Air-Conditioning, Heating and Refrigeration Technology Institute

Final Report

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LIFE CYCLE CLIMATE PERFORMANCE MODEL FOR RESIDENTIAL HEAT PUMP SYSTEMS

Final Report

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Life Cycle Climate Performance Model for Residential Heat Pump Systems

Executive Summary

A Microsoft Excel based program has been developed to simulate life cycle climate performance (LCCP) for residential heat pumps. The LCCP model includes the direct impacts of refrigerant emissions, the indirect impacts of energy consumption used to operate the heat pump system, and the energy to manufacture and safely dispose the system and refrigerant. The annual energy consumption for heat pump operation is calculated using input performance data at several operating points, in a number of different formats, assuming a linear relationship as defined in AHRI Standard 210/240. With appropriate input, the program can handle different heat pump systems, refrigerants, locations, and CO_2 emission profiles of power plants.

With its modular structure, the program can be easily modified to evaluate other air conditioning or refrigeration systems.

Introduction

Under increasing pressure to address global warming concerns, the industry is spending more effort to understand the environmental impact of air conditioning systems using different refrigerants and technologies. The environmental performance of air conditioning or heat pump systems is partially defined by life cycle impacts on climate, including the direct impacts of refrigerant emissions, the indirect impacts of energy consumption used to operate the heat pump system, and the energy to manufacture, transport, and safely dispose of the system, all expressed in terms of CO_2 equivalent emissions. Thus it is necessary to have a comprehensive analytical tool to count all aspects of the environmental impact by air conditioning or heat pump systems.

It is the objective of this project to develop a Microsoft Excel based simulation tool to calculate life time direct or indirect emissions generated by residential heat pumps. The simulation tool can handle different types of heat pump systems with different refrigerants, locations, and power generation CO_2 emission profiles. The following sections introduce the past work on TEWI and LCCP including the simulation tool for automotive air conditioning, the methodology and development of the new simulation program for residential heat pumps, and the ways to use the tool. In addition to the main frame for broad data input and emission calculation, the important part of the simulation tool is the computation of the annual energy for heat pump operation, which is described in detail in the appendix.

Literature Survey

Past TEWI and LCCP Study

In the early 1990s when alternative refrigerants were implemented to replace CFC and HCFC, the US DOE and AFEAS jointly sponsored projects to identify the major applications of refrigerants worldwide and to examine the impacts of the refrigerants on overall emissions of greenhouse gases (Fisher et al., 1991, 1994). Baseline and alternative refrigerants, as well as technologies were examined for typical equipment in five applications - automobile air conditioning, supermarket refrigeration, unitary heat pumps and air conditioners, chillers for cooling large office and commercial buildings, and household refrigeration. Conventional systems for these applications all employ compressors, fans and sometimes pumps to move heat either out of a cooled space or into a heated space. Consequently, these systems can lead to the emission of two different greenhouse gases (GHGs). First, the energy consumed by the systems, in the form of electricity or the direct combustion of a fossil fuel, results in the release of carbon dioxide. Second, almost all of the refrigerants used in these applications are GHGs. If the refrigerant leaks out of the system during operation, is lost during maintenance, or is not recovered when the system is scrapped, it contributes to global warming. The two GHG contributions are expressed in Total Equivalent Warming Impact (TEWI). The TEWI methodology explicitly seeks to identify both the "direct" effect of greenhouse emissions from the product and the "indirect" effect of carbon dioxide emissions related to the energy consumption of the product. Later on, the DOE evaluated the global impacts of alternative HFCs, natural refrigerants, and new technologies that have a reasonable potential of becoming commercial products for the five applications before 2015 in terms of TEWI (Sand et al., 1997). The study included applications in Europe, Japan and North America and used representative data for each region for equipment size and efficiency, weather and climate, and CO_2 emissions from power generation.

Gopalnarayanan et al. (1999) reported their work on TEWI of R-22 alternatives in air-conditioning and heat pump applications. A total of eight R-22 alternatives (seven HFCs plus R-290) were analyzed. The performance of the system in heating and cooling modes was determined by using a comprehensive computer simulation model written in Engineering Equation Solver (EES, 2010). The method or assumptions for the detailed simulation are as follows:

- Using the EES program and REFPROP (version 9.0, Lemmon et al., 2010) for refrigerant properties, the equations for mass and energy balances as well as heat transfer coefficients were solved simultaneously.
- The model was based on an actual 12 SEER, 8 HSPF, and 3.5 RT heat pump.
- The heat exchanger performance was determined using the log mean temperature difference (LMTD) method with the condenser being divided into three regions (desuperheating, two-phase, and subcooling) and two evaporator regions (superheat and two phase).
- The air side heat transfer coefficient was assumed to be 70 W/m^2K.
- The isentropic efficiencies of the blowers as well as the compressor were assumed to be 75%.
- The degradation coefficient for cyclic performance was assumed to be the same for all fluids and was 0.1for the cooling mode and 0.15 for the heating mode.
- The direct emission was based on an assumed annual leak rate of 4% of the initial system charge. The amount of R-22 in the heat pump was assumed to be 5 kg (11 lb).
- The equipment life was assumed to be 15 years.

In the above work, the simulation was carried out for the different test conditions specified in ASHRAE Standard 116. The seasonal energy efficiency ratio (SEER) as well as the heating seasonal performance factor (HSPF) was calculated for the different regions of the United States. The results indicated that R-32 had the lowest TEWI, and the direct contribution to global warming from refrigerant leakage was just a small fraction of the TEWI.

At almost the same time Sand et al. (1999) also reported their work on TEWI comparison for fluorocarbon alternative refrigerants in residential heat pumps and air conditioners. In this work, they used the SEER and HSPF data for R-22 directly and efficiency data relative to R-22 to calculate SEER and HSPF for R-407C, R-410A and R-290. They concluded that the TEWIs for residential systems using blends of HFCs as alternatives are not significantly different from those calculated for R-22.

Although these two studies used similar assumptions (4% annual leak rate and 15 years of equipment life time), Sand et al.'s study used 5% lower CO_2 emission from electricity production than Gopalnarayanan et al.'s study and considered an end-of-life charge loss of 15%. Therefore, Sand et al.'s study considered more direct effects than Gopalnarayanan et al. The findings from both studies support the argument that the major TEWI contribution of the air conditioning system is the indirect effect.

The concept of Life Cycle Climate Performance (LCCP) is more comprehensive than the TEWI, which ignores the energy embodied in product materials, the greenhouse gas emissions during chemical manufacturing, and the end-of- life loss. The LCCP concept was first proposed by the TEAP of the UNEP (1999) to calculate the “cradle-to-grave” climate impacts of the direct and indirect greenhouse gas emissions.

The ADL reports (1999, 2002) used the LCCP concept to investigate overall environmental performance of specific HFCs compared to other fluids and technologies in the applications including automobile air conditioning, residential and commercial refrigeration, unitary air conditioning, HVAC chillers, foam insulation, solvent cleaning, aerosols, and fire protection. The embodied energy and GHG emissions associated with fluorocarbon production and end of product life loss/emission of working fluids were counted. The reports claim that it is inappropriate to use a 100 year integration time horizon (ITH) in conjunction with certain compounds, because carbon dioxide has a lifetime over 100 years but many HFCs do not; nonetheless, the report uses this ITH in its calculation.

Spatz, M. W. (2003) studied performance and LCCP of thee R-22 alternatives in heat pumps including R-410A, R-407C, and R-290. The LCCP impacts included direct effect of refrigerant leakage and end-of-life loss, and indirect effect of power consumption. A detailed system modeling for energy use was conducted using the compressor maps, tube-to-tube modeling for evaporators and condensers, and analytical models or correlations for expansion devices. Five (5) European cities with their temperature bins, the average of twenty-nine (29) American cities, and Phoenix were chosen for the analysis. A linear cooling and heating load was assumed. The evaluation showed that the indirect effect dominates the LCCP of heat pumps, and that R-410A is an efficient refrigerant in this application.

To help provide a clear understanding of the relative performance potential of HFCs (R-404A and R-410A) as compared to R-290 for walk-in refrigeration systems representing direct expansion commercial refrigeration systems with small charge, the CEEE at the University of Maryland (Hwang et al., 2007) performed an experimental evaluation of the three refrigerants. To compare the environmental impact of refrigerants over the entire life cycle of fluid and equipment, including power consumption, the life cycle climate performance (LCCP) of the three refrigerants was evaluated based on measured data. The estimated LCCPs at various emission rates indicate that the LCCP of R-290 is always lower than that of R-404A. The LCCP of R-410A is lower than that of R-290 as long as the annual emission is kept below 10%. It was concluded that R-410A has less or equivalent environmental impact as compared to R-290 when safety (toxicity and flammability), environmental impact (climate change), cost and performance (capacity and COP) are considered.

GREEN-MAC-LCCP

As best as we can discern, the GREEN-MAC-LCCP is the first comprehensive analytical tool that is publicly available to measure environment impact performance of mobile air conditioning (MAC) systems (Papasavva et al., 2010). The proposal for a global peer-reviewed analytical tool that assesses the GHG emissions of alternative refrigerants originated from the U.S. EPA, and in early 2005 SAE International established a working group with GM’s Stella Papasavva. Additionally, William Hill chaired a global LCCP team that was tasked with the goal of developing and disseminating the model for public use. Since then more than 50 world experts have participated in the development and perfection of the GREEN-MAC-LCCP. The first version was released for public use in 2006, and the current version is 3b. It has become the standard tool in the MAC industry.

The GREEN-MAC-LCCP is an Excel format based program. It is a sophisticated accounting of the expected life-cycle climate impacts of any MAC system including direct and indirect emissions as follows:

LCCP = GWP [Direct from MAC leaks] + GWP [Direct from additional sources :(atmospheric reaction products of refrigerant) + (manufacturing, transport & service leakage) + (EOL refrigerant emissions)] + GWP [Indirect from MAC operation] + GWP [Indirect from additional sources: (chemical production of refrigerant & transport) + (MAC manufacturing & its vehicle assembly) + (EOL recycling processes)] (1)

Direct emissions result from refrigerant leaks into the atmosphere and are an aggregate of:

- Regular emissions, due to refrigerant leaks from the A/C system during operation
- Irregular emissions due to accidents, stone hits, product defects etc
- Service emissions from professional and DIYer (Do-it-Yourselfer) servicing operations
- EOL emissions considering the recovery, if any, of refrigerant at the EOL (End-of-Life) of the vehicle
- Leakage in assembly plants
- Atmospheric reaction products from the atmospheric breakdown of HFCs

The direct CO_2 equivalent emissions are estimated using the GWP of each chemical and its mass emitted into the atmosphere.

Indirect emissions result from the energy consumption due to MAC manufacturing, operation and EOL and are an aggregate of:

- Manufacturing and EOL energy of alternative refrigerants and MAC system components
- Energy consumption from MAC operation during vehicle's lifetime
- Energy from additional fuel consumption to transport the MAC mass on board the vehicle during vehicle's lifetime

The Indirect CO_2 emissions are a function of the carbon content of the fuel utilized in each process and during vehicle operation.

When starting the program, the user is requested to input refrigerant selection and vehicle selection for each drive cycle type (FTP, SC03, NEDC, Japan JC08, and India). Then the user enters or edits input for refrigerant leakage, energy consumption or equivalent CO_2 emission for manufacturing and EOL of components and refrigerants, A/C system capacity and COP, drive cycle, and fan power. The user can use default numbers or test matrix data for some of the input. TMY2 weather data is used. All the input data on the spreadsheet is automatically integrated into the analysis. The results include detailed numerical data of direct and indirect emissions, and charts that present lifetime LCCP, annual LCCP, and lifetime LCCP components.

Figure 1 shows one example of the lifetime LCCP CO_2-eq. contributions of a baseline system (Papasavva et al., 2010). The GREEN-MAC-LCCP program is a powerful transparent tool. It accounts for all important emission factors, and allows users to input all custom data which are unique to the technologies or refrigerants to be evaluated.

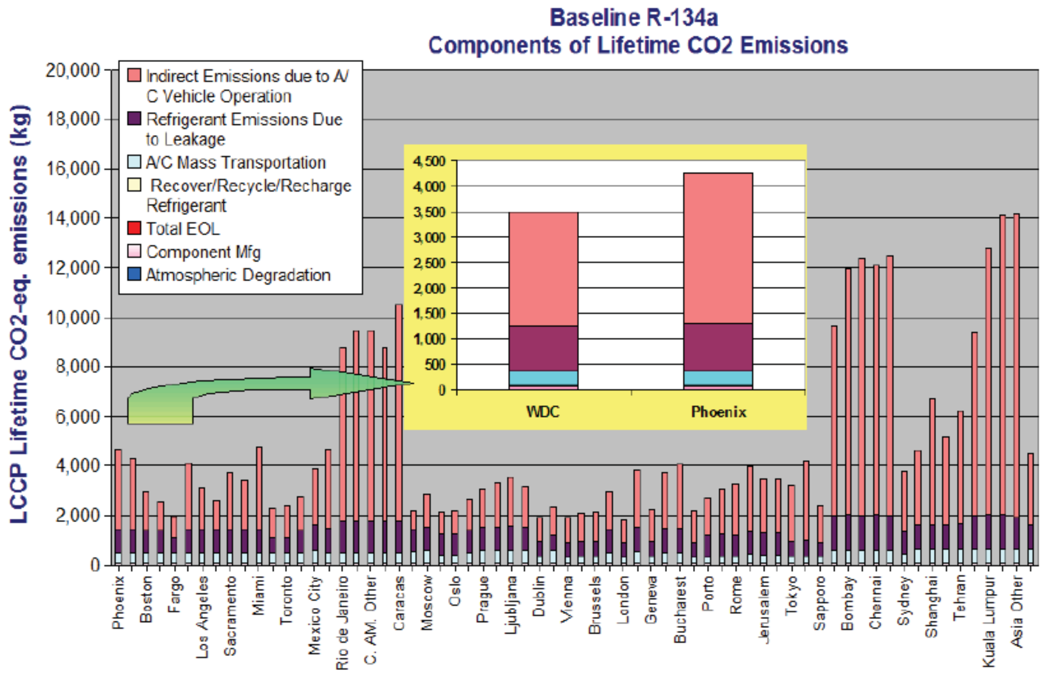


Figure 1: LCCP CO_2-eq. Direct and Indirect contributions of baseline R134a system (Papasavva et al., 2010)

With the GREEN-MAC-LCCP program, Koban M. (2009) conducted a LCCP analysis for HFO-1234yf, which has a 100 year direct GWP of 4, versus the current refrigerant, HFC-134a, with a 100 year direct GWP of 1,430. LCCP calculations were done in several key cities representing different climatic regions of the world. In this investigation, several potential scenarios were run to investigate this premise. HFO-1234yf was compared with 1) equal capacity and equal COP, 2) equal capacity and 3% improved COP and 3) equal capacity and 5% improved COP versus HFC-134a. It was noted that in scenarios 2 and 3, an internal heat exchanger was added to achieve the desired COP increases. The model was appropriately upgraded to capture the increase in weight due to charge requirements and additional piping. Results of annual LCCP values for selected key cities indicated that in all three scenarios, HFO-1234yf provided significant reductions to total contribution to climate change versus the baseline refrigerant HFC-134a. Anywhere from 5.2 to 5.9 million metric tons CO_2 equivalent would be removed if HFC-134a were to be replaced by HFO-1234yf on a global basis by the year 2017 in all new vehicles with AC.

AHRTI LCCP Calculation Methodology

This section introduces the method used for the new simulation tool.

For residential heat pumps, the direct emission due to refrigerant leakage includes the following:

- Regular and irregular refrigerant leakage from heat pump equipment
- Refrigerant loss at EOL

Other minor direct emission such as leakage during the manufacturing process is not listed here but it can be taken into account by adjusting the input to the above major refrigerant loss.

The indirect emission due to energy consumption is an aggregate of:

- System operating energy
- Energy consumption for components manufacturing (including refrigerant manufacturing)
- Energy consumption for components EOL (including refrigerant EOL)

Thus the lifetime LCCP is calculated as follows:

LCCP = Direct emission + Indirect emission

= (*Ref. GWP + Adp.GWP*) x *(annual leakage* x *years of lifetime + refrigerant loss at EOL)* + *years of lifetime* x Σ *(equivalent CO_2 kg/kWh* x *operating energy kWh)$_{annual}$* + Σ *(equivalent CO_2 kg/kg material* x *mass of materials kg)* + Σ *(equivalent CO_2 kg/kg material* x *mass of recycled materials kg)* (2)

where *Ref.GWP* is the refrigerant GWP value and *Adp.GWP* is the GWP of atmospheric degradation product of the refrigerant. *Annual leakage* includes both regular leakage and irregular leakage (such as leakage due to service) and represents the average over the years of lifetime being evaluated. Note that the indirect emission due to transport of refrigerant, components, and systems is not addressed here because it is relatively small.

The details regarding the calculation of heat pump operating energy are described in the appendix. The method adopted is based on the AHRI 210/240 standard (2008). It essentially uses test data obtained at specific conditions (95°F and 82°F for cooling, and 47°F, 35°F, and 17°F for heating) and a linear relationship to derive energy for each temperature bin to obtain annual energy consumption. Different algorithms (equations) are used for different types of units (single speed, two capacity, and variable speed). The ambient temperature data can be obtained from the TMY3 database (Wilcox et al., 2008). The annual energy plus the information describing equivalent CO_2 emission per kWh by the utility can give indirect emission due to heat pump operation. All other direct and indirect emissions are calculated with appropriate input or assumptions such as leakage rate.

The flowchart of the LCCP calculation is presented in Figure 2.

Development of the Excel Simulation Tool

Using the methodology described in the previous section and appendix, a Microsoft Excel-based simulation has been developed. A series of macros of Visual Basic for Application (VBA) are implemented for the energy calculation described in the appendix and other direct or indirect emission computation. The Excel simulation program has the following spreadsheets:

Main

This spreadsheet provides high level input data like refrigerant, location, name of the heat pump data sheet, and path for TMY3 weather database. It also gives high level calculation results.

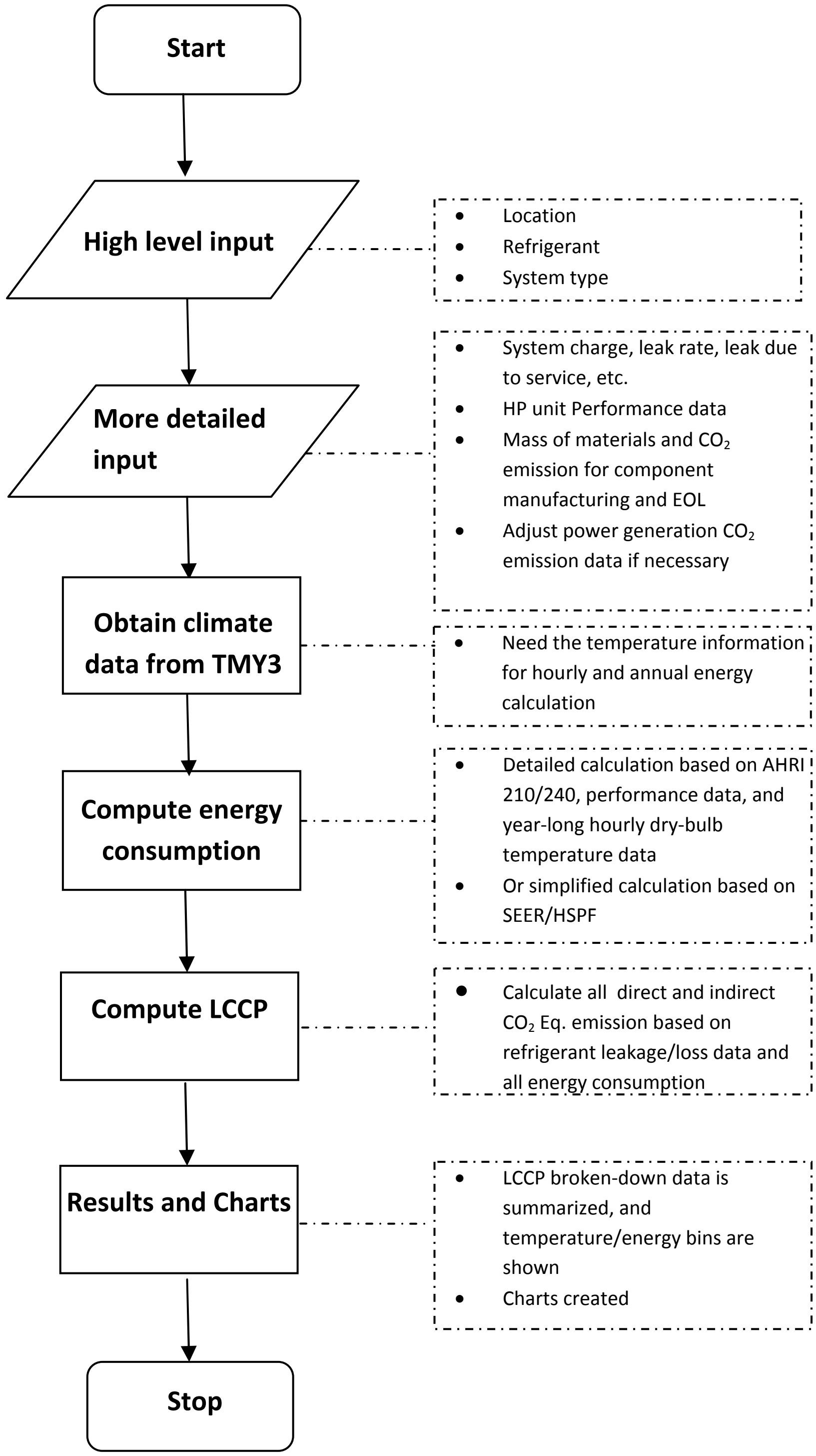


Figure 2: Flowchart for LCCP calculation

Refrigerants

It lists a majority of refrigerants that are of interest to the industry with the refrigerant GWP value, CO_2-equivalent emission for virgin refrigerant manufacturing, and the GWP for atmospheric reaction byproducts. The refrigerant GWP is from AR4 reports (IPCC, 2007). The GWP values of other refrigerants, currently not listed in AR4, were provided by manufacturers or compiled from publicly available information. The user can add more refrigerants to this spreadsheet.

CityUtilityInfo

The sheet provides the list of the cities and their heating region and utility region that is needed to obtain the CO_2 emission for each kWh of electricity by the power plant. The user can define the CO_2 emission rate as a function of hour in a day and month in a year. The existing number of the average CO_2 rate for each region is obtained from the NREL technical report NREL/TP-550-38617 (Deru et al., 2007). The NREL report divides North America into five (5) interconnected utility regions – Eastern Interconnection, Western Interconnection, Ercot Interconnection, Alaska, and Hawaii, which have the average CO_2 rates of 0.788, 0.594, 0.834, 0.774, and 0.865 kg CO_2/kWh, respectively. Within each of the five utility regions, the power network is interconnected and one cannot tell which specific location or power plant electricity comes from, and so the CO_2 emission rate for power generation within each utility region is considered to be the same. This method is utilized in current work.

The set of tables consisting of the CO_2 rate for each region as a function of time and month on the sheet is only a framework due to lack of time related information (the average CO_2 rate for each region from the NREL report is currently applied for each hour and month), and it can be updated in the future as information becomes available.

The user can also expand the city list by including more cities following the user guide.

Heat pump data sheet

This sheet provides heat pump performance data at operating conditions required by the AHRI 210/240 (e.g., 95°F and 82°F for cooling, and 47°F, 35°F, and 17°F for heating), leakage information, and CO_2 emission for components manufacturing and EOL. It also provides input for backup heat, setting for heat pump shutdown when ambient temperature is too low, etc. The user can choose to do energy calculation for both heating and cooling, or cooling only; detailed energy calculation using performance data, or simple energy estimate based on SEER and HSPF; backup heat using electric heating, or gas/oil heating, or without backup heat; calculation of CO_2 emission for components manufacturing and EOL based on detailed material mass of each individual component, or lump sum mass of the unit. Different types of equipment (single speed, two capacity, variable speed, and custom unit) use different spreadsheets for data input.

Information on default numbers of some inputs from our investigation is as follows:

- Heat pump lifetime – fifteen (15) years is used as the default lifetime, as assumed by Sand et al. (1997, pp43) for U.S. and European unitary equipment.

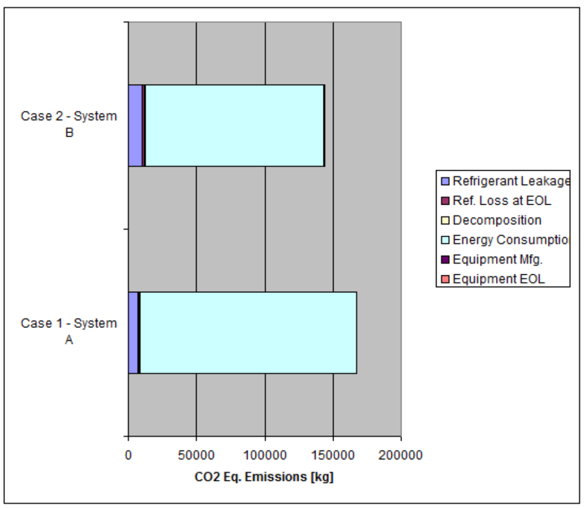
- System refrigerant charge – Sand et al. (1997, pp52) gave an average charge of 0.26 kg per kW for ducted R22 residential systems. In the present work, AHRI provided an estimate from heat pump manufacturers that the R-410A charge is roughly 9.75 pounds for a 3 ton system with SEER of 13 and 14.5 pounds for a 3 ton system with SEER of 16. This is close to the value obtained from another OEM – 10 pounds for 3 tons with SEER of 13. Thus we chose the default number of 10 for 3 tons with SEER of 13.
- Annual leak rate – it is more difficult to establish a universal number for annual leak rate. Sand et al. (1997, pp43) mentioned a maximum residential heat pump annual leak rate of 4% of the charge for 1996 equipment and 2% per year for equipment available in 2005. Bateman (1999) gave a trend of annual emission rate for unitary equipment from 10% in the 1980's to 5% in the late 1990's to 2% in the future. Considering the wide range of the annual leak rate and the fact that the equivalent annual emission from irregular leakage such as leakage due to service is also counted in the annual leak rate in this program, per suggestion of PMS members, we have set the default value for annual leak rate to 5% at present. The user can adjust it as a more accurate number becomes available.
- Refrigerant loss at end of life (EOL) – the default number of charge loss at EOL is set at 15%, which was decided upon for residential units based on the idea of recovering 90% of the charge from 95% of the field units, but allowing for a 100% charge loss from about 5% of the field stock (Sand et al., 1997, pp43).
- Equivalent CO_2 emission for component manufacturing – five (5) major materials – aluminum, steel, copper, brass, and plastics are counted in the CO_2 emission calculation. The values of equivalent CO_2 emission per kilogram of material were initially adopted from the GREEN-MAC-LCCP program, which used data from GaBi and Boustead LCA Databases. Then we performed a search of public domain documents and obtained the following numbers:
    - Aluminum – 10.6 kg CO_2/kg ("U.S. Energy Requirements for Aluminum Production", BCS report prepared for DOE, Feb 2007)
    - Copper – 4.0 kg CO_2/kg ("Energy Requirements and CO2-emissions from Manufacturing and Maintenance of Locomotives and Trains", Simonsen, M., March 2009, http://vfp1.vestforsk.no/sip/pdf/Jernbane/TrainManufacturing.pdf)
    - Steel – 2.02 kg CO_2/kg (US Life Cycle Inventory Database – Primary Metal Manufacturing, http://www.nrel.gov/lci/database/default.asp)
    - Plastics – 1.8 ~3.8 (average 2.8) kg CO_2/kg ("Cradle-To-Gate Life Cycle Inventory of Nine Plastic Resins and Four Polyurethane Precursors", Franklin Associates report prepared for the American Chemistry Council , July 2010)

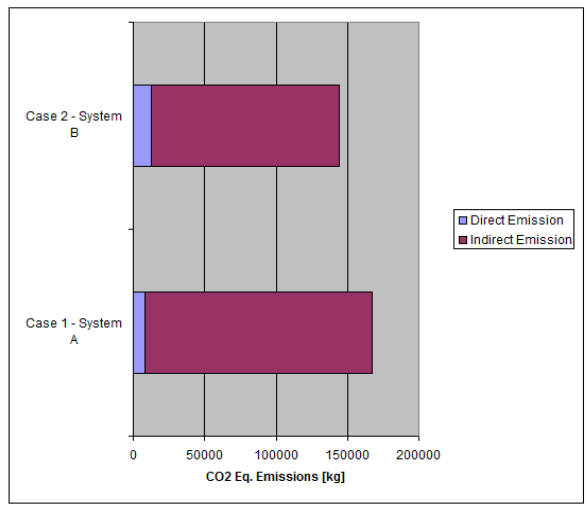
  The values of copper, steel, and plastics are consistent with those of GREEN-MAC-LCCP, but the value of aluminum is significantly different from GREEN-MAC-LCCP (1.6 kg CO_2/kg). The numbers from GREEN-MAC-LCCP are used as default values for the present time. Further investigation may be needed.
- Mass of materials – the default values of mass of components used to calculate emission for components manufacturing and recycling were provided by an AHRI member for a typical 3 ton residential heat pump. The user can adjust the values for different tonnage units.

Results

Once input data are provided and computation starts, the program reads in climate data from the TMY3 database, conducts hourly energy calculation for all 8760 hours of a meteorological year, and computes all indirect and direct emission. The *Results* sheet gives detailed calculation results which include: emission charts; numerical data – direct and indirect emission, as well as the breakdown of numbers; temperature bins and energy bins for heat pump operation. Two useful numbers – SEER and HSPF are also given on the results sheet. SEER and HSPF are calculated using the unit performance data and standard temperature bins defined in AHRI 210/240. One must be careful about the HSPF calculation because several factors including heating region, low temperature cut-off, the use of minimum or maximum heating requirement, and cyclic degradation coefficient can affect the result.

Figure 3 shows results for a sample case comparison for a residential heat pump system. The numerical result includes the temperature bins and energy for the same simulation. From the sample case data, one can see that for a residential heat pump system, the majority of the emission is indirect emission due to energy consumption. In contrast, the direct effect of the automotive A/C application is much more noticeable (see above Figure 1).





Detailed Results:			
Case #		1	2
Case Name		System A	System B
HP Data Worksheet		HPData-SS-FF-E	HPData-TC-EN
City		CHICAGO, IL	CHICAGO, IL
Refrigerant		R410A	R410A
Total Lifetime Emission	kg CO2-Eq.	168324	144716
Total Direct Emission	kg CO2-Eq.	8524	12360
Ref. Leakage	kg/year	0.23	0.33
Emission - Ref. Leakage	kg CO2-Eq.	7103	10300
Ref. Loss at EOL	kg	0.68	0.99
Emission - Ref. Loss at EOL	kg CO2-Eq.	1421	2060
Emission - Decomposition	kg CO2-Eq.		
Total Indirect Emission	kg CO2-Eq.	159800	132356
Annual Energy Consumption	kW-hr	13474	11149
Emission - Energy Consumption	kg CO2-Eq.	159260	131776
Emission - Equipment Mfg	kg CO2-Eq.	517	557
Emission - Equipment EOL	kg CO2-Eq.	23	23
Detailed Energy Calculation			
Total Annual Energy Consumption	kW-hr	13474	11149
Annual Cooling Energy	kW-hr	2172	1639
Annual Heating Energy	kW-hr	11302	9509
Backup Heat	kW-hr	2521	2222
SEER		13.3	15.8
HSPF		7.5	9.0

Cooling Season Temp./Energy Bins				
65 ~ 69F		hrs	767	767
	(building load)	MBtu	2936	2634
		MJ	3097	2779
	(cooling delivered)	MBtu	2936	2634
		MJ	3097	2779
	(power consumed)	kW-hr	184	156
70 ~ 74F		hrs	538	538
	(building load)	MBtu	4926	4420
		MJ	5197	4663
	(cooling delivered)	MBtu	4926	4420
		MJ	5197	4663
	(power consumed)	kW-hr	327	264
75 ~ 79F		hrs	531	531
	(building load)	MBtu	7706	6915
		MJ	8130	7296
	(cooling delivered)	MBtu	7706	6915
		MJ	8130	7296
	(power consumed)	kW-hr	543	417
80 ~ 84F		hrs	428	428
	(building load)	MBtu	9101	8166
		MJ	9601	8616
	(cooling delivered)	MBtu	9101	8166
		MJ	9601	8616
	(power consumed)	kW-hr	686	499
85 ~ 89F		hrs	160	160
	(building load)	MBtu	4452	3995
		MJ	4697	4215
	(cooling delivered)	MBtu	4452	3995
		MJ	4697	4215

Figure 3: Sample results of LCCP calculation

Validation of the Program

It is necessary to validate the LCCP program. Since indirect emission due to energy consumption accounts for a majority of the total LCCP, it is critical to validate the energy calculation. Two steps were taken to validate the program: (1) Implement the equations using Excel cells, and compare to the program from the National Institute of Standards and Technology (NIST) ("NIST -SEER-HSPF-MacroV4.xls"); (2). Implement the equations of annual energy and total emission to Excel files step-by-step and compare the results with output of our LCCP program. The validation covers single speed, two capacity, and variable speed.

1. Implement the equations to Excel cells, and compare to the NIST program ("NIST-SEER-HSPF-MacroV4.xls")

   All the equations for energy calculation used in our program were implemented in the Excel cells step by step. Unit performance data were kept the same as the default numbers in the NIST program, and the calculated SEER and HSPF based on standard bins were compared to that from the NIST program. The comparison is given in Table 1.

Table 1: Comparison of SEER and HSPF values calculated using NIST program and equations in LCCP program

			Our method	NIST program	Note
Single speed	SEER		10.23	10.23	
	HSPF	DHR_min	7.21	7.43	difference is due to defrost credit 1.03 in NIST program
		DHR_max	6.22	6.4	difference is due to defrost credit 1.03 in NIST program
Two capacity	SEER		13.36	13.36	
	HSPF	DHR_min	8.21	8.45	difference is due to defrost credit 1.03 in NIST program
		DHR_max	6.54	6.72	difference is due to defrost credit 1.03 in NIST program
Variable speed	SEER		15.75	15.75	
	HSPF	DHR_min	11.77	11.76	
		DHR_max	6.81	6.82	

One can see from the table that our calculation results match the NIST program except for single speed and two capacity heating HSPF data, for which the difference is caused by the fact that the NIST program uses "demand defrost credit" of 1.03. The defrost heat is not included in the LCCP program at present because it may involve extensive information to define details of the defrost cycle.

2. Implement the equations of annual energy and total emission used in the LCCP program to Excel files step-by-step and compare the results with output of our LCCP program
   This was implemented for several cities including Chicago, Miami, LA, etc. The hand calculation completely matches the output from the LCCP program.

The consistency of the LCCP computed for different scenarios presented in the next session is a further validation of the program.

Use of the Tool for Case Studies

This section presents the use of the program for case studies and the consistency of the prediction results. One can refer to the Users' Guide or Help file for instructions on how to use the program.

1. Comparison of different units with the same refrigerant and location

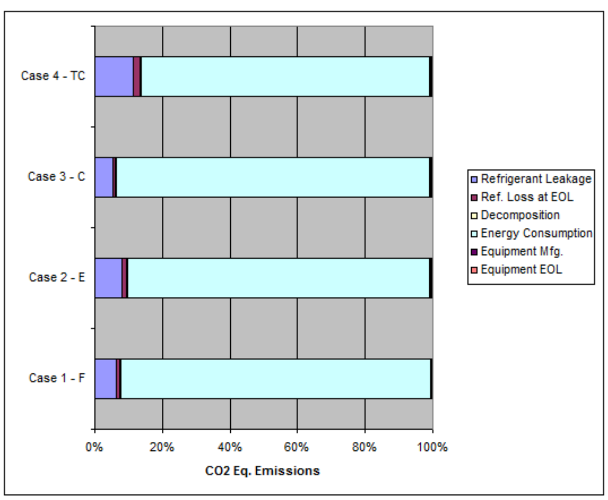
   Table 2 gives performance data of three single speed units provided by AHRI. Unit "F" is a 3 ton unit with a SEER of 13, unit "E" is a 3 ton with SEER a of 14.5, and unit "C" is a 5 ton with a SEER of 13.5. Table 3 shows data of a 3 ton two capacity unit with a SEER of 16 (data provided by an OEM). It is assumed that the location is Washington DC, and the refrigerant is R-410A. The mass of component materials needs to be scaled up for the 5 ton unit (unit "C") from the default 3 ton numbers. With all data input the program can complete the calculations quickly (1 ~ 2 minutes). Figure 4 shows detailed results and a comparison of the units.

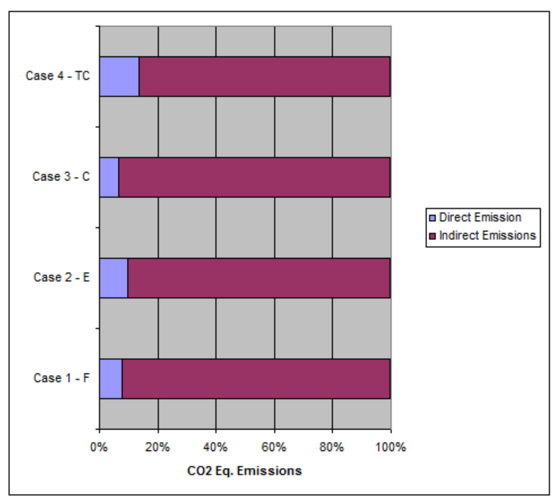
Table 2: Representative data provided by AHRI for single speed units

Unit Code	C	E	F
Capacity95FHigh, btu/h	58500	35149	36170
EER95F	11.20	12.92	11.82
Capacity82FHigh, btu/h	63000	37449	38496
EER82F	13.90	15.61	14.17
Cooling Degrad. Coef.	0.02	0.13	0.13
SEER	13.70	14.60	13.26
IndoorCoilAirQty, cfm	1900	1045.00	1200.00
IndoorCoilAirQty2, cfm	n/a	n/a	n/a
HighHeat47F, btu/h	56000	32292.00	34015.00
HighCOP47F	3.48	3.51	3.56
LowHeat17F, btu/h	35800	19357.00	20337.00
LowCOP17F	2.46	2.33	2.40
Heating Degrad. Coef.	0.06	0.12	0.25
HSPF	8.70	8.26	8.24
Charge (R410A.):	12 lbs. 11 oz.	~ 10.6 lbs	~ 9 lbs. 12 oz.

Table 3: Performance data of two capacity unit

(High capacity)							
Test	A_2 (95)	B_2 (82)		$H1_2$ (47)	$H3_2$ (17)	$H2_2$ (35)	
caapcity, Btu/h	36526	39214		37585	23968	31105	
EER/COP	12.3	14.76		3.68	2.78	3.34	
Evap. SCFM	1235	1236		1185	1192	1187	
C_d		0.19				0.25	
(Low Capacity)							
Test	B_1 (82)	F_1 (67)		$H0_1$ (62)	$H1_1$ (47)	$H3_1$ (17)	$H2_1$ (35)
Capacity, Btu/h	28077	31000		32000	25255	15589	21190
EER/COP	17.86	22		14.34	3.61	2.59	3.17
Evap. SCFM	883				844	843	851
C_d		0.19					0.25
SEER/HSPF		16.2					9.1
Charge (R-410A)	~ 14.5 lbs.						



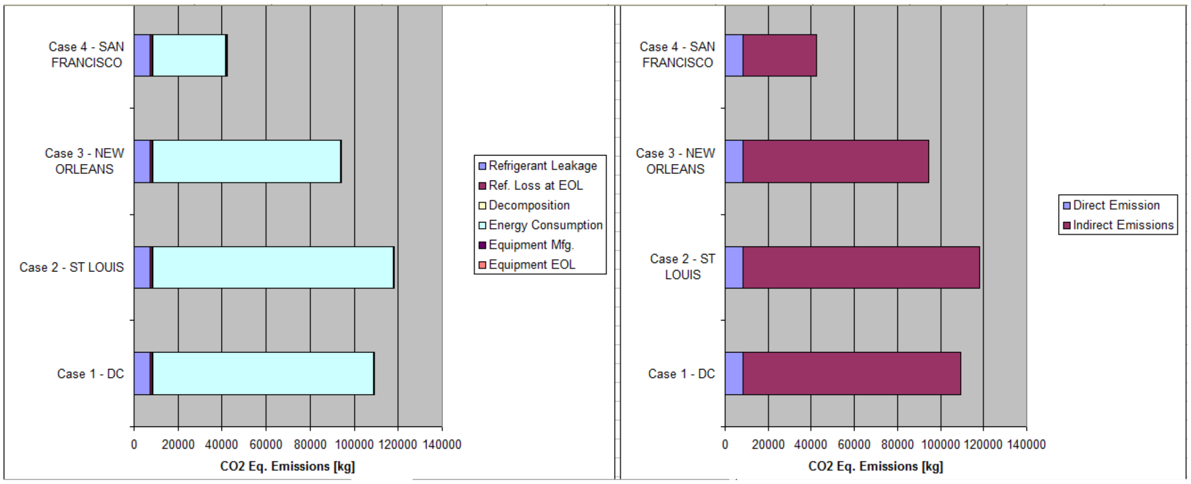


Detailed Results:					
Case #		1	2	3	4
Case Name		F	E	C	TC
HP Data Worksheet		HPData-SS-FF-EN	HPData-SS-FF-EN-1	HPData-SS-FF-EN-2	HPData-TC-EN
City		WASHINGTON DC	WASHINGTON DC	WASHINGTON DC	WASHINGTON DC
Refrigerant		R410A	R410A	R410A	R410A
Total Lifetime Emission	kg CO2-Eq.	109579	93813	168150	90617
Total Direct Emission	kg CO2-Eq.	8311	8950	10826	12360
Ref. Leakage	kg/year	0.22	0.24	0.29	0.33
Emission - Ref. Leakage	kg CO2-Eq.	6926	7459	9021	10300
Ref. Loss at EOL	kg	0.66	0.71	0.86	0.99
Emission - Ref. Loss at EOL	kg CO2-Eq.	1385	1492	1804	2060
Emission - Decomposition	kg CO2-Eq.				
Total Indirect Emission	kg CO2-Eq.	101268	84863	157325	78257
Annual Energy Consumption	kW-hr	8522	7134	13237	6572
Emission - Energy Consumption	kg CO2-Eq.	100730	84319	156462	77676
Emission - Equipment Mfg	kg CO2-Eq.	514	521	824	557
Emission - Equipment EOL	kg CO2-Eq.	23	23	38	23
Detailed Energy Calculation					
Total Annual Energy Consumption	kW-hr	8522	7134	13237	6572
Annual Cooling Energy	kW-hr	3189	2812	4967	2511
Annual Heating Energy	kW-hr	5333	4321	8270	4060
Backup Heat	kW-hr	83	36	112	36
SEER		13.2	14.5	13.7	16.2
HSPF		8.0	8.3	8.7	9.1

Figure 4: Comparison of different units with the same refrigeration and location

For this specific case the direct emission is only 8 ~ 13% of the total emission. Unit "E" has lower lifetime CO_2 emission than unit "F" because it is more energy efficient with a SEER of 14.5 compared to a SEER of 13 for unit "F". Unit "C" has much higher emission because it is a 5 ton unit compared to 3 ton of others. The two capacity unit has the lowest lifetime emission. It appears that all other elements (equipment manufacturing, etc.) in the LCCP are negligible except for direct effect of refrigerant leakage and EOL, and indirect effect of energy consumption.

2. Comparison of different locations
   Unit "F" in Table 2 is modeled for different locations – Washington DC, St Louis, New Orleans, and San Francisco, and results are shown in Figure 5. St Louis has both higher cooling energy and higher heating energy than the DC area although their latitudes are almost the same. This may be because of the more extreme weather in the Midwest. New Orleans has much higher cooling energy, but also much lower heating energy, and the total energy and lifetime emission are lower than DC and St Louis. San Francisco has the lowest CO_2 emission.

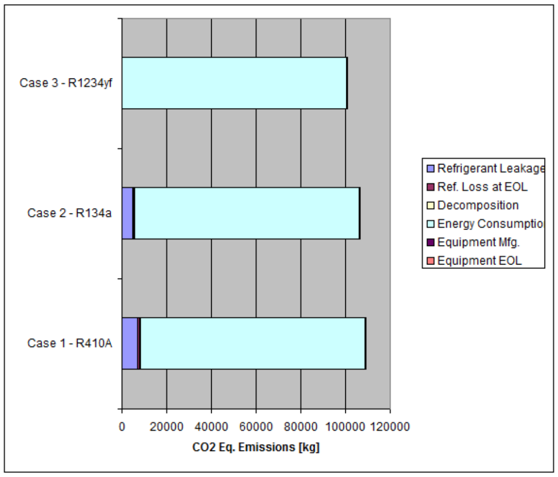


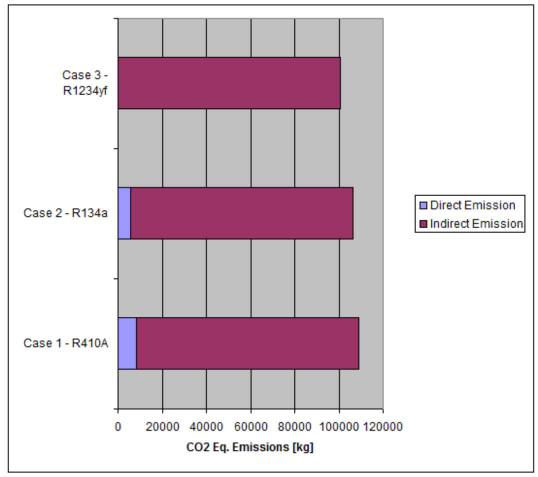
Detailed Results:

Case #		1	2	3	4
Case Name		DC	ST LOUIS	NEW ORLEANS	SAN FRANCISCO
HP Data Worksheet		HPData-SS-FF-E	HPData-SS-FF-EN	HPData-SS-FF-EN	HPData-SS-FF-EN
City		WASHINGTON	ST LOUIS, MO	NEW ORLEANS, LA	SAN FRANCISCO, CA
Refrigerant		R410A	R410A	R410A	R410A
Total Lifetime Emission	kg CO2-Eq.	109579	118517	94607	42459
Total Direct Emission	kg CO2-Eq.	8311	8311	8311	8311
Ref. Leakage	kg/year	0.22	0.22	0.22	0.22
Emission - Ref. Leakage	kg CO2-Eq.	6926	6926	6926	6926
Ref. Loss at EOL	kg	0.66	0.66	0.66	0.66
Emission - Ref. Loss at EOL	kg CO2-Eq.	1385	1385	1385	1385
Emission - Decomposition	kg CO2-Eq.				
Total Indirect Emission	kg CO2-Eq.	101268	110206	86296	34148
Annual Energy Consumption	kW-hr	8522	9278	7255	3772
Emission - Energy Consumption	kg CO2-Eq.	100730	109668	85758	33610
Emission - Equipment Mfg	kg CO2-Eq.	514	514	514	514
Emission - Equipment EOL	kg CO2-Eq.	23	23	23	23
Detailed Energy Calculation					
Total Annual Energy Consumption	kW-hr	8522	9278	7255	3772
Annual Cooling Energy	kW-hr	3189	3385	5680	381
Annual Heating Energy	kW-hr	5333	5893	1575	3391
Backup Heat	kW-hr	83	533		
SEER		13.2	13.2	13.2	13.2
HSPF		8.0	8.7	9.1	9.3

Figure 5: Results for different locations

3. Modeling for different refrigerants





Detailed Results:

Case #		1	2	3
Case Name		R410A	R134a	R1234yf
HP Data Worksheet		HPData-SS-FF-EN	HPData-SS-FF-EN	HPData-SS-FF-EN
City		WASHINGTON DC	WASHINGTON DC	WASHINGTON DC
Refrigerant		R410A	R134a	R1234yf
Total Lifetime Emission	kg CO2-Eq.	109579	106955	101271
Total Direct Emission	kg CO2-Eq.	8311	5698	29
Ref. Leakage	kg/year	0.22	0.22	0.22
Emission - Ref. Leakage	kg CO2-Eq.	6926	4743	13
Ref. Loss at EOL	kg	0.66	0.66	0.66
Emission - Ref. Loss at EOL	kg CO2-Eq.	1385	949	3
Emission - Decomposition	kg CO2-Eq.		6	13
Total Indirect Emission	kg CO2-Eq.	101268	101257	101241
Annual Energy Consumption	kW-hr	8522	8522	8522
Emission - Energy Consumption	kg CO2-Eq.	100730	100730	100730
Emission - Equipment Mfg	kg CO2-Eq.	514	503	488
Emission - Equipment EOL	kg CO2-Eq.	23	23	23
Detailed Energy Calculation				
Total Annual Energy Consumption	kW-hr	8522	8522	8522
Annual Cooling Energy	kW-hr	3189	3189	3189
Annual Heating Energy	kW-hr	5333	5333	5333
Backup Heat	kW-hr	83	83	83
SEER		13.2	13.2	13.2
HSPF		8.0	8.0	8.0

Figure 6: Comparison of different refrigerants

To understand the LCCP for different refrigerants, unit "F" in Table 2 is modeled with R-134a, R-410A, and R-1234yf. This is not a perfect way to compare refrigerants because a unit with a different refrigerant should have different performance, or, in order for a system to have the same performance, the system should have different components or system design. In this work we simply assume unit "F" maintains the same performance for each refrigerant. This actually only allows one to see the effect of refrigerants on direct emission at the same charge level. Figure 6 gives the calculation results. As expected, R-1234yf has the lowest lifetime emission because its GWP is only 4 compared to a GWP of 1430 for R-134a and 2088 for R-410A.

4. Simple calculation using SEER and HSPF
   The user can also choose to perform a simple energy calculation using SEER and HSPF data. The details of this method are introduced in the appendix. Unit "F" in Table 2 is modeled using SEER and HSPF data for four cities – Washington, St Louis, New Orleans, and San Francisco. The comparison of energy consumption based on detailed calculation and SEER/HSPF is given in Table 4.

Table 4: Comparison of energy consumption based on detailed calculation and SEER/HSPF

		Washington, DC	St Louis	New Orleans	San Francisco
(Cooling energy)					
Detailed calculation	kWh	3189	3385	5680	381
Using SEER	kWh	2339	2937	5711	1632
(Heating energy with minimum design heating requirement)					
Detailed calculation	kWh	5333	5893	1575	3391
Using HSPF	kWh	7091	4667	1842	4709
(Heating energy with maximum design heating requirement)					
Detailed calculation	kWh	11553	13160	2981	6452
Using HSPF	kWh	17357	11424	4509	11527

Compared to the detailed calculation, the SEER/HSPF method works well for New Orleans, but under predicts cooling energy for Washington DC, over predicts cooling energy for San Francisco, under predicts heating energy for St Louis, and over predicts heating energy for Washington DC and San Francisco. The under or over prediction by the SEER/HSPF method could be caused by the following factors:

- The actual local heating or cooling hours are different from those obtained from Figure 7 and 8 in the appendix;
- The actual local outdoor design temperature is different from the standard outdoor design temperature;
- The HSPF value is normally published for Region IV. Although a correction factor is introduced to correct for other heating regions (see the appendix), this still could be a factor affecting results.

The user should take the results from the simple SEER/HSPF method as reference only.

Since the value of HSPF for unit "F" (in Table 2) is based on minimum design heating requirement, the HSPF method using maximum design heating requirement tends to over predicts the heating energy significantly for most cities, as shown by the data of the last two rows of Table 4.

Conclusions

- An Excel program with VBA subroutines has been developed for predicting the LCCP of residential heat pumps. The program uses heat pump performance data and a linear relationship to derive annual energy consumption, as well as inputs for refrigerant charge and loss, mass of component materials, and others to calculate all direct or indirect emission.
- The program has been validated by implementing the equations and computing process used in the program to cells of an Excel file step by step (hand calculation), and comparing the (hand calculation) results to the NIST program and output of the LCCP program.
- The program has been utilized to analyze the LCCP of different units with different refrigerants and locations. The program gives consistent results for different scenarios. It appears that all other elements (equipment manufacturing, etc.) in the LCCP composition are negligible except for the direct effect of refrigerant leakage and EOL and the indirect effect of energy consumption.
- A simplified energy calculation method utilizing nominal SEER and HSPF data is provided, but caution is warranted as the results using this method do not appear to correlate with the results from the detailed calculation.
- Significant effort has been made to investigate default parameter inputs such as annual leak rate, recycling loss, equivalent CO_2 emission for component manufacturing, etc. Further study may be needed.

Future Work

This simulation tool is intended to become the foundation for further expansion. We recommend the following future work:

- Add or research more accurate input data, such as the CO_2 emission rate for power plants as a function of time and season, energy and CO_2 emission for material or components manufacturing, regular refrigerant leakage rate, etc. The five (5) interconnected utility regions could be further divided into sub-regions or even states if data is available.
- Use the tool to further study different heat pump systems, and make conclusions on their performance and environmental characteristics.
- Expand the tool to other heat pump and refrigeration applications.
- Expand the tool to other regions of interest internationally.

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Appendix

LCCP Calculation Algorithm

The general equations for LCCP calculation are as follows:

Lifetime LCCP = Direct emission + Indirect emission

Direct emission = Eq. CO_2 *emission due to lifetime leakage + Eq.* CO_2 *emission from decomposition*

Eq. CO_2 *emission due to lifetime leakage = ref.GWP* x *lifetime leakage*

= Ref.GWP x *(annual leakage* x *years of lifetime + refrigerant loss at EOL)*

= Ref. GWP x *(refrigerant charge* x *annual leakage rate* x *years of lifetime + refrigerant charge* x *percentage of loss when reclaim)*

Eq. CO_2 *emission from decomposition = Adp_GWP* x *lifetime leakage*

Where *Ref.GWP* is the refrigerant GWP number and *Adp.GWP* is GWP of atmospheric degradation product of the refrigerant.

Indirect emission = CO_2 *emission due to system operating energy consumption +* CO_2 *emission due to energy consumption for components manufacturing & EOL*

CO_2 *emission due to system operating energy consumption = years of lifetime* x Σ *(*CO_2 *kg/kWh* x *operating energy kWh)*$_{annual}$

(Operating energy includes cooling, heating, and backup heat. The calculation details are given the following session of Heat Pump Operating Energy Calculation)

CO_2 *emission due to energy consumption for components manufacturing =* Σ *(equivalent* CO_2 *kg/kg material* x *mass of materials kg)*

(Refrigerant manufacturing included)

CO_2 *emission due to energy consumption for components EOL=* Σ *(equivalent* CO_2 *kg/kg material* x *mass of recycled materials kg)*

(Refrigerant EOL included)

Heat Pump Operating Energy Calculation

(The method and equations are based on AHRI standard 210/240)

When T_j (outdoor dry bulb temperature, ^oF) is lower than 65^{o}F, the heat pump runs in heating mode; when it is equal to or higher than 65^{o}F, the heat pump runs in cooling mode.

Cooling

Building Cooling Load:

$$BL_c(T_j) = \frac{(T_j - 65)}{95 - 65} \cdot \frac{\dot{Q}_c(95)}{1.1} \qquad (1)$$

Where T_j is outdoor dry bulb temperature (^oF), $Q_c(95)$ is unit cooling capacity tested at outdoor 95^{o}F (dry bulb)/75^{o}F (web bulb) and indoor 80^{o}F (dry bulb)/67^{o}F (wet bulb) and at high capacity (for two capacity units).

$$Annual\ Cooling\ Energy = \sum_j E_{Cooling}(T_j) \qquad (2)$$

Where $E_{cooling}(T_j)$ is actual cooling energy consumption in an hour at temperature T_j, and its calculation is described as follows

1) Single speed
    a. <u>When unit cooling capacity is higher than building cooling load</u>

$$E_{Cooling}(T_j) = \frac{\left(\frac{BL_c(T_j)}{\dot{Q}_c(T_j)}\right) \cdot \dot{E}_c(T_j)}{PLF_j} \qquad (3)$$

$$PLF_j = 1 - C_D{}^c\left(1 - \frac{BL(T_j)}{\dot{Q}_c(T_j)}\right) \qquad (4)$$

$$\dot{Q}_c(T_j) = \dot{Q}_c(82) + \frac{\dot{Q}_c(95) - \dot{Q}_c(82)}{95 - 82} \cdot (T_j - 82) \qquad (5)$$

$$\dot{E}_c(T_j) = \dot{E}_c(82) + \frac{\dot{E}_c(95) - \dot{E}_c(82)}{95 - 82} \cdot (T_j - 82) \qquad (6)$$

Where Q_c, E_c, and PLF are unit cooling capacity, unit energy consumption and part load factor, respectively. $Q_c(82)$, $Q_c(95)$, $E_c(82)$, and $E_c(95)$ are cooling capacity and energy consumption data tested at 82^{o}F and 95^{o}F (outdoor dry bulb temperature), respectively. $C_d{}^c$ is cyclic-degradation coefficient with the default value of 0.25.

    b. <u>When unit cooling capacity is equal to or lower than building cooling load</u>

$$E_{Cooling}(T_j) = \dot{E}_c(T_j) \qquad (7)$$

Where E_c is unit energy consumption and is calculated using Equation (6).

(If the user has custom data instead of standard test data, the data input will be curve-fitted to calculate Q_c and E_c)

2) Two capacity unit

a. When building cooling load is equal to or lower than low capacity

The $E_{Cooling}$, and PLF are calculated using Equations (3) and (4), respectively, but the unit cooling capacity and energy consumption are calculated using the following equations:

$$\dot{Q}_c(T_j) = \dot{Q}_c(67) + \frac{\dot{Q}_c(82) - \dot{Q}_c(67)}{82 - 67} \cdot (T_j - 67) \qquad (8)$$

$$\dot{E}_c(T_j) = \dot{E}_c(67) + \frac{\dot{E}_c(82) - \dot{E}_c(67)}{82 - 67} \cdot (T_j - 67) \qquad (9)$$

Where $Q_c(67)$, $Q_c(82)$, $E_c(67)$, and $E_c(82)$ are unit cooling capacity and energy consumption tested at 67°F and 82°F outdoor dry bulb, respectively, and all tested at low capacity mode.

b. When building cooling load is between low capacity and high capacity of the unit

$$E_{cooling}(T_j) = [X_1(T_j) \cdot \dot{E}_{c,1}(T_j) + X_2(T_j) \cdot \dot{E}_{c,2}(\dot{T}_j)] \qquad (10)$$

$$X_1(T_j) = \frac{\dot{Q}_{c,2}(T_j) - BL_c(T_j)}{\dot{Q}_{c,2}(T_j) - \dot{Q}_{c,1}(T_j)} \qquad (11)$$

$$X_2(T_j) = 1 - X_1(T_j) \qquad (12)$$

Where $Q_{c,1}$ and $E_{c,1}$ are unit cooling capacity and energy consumption, respectively, calculated using Equation (8) and (9) with $Q_c(82)$, $Q_c(67)$, $E_c(82)$, and $E_c(67)$ tested at low capacity mode; $Q_{c,2}$ and $E_{c,2}$ are calculated using Equation (5) and (6) with $Q_c(95)$, $Q_c(82)$, $E_c(95)$ and $E_c(82)$ tested at high capacity mode.

The above method assumes the unit alternates between high and low compressor capacity to satisfy the building load (this method is used in the current computer program). If the unit locks out low compressor capacity operation, then Equation (3) to (6) should be used to calculate cooling energy with $Q_c(95)$, $Q_c(82)$, $E_c(95)$ and $E_c(82)$ tested at high capacity mode.

c. When building cooling load is equal to or higher than high capacity

Unit energy consumption is calculated using Equations (7) and (6) with $E_c(82)$ and $E_c(95)$ tested at high capacity mode.

(If the user has custom data instead of standard test data, the data input will be curve-fitted to do calculation)

3) Variable speed unit

a. <u>When building cooling load is equal to or lower than unit capacity at minimum compressor speed</u>

Equation (3), (4), (8), and (9) are used to calculate cooling energy consumption with $Q_c(67)$, $Q_c(82)$, $E_c(67)$, and $E_c(82)$ tested at minimum compressor speed.

b. <u>Unit operates at an intermediate compressor speed to match the building load</u>

$$E_{Cooling}\left(T_j\right) = \frac{BL_c(T_j)}{EER_i(T_j)} \qquad (13)$$

$$EER_i\left(T_j\right) = A + B \cdot T_j + C \cdot {T_j}^2 \qquad (14)$$

Where,

$$D = \frac{{T_2}^2 - {T_1}^2}{{T_v}^2 - {T_1}^2}$$

$$B = \frac{EER_1(T_1) - EER_2(T_2) - D \cdot [EER_1(T_1) - EER_v(T_v)]}{T_1 - T_2 - D \cdot (T_1 - T_v)}$$

$$C = \frac{EER_1(T_1) - EER_2(T_2) - B \cdot (T_1 - T_2)}{{T_1}^2 - {T_2}^2}$$

$$A = EER_2(T_2) - B \cdot T_2 - C \cdot {T_2}^2$$

T_1 = the outdoor temperature at which the unit, when operating at minimum compressor speed, provides a space cooling capacity that is equal to the building load ($Q_{c,1}(T_1) = BL_c(T_1)$). Determine T_1 by equating Equations (8) and (1).

T_2 = the outdoor temperature at which the unit, when operating at maximum compressor speed, provides a space cooling capacity that is equal to the building load ($Q_{c,2}(T_2) = BL_c(T_2)$). Determine T_2 by equating Equations (5) and (1).

T_v = the outdoor temperature at which the unit, when operating at intermediate compressor speed used during intermediate speed test, provides a space cooling capacity that is equal to the building load ($Q_{c,v}(T_v) = BL_c(T_v)$). Determine T_v by equating Equations (1) and (17).

$$EER_1(T_1) = \frac{Q_{c,1}(T_1) \quad \left[Eqn. (8), substituting\ T_1\ for\ T_j\right]}{E_{c,1}(T_1) \quad \left[Eqn. (9), Substtuting\ T_1 for\ T_j\right]} \qquad (15)$$

$$EER_v(T_v) = \frac{Q_{c,v}(T_v) \quad \left[Eqn. (17), substituting\ T_v\ for\ T_j\right]}{E_{c,v}(T_v) \quad \left[Eqn. (18), Substtuting\ T_v for\ T_j\right]} \qquad (16)$$

$$Q_{c,v}(T_j) = Q_{c,v}(87) + M_Q(T_j - 87) \qquad (17)$$

$$E_{c,v}(T_j) = E_{c,v}(87) + M_E(T_j - 87) \qquad (18)$$

Where $Q_{c,v}(87)$ and $E_{c,v}(87)$ are determined from intermediate speed test at 87°F.

$$M_Q = \left[\frac{Q_{c,1}(82) - Q_{c,1}(67)}{82 - 67} \cdot (1 - N_Q)\right] + \left[N_Q \cdot \frac{Q_{c,2}(95) - Q_{c,2}(82)}{95 - 82}\right]$$

$$M_E = \left[\frac{E_{c,1}(82) - E_{c,1}(67)}{82 - 67} \cdot (1 - N_E)\right] + \left[N_E \cdot \frac{E_{c,2}(95) - E_{c,2}(82)}{95 - 82}\right]$$

Where, $Q_{c,1}(82)$, $Q_{c,1}(67)$, $E_{c,1}(82)$, and $E_{c,1}(67)$ are data tested at minimum speed, and $Q_{c,2}(95)$, $Q_{c,2}(82)$, $E_{c,2}(95)$ and $E_{c,2}(82)$ are data tested at maximum speed.

$$N_Q = \frac{Q_{c,v}(87) - Q_{c,1}(87)}{Q_{c,2}(87) - Q_{c,1}(87)}$$

$$N_E = \frac{E_{c,v}(87) - E_{c,1}(87)}{E_{c,2}(87) - E_{c,1}(87)}$$

Use Equations (8) and (9) for T_j = 87°F to determine $Q_{c,1}(87)$ and $E_{c,1}(87)$, respectively.
Use Equations (5) and (6) for T_j = 87°F to determine $Q_{c,2}(87)$ and $E_{c,2}(87)$, respectively.

c. <u>When building cooling load is equal to or higher than unit capacity at maximum compressor speed</u>
   Unit energy consumption is calculated using Equations (7) and (6) with $E_c(82)$ and $E_c(95)$ tested at maximum compressor speed
   .

Heating

Building heating load:

$$BL_h(T_j) = \frac{(65 - T_j)}{65 - T_{OD}} \cdot C \cdot DHR \qquad (19)$$

Where T_j is outdoor dry bulb temperature (°F), and T_{OD} is the outdoor design temperature (°F) specified as follows:

Heating Climate Region	I	II	III	IV	V	VI
T_{OD}	37	27	17	5	-10	30

C = 0.77, a correction factor which tends to improve the agreement between calculated and measured building loads, dimensionless. DHR = the design heating requirement, Btu/h.
Calculate the design heating requirements for each generalized climatic region as follows:

$$DHR_{min} = \begin{Bmatrix} \dot{Q}_h^k(47) \cdot \left[\frac{65 - T_{OD}}{60}\right], \text{ for Regions I, II, III, IV, \& VI} \\ \\ \dot{Q}_h^k(47), \qquad\qquad \text{for Region V} \end{Bmatrix} \quad (20a)$$

$$DHR_{max} = \begin{Bmatrix} 2 \cdot \dot{Q}_h^k(47) \cdot \left[\frac{65 - T_{OD}}{60}\right], \text{ for Regions I, II, III, IV, \& VI} \\ \\ 2.2 \cdot \dot{Q}_h^k(47), \qquad\qquad \text{for Region V} \end{Bmatrix} \quad (20b)$$

Where $Q_h^k\ (47)$ is expressed in units of Btu/h and defined as unit heating capacity tested at outdoor 47°F (dry bulb)/43°F (wet bulb) and indoor 70°F (dry bulb)/60°F (wet bulb) and at high capacity (for two capacity units).

$$Annual\ Heating\ Energy = \sum_j \left[E_{heating}(T_j) + RH(T_j)\right] \quad (21)$$

Where $E_{heating}(T_j)$ is actual heat pump energy consumption in an hour at temperature T_j, and $RH(T_j)$ is backup resistive heating when heat pump capacity is lower than building heating load, or when the heat pump automatically turns off at the lowest outdoor temperatures (assume electric heating is used for backup heat. If other sources such as natural gas combustion are used for backup heat, $RH(T_j)$ will not be included in the heat pump power consumption, but it will be counted as heating energy for other heating sources). Their calculation is described as follows.

1) Single speed
   a. <u>When unit heating capacity is higher than building heating load</u>

$$E_{Heating}(T_j) = \frac{\left(\frac{BL_h(T_j)}{\dot{Q}_h(T_j)}\right) \cdot \dot{E}_h(T_j) \cdot \delta(T_j)}{PLF(T_j)} \quad (22)$$

$$PLF(T_j) = 1 - C_D{}^h \cdot \left(1 - \frac{BL_h(T_j)}{\dot{Q}_h(T_j)}\right) \quad (23)$$

$$RH(T_j) = \frac{BL_h(T_j)[1 - \delta(T_j)]}{3.4123 \ \frac{Btu/_h}{W}} \qquad (24)$$

$$\dot{Q}_h(T_j) = \begin{cases} \dot{Q}_h(17) + \frac{[\dot{Q}_h(47) - \dot{Q}_h(17)] \cdot (T_j - 17)}{47 - 17}, if\ T_j \geq 45°F\ or\ T_j \leq 17°F \\ \dot{Q}_h(17) + \frac{[\dot{Q}_h(35) - \dot{Q}_h(17)] \cdot (T_j - 17)}{35 - 17}, if\ 17°F < T_j < 45°F \end{cases} \qquad (25)$$

$$\dot{E}_h(T_j) = \begin{cases} \dot{E}_h(17) + \frac{[\dot{E}_h(47) - \dot{E}_h(17)] \cdot (T_j - 17)}{47 - 17}, if\ T_j \geq 45°F\ or\ T_j \leq 17°F \\ \dot{E}_h(17) + \frac{[\dot{E}_h(35) - \dot{E}_h(17)] \cdot (T_j - 17)}{35 - 17}, if\ 17°F < T_j < 45°F \end{cases} \qquad (26)$$

Where Q_h, E_h, δ, and *PLF* are unit heating capacity, energy consumption, low temperature cut-out factor and part load factor, respectively. $Q_h(17)$, $Q_h(35)$, $Q_h(47)$, $E_h(17)$, $E_h(35)$, and $E_h(47)$ are heating capacity and energy consumption data tested at 17°F, 35°F and 47°F (outdoor dry bulb temperature), respectively. C_d^h is cyclic-degradation coefficient with the default value of 0.25.

b. <u>When unit heating capacity is lower than building heating load</u>

$$E_{Heating}(T_j) = \dot{E}_h(T_j) \cdot \delta(T_j) \qquad (27)$$

$$RH(T_j) = \frac{BL_h(T_j) - [Q_h \cdot \delta(T_j)]}{3.4123 \ \frac{Btu/_h}{W}} \qquad (28)$$

Where Q_h and E_h are calculated using Equation (25) and (26), respectively.

(If the user has his custom data instead of standard test data, the data input will be curve-fitted to calculate Q_h and E_h)

2) Two capacity unit

a. <u>When building heating load is equal to or lower than low heating capacity</u>

The $E_{heating}$, *PLF,* and *RH* are calculated using Equation (22), (23), and (24), respectively, but the unit heating capacity and energy consumption are calculated using the following equations:

$$\dot{Q}_h(T_j) = \begin{cases} \dot{Q}_h(47) + \dfrac{[\dot{Q}_h(62) - \dot{Q}_h(47)] \cdot (T_j - 47)}{62 - 47}, if\ T_j \geq 40°\text{F} \\ \dot{Q}_h(17) + \dfrac{[\dot{Q}_h(35) - \dot{Q}_h(17)] \cdot (T_j - 17)}{35 - 17}, if\ 17°\text{F} \leq T_j \leq 40°\text{F} \\ \dot{Q}_h(17) + \dfrac{[\dot{Q}_h(47) - \dot{Q}_h(17)] \cdot (T_j - 17)}{47 - 17}, if\ T_j < 17°\text{F} \end{cases} \quad (29)$$

$$\dot{E}_h = \begin{cases} \dot{E}_h(47) + \dfrac{[\dot{E}_h(62) - \dot{E}_h(47)] \cdot (T_j - 47)}{62 - 47}, if\ T_j \geq 40°\text{F} \\ \dot{E}_h(17) + \dfrac{[\dot{E}_h(35) - \dot{E}_h(17)] \cdot (T_j - 17)}{35 - 17}, if\ 17°\text{F} \leq T_j < 40°\text{F} \\ \dot{E}_h(17) + \dfrac{[\dot{E}_h(47) - \dot{E}_h(17)] \cdot (T_j - 17)}{47 - 17}, if\ \ T_j < 17°\text{F} \end{cases} \quad (30)$$

Where $Q_h(62)$, $Q_h(47)$, $Q_h(35)$, $Q_h(17)$, $E_h(62)$, $E_h(47)$, $E_h(35)$, and $E_h(17)$ are unit heating capacity and energy consumption data tested at 62, 47, 35, and 17°F, respectively, and all tested at low capacity mode.

b. <u>When building heating load is between low capacity and high capacity of the unit</u>

$RH(T_j)$ is calculated using Equation (24).

$$E_{heating}(T_j) = [X_1(T_j) \cdot \dot{E}_{h,1}(T_j) + X_2(T_j) \cdot \dot{E}_{h,2}(\dot{T}_j)] \cdot \delta(T_j) \quad (31)$$

$$X_1(T_j) = \frac{\dot{Q}_{h,2}(T_j) - BL_h(T_j)}{\dot{Q}_{h,2}(T_j) - \dot{Q}_{h,1}(T_j)} \quad (32)$$

$$X_2(T_j) = 1 - X_1(T_j) \quad (33)$$

Where $Q_{h,1}$ and $E_{h,1}$ are unit heating capacity and energy consumption, respectively, calculated using Equations (29) and (30) with $Q_h(62)$, $Q_h(47)$, $Q_h(35)$, $Q_h(17)$, $E_h(62)$, $E_h(47)$, $E_h(35)$, and $E_h(17)$ tested at low capacity mode; $Q_{h,2}$ and $E_{h,2}$ are calculated using Equations (25) and (26) with $Q_h(47)$, $Q_h(35)$, $Q_h(17)$, $E_c(47)$, $E_c(35)$, and $E_c(17)$ tested at high capacity mode.

The above method assumes the unit alternates between high and low compressor capacity to satisfy the building load (this method is used in the current computer program). If the unit locks out low compressor capacity operation, then Equation (22) to (26) should be used to calculate heating energy with $Q_h(17)$, $Q_h(35)$, $Q_h(47)$, $E_h(17)$, $E_h(35)$, and $E_h(47)$ tested at high capacity mode.

c. <u>When building heating load is higher than unit high capacity</u>

Heat pump unit energy consumption is calculated using Equation (27), (28), (25), and (26) with $Q_h(47)$, $Q_h(35)$, $Q_h(17)$, $E_h(47)$, $E_h(35)$ and $E_h(17)$ tested at high capacity mode.

(If the user has custom data instead of standard test data, the input data will be curve-fitted to calculate Q_h and E_h)

3) Variable speed unit

   a. <u>When building heating load is equal to or lower than heating capacity at minimum compressor speed</u>

      Equations (22), (23), (24), (39) and (40) are used with the $Q_h(62)$, $Q_h(47)$, $E_h(62)$, and $E_h(47)$ tested at minimum compressor speed.

   b. <u>Heat pump operates at an intermediate compressor speed to match the building load</u>

$$E_{heating}(T_j) = \frac{BL_h(T_j) \cdot \delta(T_j)}{3.4123 \frac{Btu/h}{W} \cdot COP_i(T_j)} \tag{34}$$

$$COP_i(T_j) = A + B \cdot T_j + C \cdot T_j^2 \tag{35}$$

Where,

$$D = \frac{T_3^2 - T_4^2}{T_{vh}^2 - T_4^2}$$

$$B = \frac{COP_2(T_4) - COP_1(T_3) - D \cdot [COP_2(T_4) - COP_v(T_{vh})]}{T_4 - T_3 - D \cdot (T_4 - T_{vh})}$$

$$C = \frac{COP_2(T_4) - COP_1(T_3) - B \cdot (T_4 - T_3)}{T_4^2 - T_3^2}$$

$$A = COP_2(T_4) - B \cdot T_4 - C \cdot T_4^2$$

T_3 = the outlet temperature at which the heat pump, when operating at minimum compressor speed, provides a space heating capacity that is equal to the building load ($Q_{h,1}(T_3) = BL_h(T_3)$). Determine T_3 by equating Equations (39) and (19).

T_{vh} = the outlet temperature at which the heat pump, when operating at the intermediate compressor speed during the intermediate speed test , provides a space heating capacity that is equal to the building load ($Q_{h,v}(T_{vh}) = BL_h(T_{vh})$). Determine T_{vh} by equating Equations (41) and (19).

T_4 = the outlet temperature at which the heat pump, when operating at maximum compressor speed, provides a space heating capacity that is equal to the building load ($Q_{h,2}(T_4) = BL_h(T_4)$). Determine T_4 by equating Equations (25) (using maximum speed data) and (19).

$$COP_1(T_3) = \frac{Q_{h,1}(T_3)[Eqn. 39, substituting\ T_3\ for\ T_j]}{3.4123 \frac{Btu/h}{W} \cdot E_{h,1}(T_3)[Eqn. 40, substituting\ T_3\ for\ T_j]} \quad (36)$$

$$COP_v(T_{vh}) = \frac{Q_{h,v}(T_{vh})[Eqn. 41, substituting\ T_{vh}\ for\ T_j]}{3.4123 \frac{\frac{Btu}{h}}{W} \cdot E_{h,v}(T_{vh})[Eqn. 42, substituting\ T_{vh}\ for\ T_j]} \quad (37)$$

$$COP_2(T_4) = \frac{Q_{h,2}(T_4)[Eqn. 25, substituting\ T_4\ for\ T_j]}{3.4123 \frac{Btu/h}{W} \cdot E_{h,2}(T_4)[Eqn. 26, substituting\ T_4\ for\ T_j]} \quad (38)$$

$$Q_{h,1}(T_j) = Q_{h,1}(47) + \frac{Q_{h,1}(62) - Q_{h,1}(47)}{62 - 47} \cdot (T_j - 47) \quad (39)$$

$$E_{h,1}(T_j) = E_{h,1}(47) + \frac{E_{h,1}(62) - E_{h,1}(47)}{62 - 47} \cdot (T_j - 47) \quad (40)$$

$$Q_{h,v}(T_j) = Q_{h,v}(35) + M_Q \cdot (T_j - 35) \quad (41)$$

$$E_{h,v}(T_j) = E_{h,v}(35) + M_E \cdot (T_j - 35) \quad (42)$$

Where $Q_{h,v}$ and $E_{h,v}$ are determined from intermediate speed test at 35°F.

$$M_Q = \left[\frac{Q_{h,1}(62) - Q_{h,1}(47)}{62 - 47} \cdot (1 - N_Q)\right] + \left[N_Q \cdot \frac{Q_{h,2}(35) - Q_{h,2}(17)}{35 - 17}\right]$$

$$M_E = \left[\frac{E_{h,1}(62) - E_{h,1}(47)}{62 - 47} \cdot (1 - N_E)\right] + \left[N_E \cdot \frac{E_{h,2}(35) - E_{h,2}(17)}{35 - 17}\right]$$

Where, $Q_{h,1}(62)$, $Q_{h,1}(47)$, $E_{h,1}(62)$, and $E_{h,1}(47)$ are data tested at minimum speed, and $Q_{h,2}(35)$, $Q_{h,2}(17)$, $E_{h,2}(35)$ and $E_{h,2}(17)$ are data tested at maximum speed. If the $H2_2$ test is not conducted, one can use $Q_{h,2}(35) = 0.90 \cdot (Q_{h,2}(17) + 0.6 \cdot (Q_{h,2}(47) - Q_{h,2}(17)))$ and $E_{h,2}(35) = 0.985 \cdot (E_{h,2}(17) + 0.6 \cdot (E_{h,2}(47) - E_{h,2}(17)))$ to determine $Q_{h,2}(35)$ and $E_{h,2}(35)$.

$$N_Q = \frac{Q_{h,v}(35) - Q_{h,1}(35)}{Q_{h,2}(35) - Q_{h,1}(35)}$$

$$N_E = \frac{E_{h,v}(35) - E_{h,1}(35)}{E_{h,2}(35) - E_{h,1}(35)}$$

Use Equations (39) and (40) for T_j = 35°F to determine $Q_{h,1}(35)$ and $E_{h,1}(35)$, respectively.

c. <u>When building heating load is equal to or higher than heat pump capacity at maximum compressor speed</u>

Heat pump unit energy consumption is calculated using Equation (27), (28), (25), and (26) with $Q_h(47)$, $Q_h(35)$, $Q_h(17)$, $E_h(47)$, $E_h(35)$ and $E_h(17)$ tested at maximum compressor speed. If the $H2_2$ test is not conducted, one can use $Q_{h,2}(35) = 0.90 \cdot (Q_{h,2}(17) + 0.6 \cdot (Q_{h,2}(47) - Q_{h,2}(17)))$ and $E_{h,2}(35) = 0.985 \cdot (E_{h,2}(17) + 0.6 \cdot (E_{h,2}(47) - E_{h,2}(17)))$ to determine $Q_{h,2}(35)$ and $E_{h,2}(35)$.

Backup Heat

If the calculated unit heating capacity is smaller than the building heating load or if the heat pump is turned off for outdoor temperature is too low, the backup heat needs to be turned on. The $RH(T_j)$ part in the previous section is the needed backup heating energy.

The backup heat can be electric heating, oil combustion, and natural gas combustion. The indirect emission due to backup heat is calculated as

CO_2 emission for backup heat by electric heating = *(CO_2 kg/kWh* x *annual backup heat kWh)* x *years of lifetime*

CO_2 emission for backup heat by oil or natural gas = *(CO_2 kg/kg fuel* x *1/ (heating value kWh/kg fuel* x *efficiency)* x *annual backup heat kWh)* x *years of lifetime*

Simplified Method Using SEER/HSPF

A simplified method to estimate annual energy using SEER/HSPF factor is as follows:

$$E = \frac{CLH_A \cdot \dot{Q}_c(95)}{SEER} + \frac{HLH_A \cdot DHR \cdot C}{HSPF \cdot C_H}$$

Where CLH_A is the actual cooling hours for a particular location as determined using the map given in Figure 7; $Q_c(95)$ is cooling capacity determined from 95°F test; HLH_A is the actual heating hours for a particular location as determined using the map in Figure 8; DHR is the design heating requirement as defined in Equation (20); C is a constant of 0.77. Normally the HSPF is published for IV heating region, thus we use a correction factor C_H to adjust the HSPF for other regions. From the "ARI Guide for estimating annual operating cost of a central air conditioner or heat pump", the following correction factors are adopted:

Region	Factor (C_H)
I	1.209
II	1.153
III	1.095
IV	1.0

V	0.870
VI	1.187

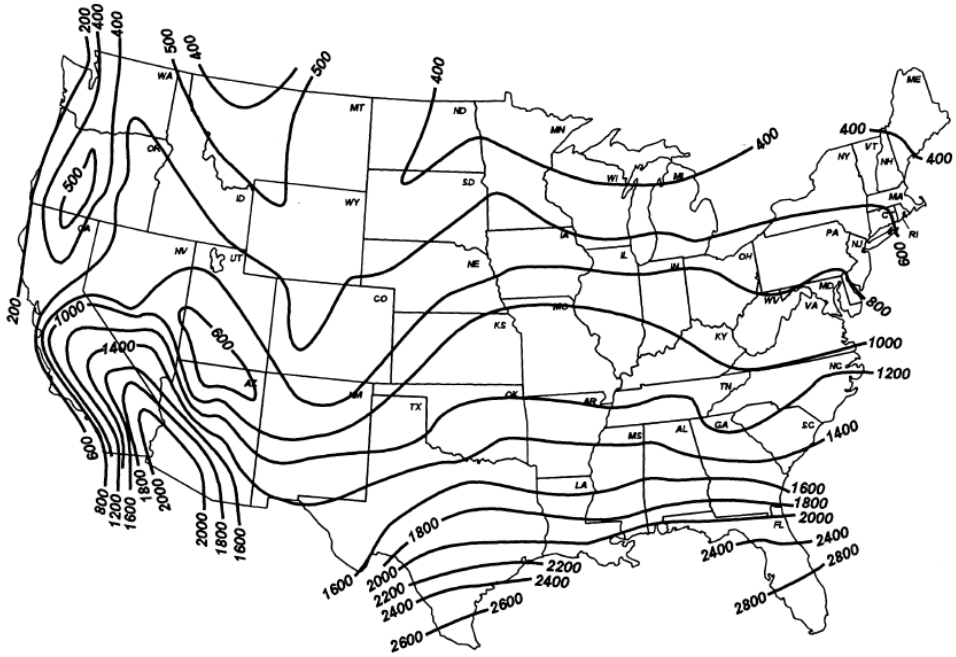

Figure 7: Cooling load hours (CLH_A) for the United States

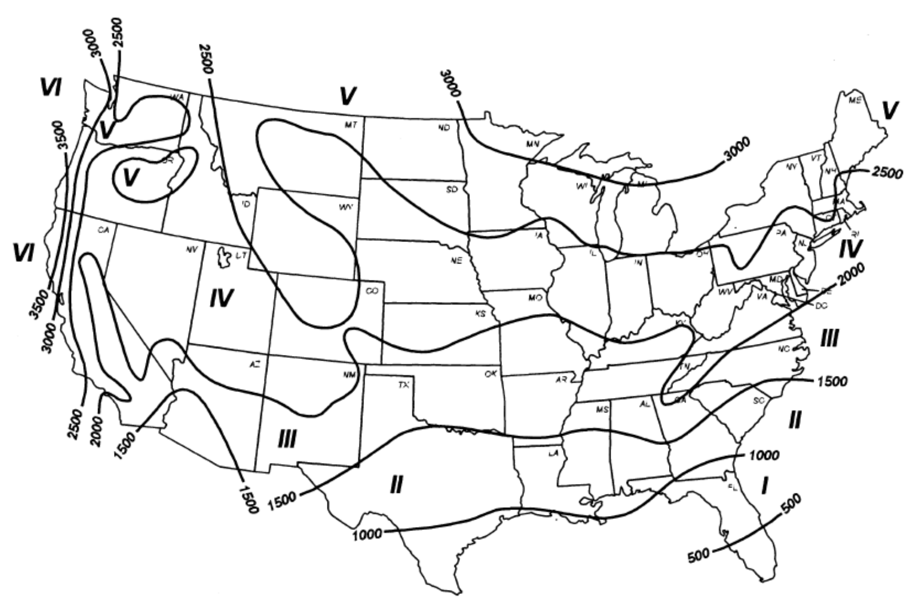

Figure 8: Heating Load Hours (HLH_A) for the United States