460	1680	1930	2200	2510	2870	3270
	Amps	6.88	7.89	8.98	10.2	11.6	13.3	15.2
	Lb/h	412	408	404	400	394	384	369
45	Btu/h	40200	38100	36000	33800	31500	29000	26100
	Watts	1420	1650	1900	2180	2490	2840	3240
	Amps	6.75	7.78	8.89	10.1	11.6	13.2	15.2
	Lb/h	452	447	444	440	434	424	409
50	Btu/h	44100	41900	39700	37300	34900	32200	29100
	Watts	1370	1620	1880	2160	2470	2820	3210
	Amps	6.61	7.66	8.79	10	11.5	13.1	15.1
	Lb/h	494	490	486	482	476	467	452
55	Btu/h	48400	46000	43600	41100	38500	35600	32400
	Watts	1330	1580	1850	2140	2450	2800	3180
	Amps	6.45	7.53	8.68	9.95	11.4	13	15
	Lb/h	541	536	532	528	522	512	498

Appendix VI Scroll Compressor Map with Rated Capacity as 56898 Btu/hr

	Tcond(F)							
Tevap(F)		80	90	100	110	120	130	140
-15	Btu/h	18200	15600					
	Watts	3080	3350					
	Amps	13.6	15.5					
	Lb/h	220	198					
-10	Btu/h	22000	19500	17000				
	Watts	3110	3400	3770				
	Amps	13.6	15.4	17.5				
	Lb/h	265	245	227				
-5	Btu/h	26000	23400	20900	18400			
	Watts	3130	3430	3810	4300			
	Amps	13.6	15.4	17.5	19.9			
	Lb/h	310	292	276	257			
0	Btu/h	30000	27400	24800	22300	19500		
	Watts	3140	3450	3840	4320	4910		
	Amps	13.5	15.3	17.4	19.7	22.5		
	Lb/h	356	339	325	308	286		
5	Btu/h	34200	31500	28900	26200	23400		
	Watts	3130	3450	3850	4340	4930		
	Amps	13.5	15.3	17.3	19.6	22.3		
	Lb/h	402	388	374	359	339		
10	Btu/h	38500	35800	33100	30300	27300	24000	20100
	Watts	3110	3450	3850	4340	4930	5620	6440
	Amps	13.4	15.2	17.2	19.5	22.2	25.4	29.2
	Lb/h	451	437	426	412	394	368	331
15	Btu/h	43200	40300	37500	34600	31400	27900	23800
	Watts	3080	3430	3840	4340	4920	5600	6410
	Amps	13.3	15.1	17.1	19.3	22	25.2	28.9
	Lb/h	501	489	479	466	449	425	390
20	Btu/h	48100	45100	42100	39000	35700	32000	27700
	Watts	3040	3400	3830	4320	4900	5580	6370
	Amps	13.2	15	17	19.2	21.9	25	28.7

	Lb/h	555	544	534	523	507	484	450
25	Btu/h	53400	50200	47000	43800	40300	36300	31800
	Watts	2990	3370	3800	4300	4880	5550	6330
	Amps	13.1	14.9	16.9	19.1	21.7	24.8	28.5
	Lb/h	612	602	593	583	568	546	513
30	Btu/h	59100	55700	52300	48900	45100	40900	36200
	Watts	2930	3330	3770	4270	4850	5520	6280
	Amps	13	14.8	16.7	19	21.6	24.6	28.3
	Lb/h	674	664	656	646	632	611	579
35	Btu/h	65200	61600	58000	54300	50300	45900	40800
	Watts	2870	3280	3730	4240	4820	5480	6230
	Amps	12.8	14.6	16.6	18.8	21.4	24.5	28.1
	Lb/h	740	731	723	714	701	680	649
40	Btu/h	71900	68000	64200	60200	55900	51200	45800
	Watts	2790	3220	3690	4200	4780	5440	6180
	Amps	12.6	14.4	16.4	18.7	21.3	24.3	27.9
	Lb/h	812	803	795	786	773	753	723
45	Btu/h	79100	74900	70800	66500	61900	56900	51200
	Watts	2720	3160	3640	4160	4740	5390	6130
	Amps	12.4	14.2	16.3	18.5	21.1	24.2	27.7
	Lb/h	889	880	873	864	851	832	802
50	Btu/h	86900	82400	78000	73400	68500	63100	57100
	Watts	2640	3100	3590	4120	4700	5350	6070
	Amps	12.1	14	16.1	18.4	21	24	27.6
	Lb/h	973	964	957	948	935	915	886
55	Btu/h	95300	90600	85800	80800	75600	69800	63400
	Watts	2550	3040	3540	4080	4660	5300	6020
	Amps	11.8	13.8	15.9	18.2	20.8	23.9	27.4
	Lb/h	1060	1050	1050	1040	1030	1010	976

Appendix VII laboratory data inputs for cooling modes

Intended Outdoor Dry-bulb [F]	Intended Indoor Dry-bulb [F]	Intended Indoor Wet-bulb [F]	Superheat F	Subcooling [F]	Indoor coil Air volume Flow rate [cfm]
87	80	67	17.83	12.36	226.72
87	74	62	22.82	12.33	229.43
95	80	67	23.01	12.07	213.89
105	80	67	24.39	10.51	213.48
105	80	56	25.77	9.62	231.72
105	80	56	25.77	9.74	240.02
95	74	66	24.03	11.84	201.82
87	74	66	21.77	12.66	201.18
115	80	67	26.83	9.50	219.25
87	74	66	21.95	12.83	200.94
95	74	66	23.96	11.96	202.06
105	80	67	25.61	10.19	169.69
105	80	67	26.50	9.92	142.44
67	80	67	14.33	11.96	230.13
67	74	62	16.22	11.09	229.84
105	74	62	27.18	10.01	229.55
67	80	56	16.62	10.53	229.43
67	80	67	13.71	10.69	141.26
67	74	66	14.95	11.74	206.71
67	74	66	14.85	11.35	223.37

Appendix VIII laboratory performance data sets for cooling modes

Intended Outdoor Dry-bulb [F]	Intended Indoor Dry-bulb [F]	Intended Indoor Wet-bulb [F]	Capacity Btu/hr	EER [Btu/W-h]	Sensible Heat Ratio	Refrigerant Flow rate [lbm/hr]	Power [W]
87	80	67	10341	13.95	0.69	140.48	778
87	74	62	8340	11.35	0.74	111.16	754
95	80	67	8412	10.46	0.71	116.60	823
105	80	67	8128	9.01	0.73	120.28	919
105	80	56	7674	8.57	1.04	114.99	901
105	80	56	7677	8.64	1.03	114.88	901
95	74	66	8176	10.46	0.58	113.19	811
87	74	66	8620	12.12	0.56	114.16	749
115	80	67	7476	8.42	0.76	117.90	1001
87	74	66	8582	12.49	0.56	113.45	746
95	74	66	8135	11.15	0.57	112.33	813
105	80	67	7879	8.94	0.66	117.26	902
105	80	67	7640	8.54	0.63	114.31	893
67	80	67	7811	17.36	0.74	91.82	436
67	74	62	7401	8.40	0.77	87.48	434
105	74	62	7367	16.41	0.78	109.26	895
67	80	56	7349	16.60	1.00	87.66	432
67	80	67	8794	21.06	0.61	108.05	422
67	74	66	7725	19.33	0.57	91.32	430
67	74	66	7698	18.14	0.61	90.72	429

Appendix IX HPDM laboratory data outputs for cooling modes

Intended Outdoor Dry-bulb [F]	Intended Indoor Dry-bulb [F]	Intended Indoor Wet-bulb [F]	Capacity Btu/hr	EER [Btu/W-h]	Sensible Heat Ratio	Refrigerant Flow rate [lbm/hr]	Power W
87	80	67	9491	12.06	0.70	122.40	787.1
87	74	62	8786	11.27	0.71	112.00	779.4
95	80	67	9137	10.84	0.68	121.00	843.1
105	80	67	8609	9.46	0.69	120.40	910.1
105	80	56	7772	8.73	1.00	108.80	890.1
105	80	56	7845	8.61	1.00	110.60	881.6
95	74	66	8639	10.41	0.57	117.00	830.0
87	74	66	9155	11.68	0.56	117.70	783.6
115	80	67	8004	8.13	0.69	114.30	984.0
87	74	66	9112	11.75	0.56	117.50	775.3
95	74	66	8784	10.57	0.57	118.80	830.7
105	80	67	8210	8.92	0.65	115.80	890.4
105	80	67	7837	8.59	0.63	110.10	882.4
67	80	67	7804	22.10	0.73	91.10	353.1
67	74	62	7396	19.23	0.74	83.70	384.6
105	74	62	7601	8.51	0.74	105.10	893.8
67	80	56	7387	19.05	1.00	86.10	387.3
67	80	67	8687	16.32	0.61	103.40	532.5
67	74	66	7724	20.23	0.58	90.40	381.9
67	74	66	7656	20.58	0.61	86.60	372.0

Appendix X laboratory data inputs for heating modes

Intended Indoor Dry-bulb [F]	Intended Indoor Wet-bulb [F]	Intended Outdoor Dry-bulb [F]	Intended Outdoor Wet-bulb F	Superheat F	Subcooling [F]	Indoor coil Air volume. Flow rate [cfm]
69.76	53.83	46.96	38.51	8.59	18.00	225
63.93	50.88	62.17	49.63	9.87	11.22	232
76.03	58.15	62.02	49.26	10.23	7.04	224
64.04	51.31	68.07	53.08	9.14	8.26	228
70.00	53.87	41.91	34.80	14.17	15.10	229
69.89	54.97	42.00	37.14	13.97	14.29	227
63.93	51.98	34.09	29.23	19.18	18.14	229
75.94	58.91	35.04	30.52	10.79	15.07	228
70.00	54.99	61.90	51.21	10.49	5.68	229
63.93	48.03	61.88	46.59	m	19.64	229
76.08	53.91	61.97	47.85	7.82	16.16	229
76.17	55.49	67.78	51.39	6.75	15.87	230
69.93	52.56	67.91	51.22	6.43	18.86	230
70.03	53.38	62.04	48.37	-0.25	2.42	146
69.91	54.03	62.01	48.66	9.69	27.38	186
69.98	51.71	46.92	37.42	9.17	20.03	229
70.03	51.75	47.03	37.42	9.40	11.34	148
70.18	53.24	41.97	40.50	6.83	22.45	228
70.07	54.90	34.99	30.28	1.54	23.13	230
69.91	51.31	26.92	23.10	4.62	30.28	230
70.05	51.03	27.14	23.92	2.22	7.23	147
70.00	49.62	17.05	15.57	10.77	29.66	229
69.94	49.43	7.03	6.33	5.62	28.60	229
69.94	49.80	34.93	27.69	3.64	11.85	187
69.96	49.95	-2.92	-2.97	1.53	26.59	230
69.98	51.64	46.84	39.09	16.38	21.51	185
70.05	51.62	34.98	29.89	1.60	18.77	187

Appendix XI laboratory performance data sets for heating modes

Intended Indoor Dry-bulb [F]	Intended Indoor Wet-bulb [F]	Intended Outdoor Dry-bulb [F]	Intended Outdoor Wet-bulb F	Capacity Btu/hr	COP [W/W]	Refrigerant Flow rate [lbm/hr]	Power W
69.76	53.83	46.96	38.51	11267	2.96	137.24	1084
63.93	50.88	62.17	49.63	12420	3.69	155.29	958
76.03	58.15	62.02	49.26	11673	3.09	157.30	1078
64.04	51.31	68.07	53.08	12922	3.72	165.90	990
70.00	53.87	41.91	34.80	11137	2.87	133.45	1112
69.89	54.97	42.00	37.14	11147	2.86	134.33	1118
63.93	51.98	34.09	29.23	13720	2.33	157.83	1688
75.94	58.91	35.04	30.52	11130	2.20	135.19	1435
70.00	54.99	61.90	51.21	11741	3.35	152.38	993
63.93	48.03	61.88	46.59	12785	3.64	155.95	990
76.08	53.91	61.97	47.85	12117	3.13	156.66	1088
76.17	55.49	67.78	51.39	10004	3.85	126.52	729
69.93	52.56	67.91	51.22	10114	4.23	123.17	668
70.03	53.38	62.04	48.37	10209	2.62	163.74	977
69.91	54.03	62.01	48.66	10769	2.99	127.16	1005
69.98	51.71	46.92	37.42	10687	3.17	129.65	939
70.03	51.75	47.03	37.42	9203	2.46	130.01	1029
70.18	53.24	41.97	40.50	11888	2.85	142.46	1177
70.07	54.90	34.99	30.28	12144	2.31	146.43	1458
69.91	51.31	26.92	23.10	12816	2.09	143.10	1741
70.05	51.03	27.14	23.92	10083	1.68	152.12	1395
70.00	49.62	17.05	15.57	11311	1.97	125.55	1599
69.94	49.43	7.03	6.33	9824	1.87	107.59	1452
69.94	49.80	34.93	27.69	7643	2.85	101.06	737
69.96	49.95	-2.92	-2.97	8367	1.74	90.68	1315
69.98	51.64	46.84	39.09	10578	2.74	131.09	1103
70.05	51.62	34.98	29.89	10588	2.23	136.03	1340

Appendix XII HPDM laboratory data outputs for heating modes

Intended Indoor Dry-bulb [F]	Intended Indoor Wet-bulb [F]	Intended Outdoor Dry-bulb [F]	Intended Outdoor Wet-bulb F	Capacity Btu/hr	COP [W/W]	Refrigerant Flow rate [lbm/hr]	Power W
69.76	53.834	47.0	38.5088	11261	2.926	151.5	1128
63.93	50.882	62.2	49.6256	12529	3.676	163.4	998.9
76.03	58.154	62.0	49.262	11521	3.213	160.6	1051.1
64.04	51.314	68.1	53.078	12916	3.885	164.2	974.4
70.00	53.87	41.9	34.8008	11152	2.846	144.4	1148.3
69.89	54.968	42.0	37.139	11108	2.841	144.6	1145.8
63.93	51.98	34.1	29.2262	13591	2.311	160.5	1723.7
75.94	58.91	35.0	30.51626	11217	2.301	146.1	1428.7
70.00	54.986	61.9	51.206	11848	3.366	168.2	1031.7
63.93	48.0254	61.9	46.5872	12972	3.675	160.3	1034.5
76.08	53.906	62.0	47.8454	11950	3.337	154.7	1049.5
76.17	55.49	67.8	51.386	9980	4.182	124.3	699.4
69.93	52.556	67.9	51.224	10036	4.688	117.3	627.5
70.03	53.384	62.0	48.3692	10198	2.77	161.2	1078
69.91	54.032	62.0	48.6608	10824	3.698	127	857.8
69.98	51.71	46.9	37.4234	10565	3.107	141.1	997
70.03	51.746	47.0	37.4162	9107	2.608	138.8	1024
70.18	53.24	42.0	40.496	11800	2.806	153.5	1232.5
70.07	54.896	35.0	30.28334	12075	2.322	155.6	1523.9
69.91	51.314	26.9	23.1044	12709	2.08	148.7	1790.6
70.05	51.026	27.1	23.9162	10143	1.814	142.2	1638.8
70.00	49.6166	17.0	15.5696	11301	2.071	122.5	1599.2
69.94	49.4312	7.0	6.332	9765	2.024	97.8	1414
69.94	49.8038	34.9	27.6872	7524	2.864	106.7	769.9
69.96	49.9496	-2.9	-2.974	8211	2.159	65.7	1114.5
69.98	51.638	46.8	39.092	10401	2.918	136.1	1045
70.05	51.62	35.0	29.894	10470	2.337	141.3	1313.3

Appendix XIII HPDM manufactural data outputs for cooling modes

Simulation manufactural data Power consumption (W)								
Indoor unit		Outdoor unit dry bulb temperature (F)						
wet bulb (F)	dry bulb (F)	68	77	89.6	95	104	109.4	114.8
57.2	68	547.8	598.1	672	704.7	746.1	780.4	814.7
60.8	71.6	546.8	598.8	674.1	708.9	747.5	786.9	822.1
64.4	77	545.5	598.6	676.3	711	757.3	793	828.9
66.2	80.6	544.9	597.6	677.1	711.7	758.9	795.6	832.9
71.6	86	542.9	596.6	677.7	713.9	762.3	800.9	838.4
75.2	89.6	542.1	595.5	677.8	714.6	763.7	801.8	840.7

Simulation manufactural data capacity (Btu/hr)								
Indoor unit		Outdoor unit dry bulb temperature (F)						
wet bulb (F)	dry bulb (F)	68	77	89.6	95	104	109.4	114.8
57.2	68	8881	8452	7839	7554	7130	6853	6574
60.8	71.6	9425	9010	8364	8046	7742	7273	6969
64.4	77	9933	9555	8950	8608	8048	7723	7381
66.2	80.6	10209	9835	9232	8946	8329	7947	7617
71.6	86	10912	10610	9989	9689	9138	8755	8351
75.2	89.6	11225	10978	10410	10138	9623	9239	8854

Appendix XIV HPDM manufactural data outputs for heating modes

Simulation manufactural data capacity (Btu/hr)							
Indoor unit	outdoor unit wet bulb temperature (F)						
dry bulb	5	14	23	32	42.8	50	59
60.8	8800	9229	10002	10393	11988	12574	13650
64.4	8725	9240	9993	10334	11718	12438	13604
68	8711	9246	9991	10273	11704	12212	13590
69.8	8639	9240	10026	10246	11640	12216	13538
71.6	8624	9243	9987	10213	11477	12217	13419
75.2	8546	9176	9899	10154	11438	12112	13238
Simulation manufactural data power consumption (W)							
Indoor unit	outdoor unit wet bulb temperature (F)						
dry bulb (F)	5	14	23	32	42.8	50	59
60.8	653.1	710	725.7	806.1	916.5	987.4	1142.2
64.4	670.9	739.3	752.6	828.1	918.6	1005.7	1176.5
68	698.9	791.4	780.4	850.4	972.2	1022.3	1216
69.8	712.7	796.5	799.6	861.7	985.5	1041.6	1228.7
71.6	725.5	829.9	810.3	872.7	988.6	1060.1	1231
75.2	746	840.7	827.3	895.2	1017.5	1070	1245

APPENDIX XV lab data suction, discharge pressure and system charge for cooling mode

Intended Outdoor Dry-bulb	Intended Indoor Dry-bulb	Intended Indoor Wet-bulb	suction pressure	discharge pressure	system charge
[F]	[F]	[F]	[psi]	[psi]	[lb]
87	74	62	114.61	349.98	2.31
95	80	67	122.14	386.82	2.31
105	80	67	127.78	429.46	2.31
105	80	56	122.63	424.96	2.31
105	80	56	122.79	424.96	2.31
95	74	66	118.41	384.21	2.31
87	74	66	117.50	351.14	2.31
115	80	67	129.45	477.76	2.31
87	74	66	117.05	351.28	2.31
95	74	66	118.41	384.21	2.31
105	80	67	124.95	427.43	2.31
105	80	67	122.01	425.25	2.31
67	80	67	120.29	265.85	2.31
67	74	62	114.57	262.08	2.31
105	74	62	118.16	423.66	2.31
67	80	56	113.90	260.34	2.31
67	80	67	134.16	269.92	2.31
67	74	66	119.00	264.55	2.31
67	74	66	117.93	262.52	2.31

APPENDIX XVI HPDM results for lab data suction, discharge pressure and system charge for cooling mode

Intended Outdoor Dry-bulb	Intended Indoor Dry-bulb	Intended Indoor Wet-bulb	suction pressure	discharge pressure	system charge
[F]	[F]	[F]	[psi]	[psi]	[lb]
87	74	62	117.6	341	2.216
95	80	67	129.1	381.5	2.267
105	80	67	131.7	428.4	2.246
105	80	56	119.5	421.5	2.164
105	80	56	120.8	422.1	2.169
95	74	66	125.4	379.5	2.248
87	74	66	123.2	344.3	2.257
115	80	67	133.6	488.8	2.36
87	74	66	123.1	344.6	2.263
95	74	66	125.5	379.8	2.253
105	80	67	125.8	425.3	2.213
105	80	67	120.6	422.6	2.186
67	80	67	108.6	258.1	2.119
67	74	62	144.8	250.4	2.422
105	74	62	131.7	428.4	2.246
67	80	56	107	257.6	2.122
67	80	67	108.6	258.1	2.119
67	74	66	144.8	250.4	2.422
67	74	66	142.5	250.4	2.415

APPENDIX XVII lab data suction, discharge pressure and system charge for heating mode

Intended Indoor Dry-bulb	Intended Indoor Wet-bulb	Intended Outdoor Dry-bulb	Intended Outdoor Wet-bulb	suction pressure	discharge pressure	system charge
[F]	[F]	[F]	[F]	[psi]	[psi]	[lb]
69.76	53.834	47.0	38.5088	120.29	462.67	2.30
63.93	50.882	62.2	49.6256	149.24	442.66	2.30
76.03	58.154	62.0	49.262	153.16	502.70	2.30
64.04	51.314	68.1	53.078	159.25	458.32	2.30
70.00	53.87	41.9	34.8008	112.09	452.08	2.30
69.89	54.968	42.0	37.139	113.04	454.11	2.30
63.93	51.98	34.1	29.2262	92.71	488.63	2.30
75.94	58.91	35.0	30.51626	95.91	491.82	2.30
70.00	54.986	61.9	51.206	148.08	458.47	2.30
63.93	48.0254	61.9	46.5872	149.24	466.73	2.30
76.08	53.906	62.0	47.8454	152.29	518.51	2.30
76.17	55.49	67.8	51.386	166.50	460.64	2.30
69.93	52.556	67.9	51.224	163.02	430.18	2.30
70.03	53.384	62.0	48.3692	162.15	526.34	2.30
69.91	54.032	62.0	48.6608	140.96	517.64	2.30
69.98	51.71	46.9	37.4234	124.67	446.43	2.30
70.03	51.746	47.0	37.4162	129.27	517.93	2.30
70.18	53.24	42.0	40.496	116.61	486.31	2.30
70.07	54.896	35.0	30.28334	96.51	494.43	2.30
69.91	51.314	26.9	23.1044	84.92	526.92	2.30
70.05	51.026	27.1	23.9162	93.46	500.24	2.30
70.00	49.6166	17.0	15.5696	73.10	485.88	2.30
69.94	49.4312	7.0	6.332	62.73	447.15	2.30
69.94	49.8038	34.9	27.6872	110.23	405.96	2.30
69.98	51.638	46.8	39.092	121.66	503.72	2.30
70.05	51.62	35.0	29.894	99.47	506.33	2.30

APPENDIX XVIII HPDM results for lab data suction, discharge pressure and system charge for heating mode

Intended Indoor Dry-bulb	Intended Indoor Wet-bulb	Intended Outdoor Dry-bulb	Intended Outdoor Wet-bulb	suction pressure	discharge pressure	system charge
[F]	[F]	[F]	[F]	[psi]	[psi]	[lb]
69.76	53.834	47.0	38.5088	115.70	458.30	2.18
63.93	50.882	62.2	49.6256	146.2	443.9	2.3
76.03	58.154	62.0	49.262	149.8	500.5	2.292
64.04	51.314	68.1	53.078	162.2	459.1	2.299
70.00	53.87	41.9	34.8008	105.7	449	2.2
69.89	54.968	42.0	37.139	102.9	447.4	2.051
63.93	51.98	34.1	29.2262	80.1	481.3	1.912
75.94	58.91	35.0	30.5163	90.6	483.8	1.974
70.00	54.986	61.9	51.206	146.8	460.2	2.269
63.93	48.0254	61.9	46.5872	146.3	462.6	2.295
76.08	53.906	62.0	47.8454	149.4	513.5	2.3
76.17	55.49	67.8	51.386	171.4	448.8	2.3
69.93	52.556	67.9	51.224	170.5	419.9	2.3
70.03	53.384	62.0	48.3692	156.8	542.7	2.3
69.91	54.032	62.0	48.6608	155.8	494.3	2.3
69.98	51.71	46.9	37.4234	117.8	441.3	2.3
70.03	51.746	47.0	37.4162	122.3	505	2.134
70.18	53.24	42.0	40.496	109.4	475.4	2.1
70.07	54.896	35.0	30.2833	87.9	479.3	2
69.91	51.314	26.9	23.1044	72.8	506.4	2.1
70.05	51.026	27.1	23.9162	80	534.8	2
70.00	49.6166	17.0	15.5696	64.4	429	1.9
69.94	49.4312	7.0	6.332	53.8	396.2	1.9
69.94	49.8038	34.9	27.6872	101.4	396.1	2.034
69.98	51.638	46.8	39.092	117.8	493.7	2.2
70.05	51.62	35.0	29.894	92.8	486.9	2.2