SIMULATION, MODELING AND ANALYSIS

OF A WATER TO AIR

HEAT PUMP

By

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SIMULATION, MODELING AND ANALYSIS

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TABLE OF CONTENTS

Chapter **Page**

LIST OF FIGURES

Figure **Page**

LIST OF TABLES

Table **Page**

NOMENCLATURE

PLR	=	Part load ratio(-)
QE	=	Evaporator cooling load- Btu/h (W)
QENOM	=	Nominal evaporator cooling load- Btu/h (W)
PNOM	=	Nominal Compressor power- Btu/h (W)
P	=	Compressor power- Btu/h (W)
ANCR	=	Available to nominal capacity ratio(-)
FFL	=	Fraction of full load power(-)
FLPR	=	Full load Power ratio(-)
PFL	=	Full load compressor power- Btu/h (W)
E	=	Energy rate- Btu/h (W)
f	=	Functional relationship
P	=	Pressure-psia(Pa)
$\dot{m}$	=	Mass flow rate-lbm/h(kg/s)
T	=	Temperature-^{o}F(^{o}C)
X	=	Refrigerant quality(-)
U	=	Energy flow rate- Btu/h (W)
R	=	gas constant(J/K.s)
PD	=	Piston displacement-CFM(m^3/s)
C	=	Clearance factor(-)
P_{dis}	=	Discharge pressure- psia(Pa)
P_{suc}	=	Suction pressure- psia(Pa)
γ	=	Isentropic exponent(-)

$h_{co}A_o$	=	External heat transfer coefficient-Btu/ ^oF(J/K)
$\dot{m}_a$	=	Air mass flow rate-lbm/h(kg/s)
Cp_a	=	Specific heat of air-Btu/lbm- ^oF(J/kg- K)
$\dot{Q}_s$	=	Source side heat transfer rate- Btu/h (W)
$\dot{Q}_L$	=	Load side heat transfer rate- Btu/h (W)
$\dot{W}$	=	Compressor power input- Btu/h (W)
W_{cat}	=	Catalog power consumption- Btu/h (W)
W	=	Model power consumption- Btu/h (W)
QL_{cat}	=	Catalog load side heat transfer- Btu/h (W)
QL	=	Model load side heat transfer- Btu/h (W)
T_{WiL}	=	Entering water load side temperature-$^oF(^oC)$
T_{WiS}	=	Entering water Source side temperature-$^oF(^oC)$
$\dot{m}_{WiL}$	=	Entering water load side mass flow rate-$^oF(^oC)$
$\dot{m}_{WiS}$	=	Entering water Source side mass flow rate-$^oF(^oC)$
S	=	Thermostatic Signal
T_{ref}	=	Reference Temperature(K or ^oR)
T_{Win}	=	Entering water temperature (K or ^oR)
T_{db}	=	Dry bulb air temperatures (Kor ^oR)
T_{wb}	=	Wet bulb air temperatures (Kor ^oR)
Q_{base}	=	Base capacity of the heat pump unit(W)
Qc	=	Cooling capacity (W)
Qh	=	Heating capacity (W)
A	=	Polynomial regression coefficient(-)
B	=	Polynomial regression coefficient(-)

C	=	Polynomial regression coefficient(-)
CT	=	Polynomial regression coefficient(-)
LT	=	Polynomial regression coefficient(-)
QT	=	Polynomial regression coefficient(-)
$A2 - F2$	=	Equation fit coefficients for the heating mode(-)
$A1 - J1$	=	Equation fit coefficients for the cooling mode(-)

CHAPTER 1

INTRODUCTION

1.1. Background

Water-to-air ground source heat pump systems have been an ideal choice for design engineers since they provide a promising eco-friendly alternative for heating and cooling of residential and commercial buildings. These units accept energy from or reject energy to a common ground loop depending on whether the zone has a requirement for heating or cooling. The single package reverse cycle water-to-air heat pump uses ground as the heat source in the heating mode. In the cooling mode, the ground acts as the heat sink by the application of a refrigerant reversing valve. An expansion device maintains the pressure difference between the high pressure condenser side and the low pressure evaporator sides of the water-to-air heat pump refrigeration system. In heating mode, the refrigerant under high pressure is directed through the refrigerant to air heat exchanger and the closed loop heat exchanger absorbs heat from the ground (typically 55F to 70F). Heat transfers to the refrigerant via the water to refrigerant heat exchanger and the condensation of the refrigerant results in heating. In the cooling mode, the high pressure refrigerant is channeled through the water to refrigerant heat exchanger connected to the ground loop and cooling is provided by the evaporation of the refrigerant in the refrigerant to air heat exchanger.

With increasing energy costs, many potential heat pump applications require frequent reassessment in terms of design and modeling. Modeling heat pumps for design and simulation is significant since it allows cost effective design solutions and permits the designer to quantitatively compare a variety of design strategies. Also, it facilitates modifications at any stage in the design process before the final documents are produced.

1.2.Thesis objective and scope

The objective of the thesis is to develop a simplified model that can simulate a water-to-air ground source heat pump and implement the model in EnergyPlus. EnergyPlus (Crawley et al; 1997) is a new energy analysis and thermal simulation engine capable of performing sub-hourly simulation of the building, fan system and the ground source heat pump system. It is a highly modularized, platform independent engine. It is developer friendly and is based on the best features of BLAST (BSO, 1991) and DOE-2 (LBNL, 1980). The steady state behavior and performance characteristics of the water loop heat pump system developed by Lash (1990) for the BLAST program have been investigated. The model which includes a ground loop configuration is significantly modified for implementation in the EnergyPlus program. Also, the model is extended to calculate sensible and latent capacity splits for the heat pump while operating in the cooling mode. The performance of the simplified model is compared with the parameter estimation water-to-air heat pump model (Jin 1999) to determine the usefulness of implementing a simple model. The simplified and the detailed models are compared in a case study and the results are summarized and analyzed. The performance data generated using the model is compared with the manufacturer's catalog data to check the validity of the simulation. A visual basic graphical user interface that generates performance coefficients from manufacture's data was also developed.

CHAPTER 2

LITERATURE REVIEW

One of the biggest challenges facing the simulation and design of performance oriented ground source heat pump systems is the degree of complexity involved in modeling the individual components of the system. The challenge also impinges on the successful implementation of the model in a computer program. Several heat pump models have been implemented in the past. However a detailed review of the literature uncovers a few limitations in the existing models.

2.1.Equation fit water-to-air heat pump and chiller models

2.1.1. Lash model

Lash(1990) developed a model for a water loop heat pump system(WLHPS) in which a network of zoned reversible packaged water source heat pumps operate independently of each other to control individual zone loads. These units are capable of supplying both heating and cooling. Fig 2.1 is an illustration of the simplified loop model used by Lash. The water loop is modeled by breaking up the loop into two sections or nodes where node1 represents the water mass between the central plant outlet and the first heat pump inlet and node2 represents the water mass between the first heat pump inlet to the central plant inlet. The Lash water loop heat pump model as implemented in the BLAST energy analysis program uses four non-dimensional performance equations to describe the heat pump as discussed in chapter 4.

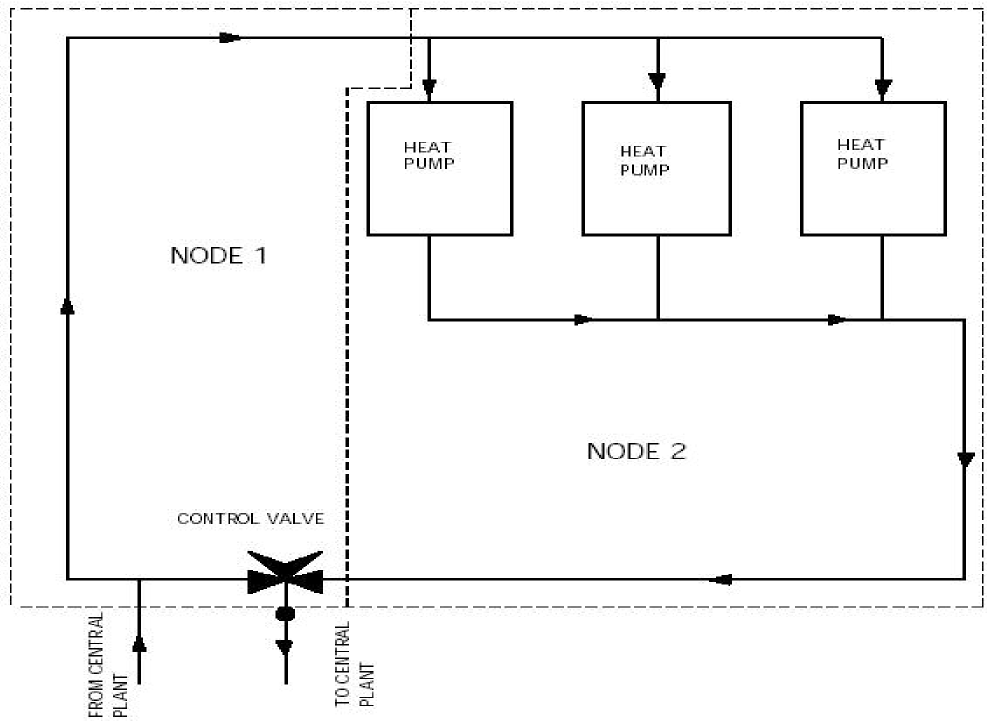

Figure 2.1. *Heat pump system nodes (Lash 1990)*

The Heat pump performance is based on entering air temperatures, entering water temperatures and the inlet mass flow rate of water. Coefficients for the non-dimensional equations are obtained by a least square fit of manufacturer's data to the form of the equation. The biggest advantage of the model is that the data is readily available. Also, since refrigeration properties are not required the model is very robust, resulting in quick execution time. However the model is not applicable beyond a particular data range and it also fails to account for the latent and sensible capacity splits in the cooling mode.

2.1.2. Allen and Hamilton model

Allen and Hamilton (1983) developed a steady state reciprocating compressor water chiller model. Fig 2.2 shows a schematic of the water chiller with the basic pieces of equipment and the variables labeled at appropriate points.

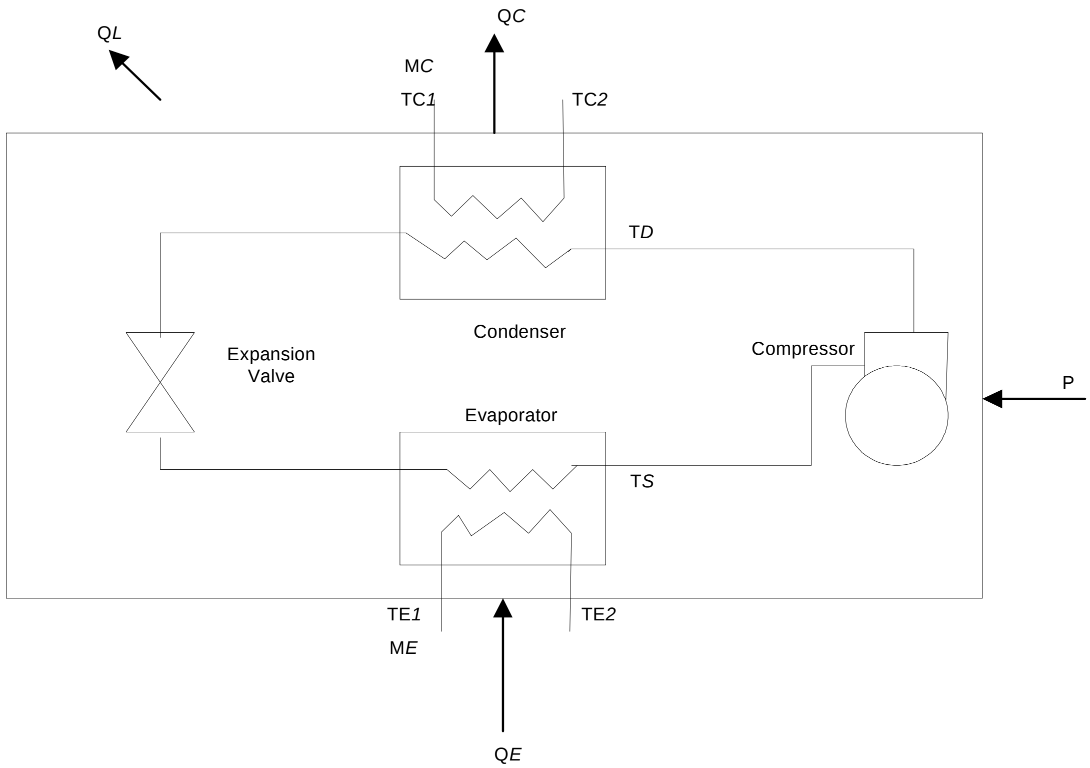

Figure 2.2. *Water chiller system schematic*

The equation fit model is capable of simulating the energy rate characteristics of the system under full load and part load conditions. The evaporator heat transfer rate and the compressor power are expressed in terms of a polynomial which is a function of the evaporator and condenser exit temperatures. The coefficients are determined from experimental data by polynomial regression. Given a water chiller, an evaporator inlet water temperature, $TE_{1,}$ an evaporator water-side mass flow rate, $M_{E,}$ a condenser inlet water temperature, $TC_{1,}$ and a condenser water-side mass flow rate, M_c, then the five system equations can be given as follows.

The evaporator heat transfer rate is given by

$Q_E = \dot{m}_E C_p (TE_2 - TE_1)$ …………………………………………..(2.1)

$Q_E = b_1 T_{E2} + b_2 T_{C2} + b_3 T_{E2} T_{C2} + b_4 T_{E2}^2 + b_5 T_{C2}^2 + b_6$ ………..……. (2.2)

And the compressor power is

$P = b_7 T_{E2} + b_8 T_{C2} + b_9 T_{E2} T_{C2} + b_{10} T_{E2}^2 + b_{11} T_{C2}^2 + b_{12}$ …………..…….(2.3)

Also, the condenser heating rejection rate is

$Q_C = Q_E + P$ …………………………………………….………...(2.4)

$Q_C = \dot{m}_c C_p (T_{C2} - T_{C1})$ …………………………………………..(2.5)

Where

b_1-b_{12} = Coefficients fitted by polynomial regression

$T_{E1}, T_{E2,}$ = Evaporator water entering and leaving temperature

$T_{C1,} T_{C2}$ = Condenser water entering and leaving temperature

T_s, T_d = Refrigerant temperature at compression suction and discharge

C_p = Specific heat at constant pressure

P = Compressor power

Q_E = Evaporator heat transfer rate

Q_C = Condenser heat transfer rate

$\dot{m}_E$ = Evaporator mass flow rate

$\dot{m}_C$ = Condenser mass flow rate

Although the model does not include the complexity of modeling individual components it is not a useful model due to the fact that its two main biquadratic equations are based on condenser and evaporator outlet temperatures which are not readily available from catalog data. In addition to this, the model neglects the power losses Q_L, shown in Figure 2.2, assuming well designed systems available from the major manufacturers.

2.1.3. DOE-2 Model

The water chiller model used in the DOE-2 computer simulation program (DOE-2 Engineers Manual, 2002) is the simplest model with a minimum number of performance coefficients. The DOE-2 model eliminates the dependency of water chiller performance on condenser and evaporator temperatures. Here, the power consumption is fit as a quadratic function of the evaporator heat transfer rate, Q_E.

The part load ratio is given as

$$PLR = \frac{Q_E}{Q_{ENOM}} \qquad (2.6)$$

$$\frac{P}{P_{Nom}} = C_T + L_T PLR + Q_T PLR^2 \qquad (2.7)$$

Where

PLR = Part load ratio

Q_E = Evaporator heat transfer rate

Q_{ENOM} = Nominal evaporator heat transfer rate

P_{NOM} = Nominal Compressor power

C_T, L_T, Q_T = Polynomial regression coefficients

P = Compressor power

Although the model exhibits simplicity with only two equations and four constants, it does not account for the effect of temperature variations on performance. This results in a large predictive error.

2.1.4. BLAST Model

The BLAST chiller model (BLAST Users Manual, 1991) slightly improves on the DOE-2 model. The model was actually developed to counter the demerits of the DOE-2 model. This model accounts for variations in performance due to condenser and evaporator temperatures. The model is based on the observation that under full load, the constant evaporator heat transfer rate lines are approximately linear and parallel. It also introduces a variable known as the equivalent temperature difference, δT, which is an expression involving the user input condenser and evaporator exit temperatures, TC_2 and TE_2, and the slope, k, of the constant evaporator heat transfer rate at full load performance. The available capacity to nominal capacity ratio, ANCR, is modeled as a quadratic function of δT. The water chiller power PFL is modeled as a function of ANCR.

$$\Delta T = \frac{(T_{C2} - T_{C2NOM})}{k} - (T_{E2} - T_{E2NOM}) \quad \text{..(2.8)}$$

$$ANCR = \frac{Q_{EFL}}{Q_{ENOM}} = b1 + b2.\Delta T + b3.\Delta T^2 \quad \text{..(2.9)}$$

$$FLPR = \frac{PFL}{Q_{EFL}} = \left(\frac{P_{NOM}}{Q_{ENOM}}\right)\left(c1 + c2.ANCR + c3.ANCR^2\right) \quad \text{....................(2.10)}$$

$$PLR = \frac{Q_E}{Q_{EFL}} \quad \text{..(2.11)}$$

$$FFL = \frac{P}{PFL} = a1 + a2.PLR + a3.PLR^2 \quad \text{..(2.12)}$$

$$P = FLPR * FFL * \frac{ANCR * Q_{ENOM}}{RatedCOP} \quad \text{..(2.13)}$$

Where

a1-a3, b1-b3, c1-c3 = Polynomial regression coefficients

ANCR = Available capacity to nominal capacity ratio

FFL = Fraction of full load power

FLPR = Full load Power ratio

PFL = Full load compressor power

PLR = Part load ratio

Q_{EFL} = Full load evaporator heat transfer rate

PLR = Part load ratio

Q_E = Evaporator heat transfer rate

Q_{ENOM} = Nominal evaporator heat transfer rate

P_{NOM} = Nominal Compressor power

P = Compressor power

2.1.5. Hamilton and Miller model

The Hamilton and Miller model (1990) is a typical detailed equation fit model. The steady state model incorporates functional fits of manufacturers' catalog performance data of various individual standard components including evaporators, compressors, condensers, capillary tubes and fans. The system model shown in Fig 2.3 is a combination of mathematical models of individual components with mass and energy flow at each component connection and conformity of pressure/temperature values at each component connection. The equations expressing the behavior and operation of each component are based on assumed steady state conditions. The components are connected such that the input to one component is the output of the previous component and the values of the state variables are updated by each component model. Each node is primarily characterized by the pressure and temperature of the refrigerant. Together the equations form a set of non-linear simultaneous equations depicting the response of the vapor compression system to fan inlet conditions. The computer model is valuable for simulating the response of air conditioning systems for a range of ambient and inside conditions. Although the model provides the flexibility to specify various coils, compressors and capillary tubes, it is important to notice that comprehensive individual component performance data is not readily available.

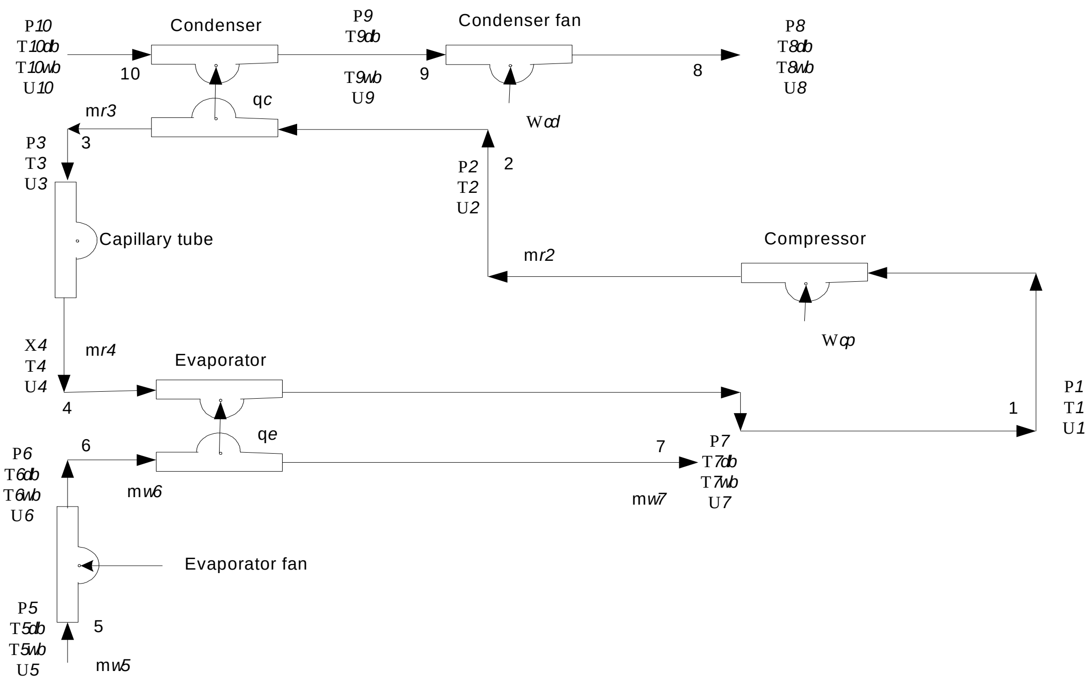

Figure 2.3. *Schematic of an air conditioning system showing connecting points between components*

2.1.6. Parent and Larue development of the Domanski Model

Parent and Larue(1989) developed a steady state water-to-air heat pump modeling program called SIMPAC which uses Domanski's (1986) model of an air-to-air heat pump as the starting point. The program involves a driver program that links the independent models of each system component together. Thermodynamic properties of pure refrigerants and zeotropic mixtures are evaluated by the application of Carnahan Starling Desantis (CSD) equation of state. The SIMPAC program requires detailed input for the individual components of the heat pump. The SIMPAC algorithm is computationally intensive and may not converge.

2.2. Jin and Spitler's parameter estimation water-to-air heat pump models

2.2.1 Overview

Jin and Spitler (2002) proposed a steady state simulation model for water-to-water and water-to-air heat pumps. The model uses a multivariable unconstrained optimization algorithm to estimate a number of parameters. The aim of the model is to determine the geometric parameters and operation of each component and replicate the performance of the actual unit in operation. The model algorithm and the equations are shown in Fig 2.4.

Where

T_{WiL} = Entering water Load side temperature

T_{WiS} = Entering water Source side temperature

$\dot{m}_{WiL}$ = Entering water Load side mass flow rate

$\dot{m}_{WiS}$ = Entering water Source side mass flow rate

T_c = Condenser temperature

T_e = Evaporator temperature

S = Thermostatic Signal

mwl = Load Mass flow rate

mws = Source Mass flow rate

QL = Load side Heat transfer rate

QS = Source side Heat transfer rate

The parameters included in the model are shown in Figure 2.5 where:

PD = Piston displacement

C = Clearance Factor

ΔP	=	Pressure drop across the suction and discharge valves
η	=	Loss factor
ΔT_{sh}	=	Superheat in ^{o}C or F
W_{loss}	=	Constant part of the electromechanical losses
$(UA)_S$	=	Source side heat transfer coefficient
$(UA)_L$	=	Load side heat transfer coefficient
(h_{co},A_o)	=	External heat transfer coefficient[Cooling mode only]

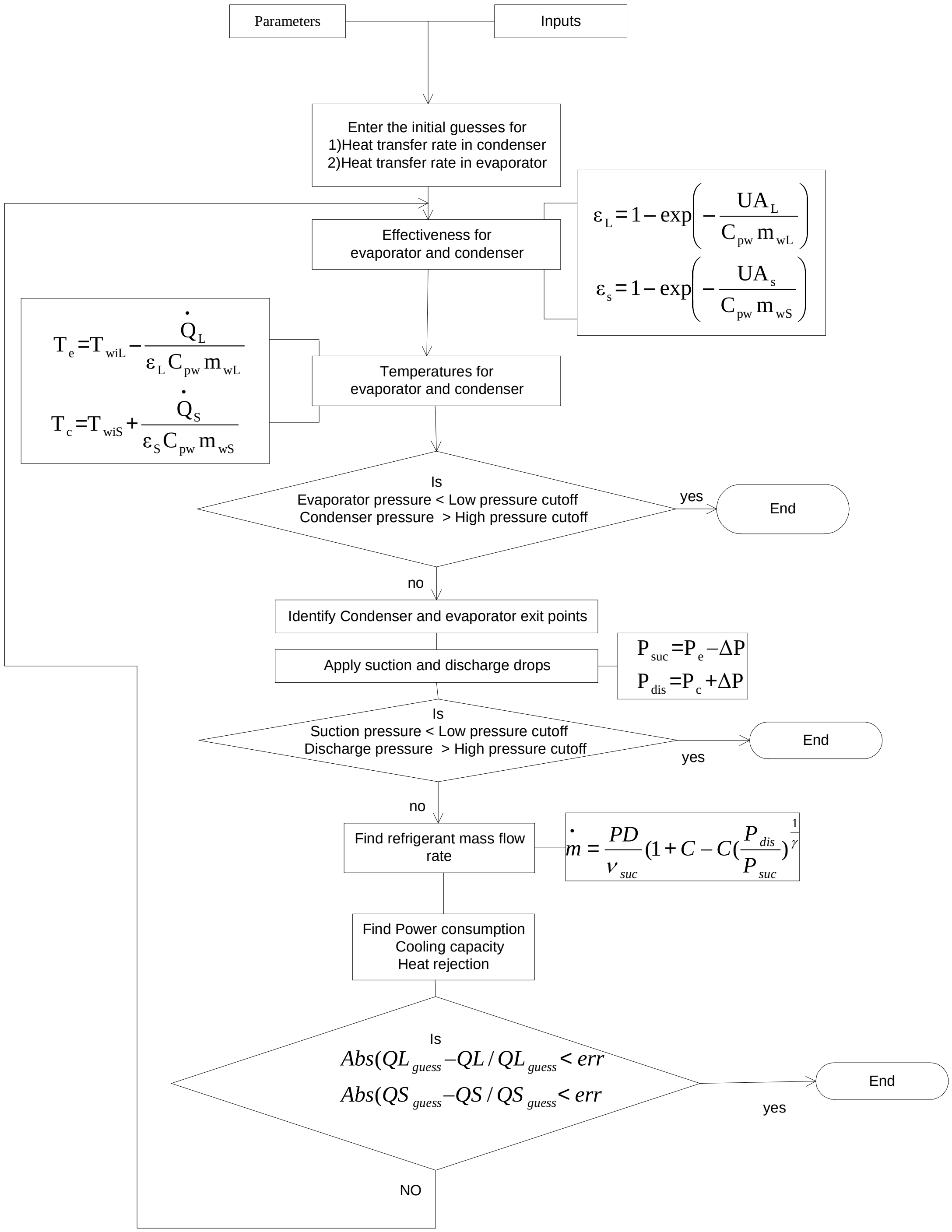

Figure 2.4. *Flow diagram for model implementation computer program*

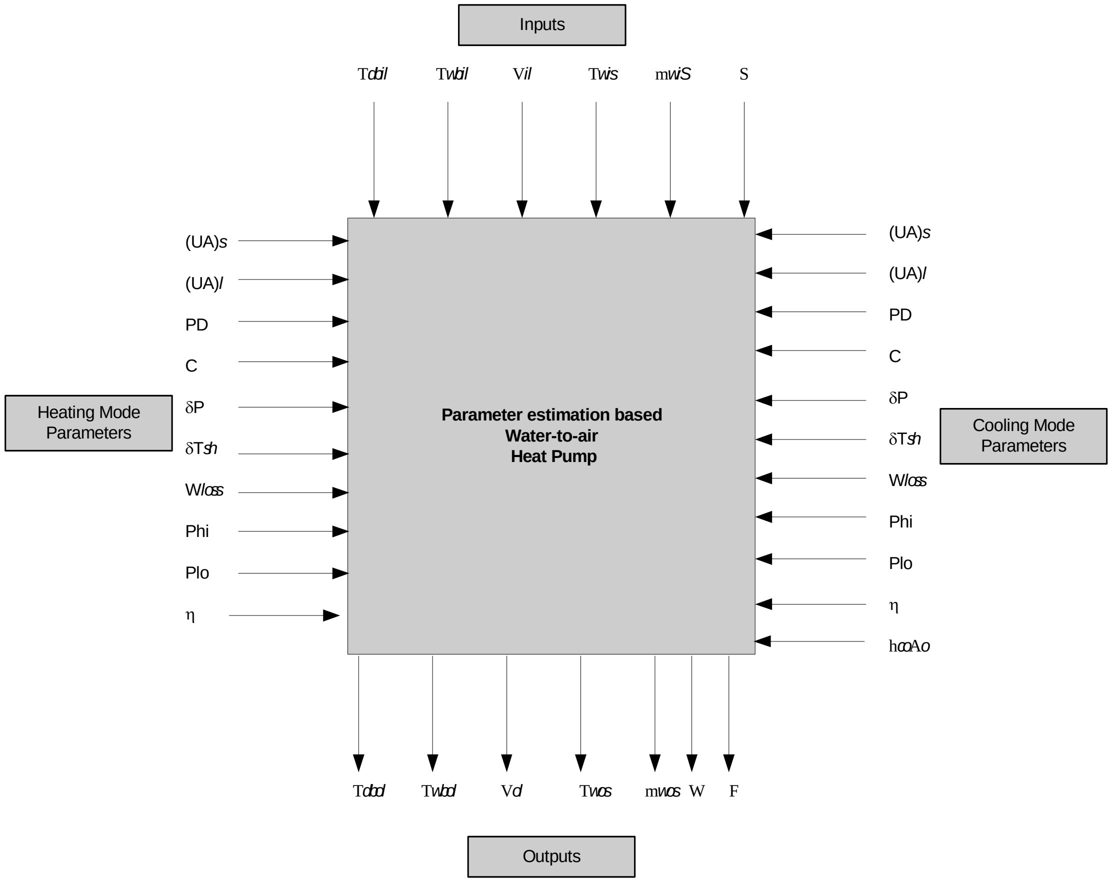

Figure 2.5. *Information flowchart for model implementation (Jin 2002)*

2.2.2. Parameter estimation procedure

A set of parameters for the cooling mode are defined on the basis of the equations used in the model. An information flowchart indicating the parameters, inputs to the model and the resulting outputs are shown in Fig 2.5. The estimation of parameters is conducted using the catalog data. The parameter estimation procedure incorporates an objective function which computes the difference between the model outputs and the catalog outputs. The objective function is then minimized by using a multi-variable, unconstrained, multi-modal, Nelder-Mead optimization algorithm. The inputs to the model include the entering water temperatures and mass flow rates on the load side and the source side. The sum of the square of the errors (SSQE) for a given set of parameter values which will be minimized is given by

$$SSQE = \sum_{i=1} \left(\frac{\left(W_{cat}\right) - \left(W\right)_i}{\left(W_{cat}\right)} \right)^2 + \left(\frac{\left(QL_{cat}\right) - \left(QL\right)_i}{\left(QL_{cat}\right)} \right)^2 \quad \text{..........................(2.23)}$$

Where

W_{cat}	=	Catalog power consumption- Btu/h (W)
W	=	Model power consumption- Btu/h (W)
QL_{cat}	=	Catalog load side heat transfer- Btu/h (W)
QL	=	Model load side heat transfer- Btu/h (W)
i	=	Number of points

2.2.3. Model implementation

A thermostat signal is used as an input parameter to tell the model which set of parameters (heating mode or cooling mode) should be used. Also, the objective function evaluation takes advantage of the fact that the heat transfer rates are known, using the catalog data as an initial guess, then minimizing the difference between the calculated and catalog heat transfer rates. However, for the model implementation, the heat transfer rates are obtained by simultaneous solution with successive substitution. An information flow chart of the model implementation is presented in Figure 2.5. Extrapolation beyond the catalog data grants the parameter estimation model an upper hand in comparison with the equation fit models. However, the model although advantageous, carries a high overhead cost associated with repeated calls to refrigerant data routines. This makes the process computationally more intensive and time consuming. In addition, refrigerant properties may go out of bounds while the model attempts to converge on the final solution. Running the model successfully in a general simulation environment requires considerable effort to hold the refrigerant properties within bounds for every iteration of the simulation. Finally the convergence on a valid set of parameters is largely dependent on the initial guesses. This places an undue burden on users who typically have little knowledge of appropriate ranges for the model parameters.

CHAPTER 3

OVERVIEW OF ENERGYPLUS

3.1.Introduction

EnergyPlus (Crawley et al; 1997) is a new energy analysis and thermal simulation engine developed in Fortran 90. It is capable of modeling HVAC systems, central plants, ideal controls and building heat transfer. It is a highly modularized, platform independent engine that was originally based on the best features of BLAST (BSO1991) and DOE-2 (LBNL1980). Program features include variable time step, user configurable modular systems and an integrated system / zone simulation. Recently, a number of components have been developed and implemented in the program to support ground source heat pump analysis. Rees(2002) implemented the shallow pond model developed by Chiasson(1999). Murugappan(2002) implemented the vertical ground loop heat exchanger model developed by Yavuzturk and Spitler(1999). Murugappan(2002) also implemented the water-to-water heat pump model developed by Jin and Spitler(2002). Most recently, the water-to-air heat pump model developed by Jin and Spitler has been implemented in EnergyPlus.

3.2.Simulation environment

Figure 3.1 shows the overall hierarchy of EnergyPlus. On the whole, the simulation environment is based on fundamental heat balance principles and is a simultaneous solution of the coupled building, system and the plant simulations.

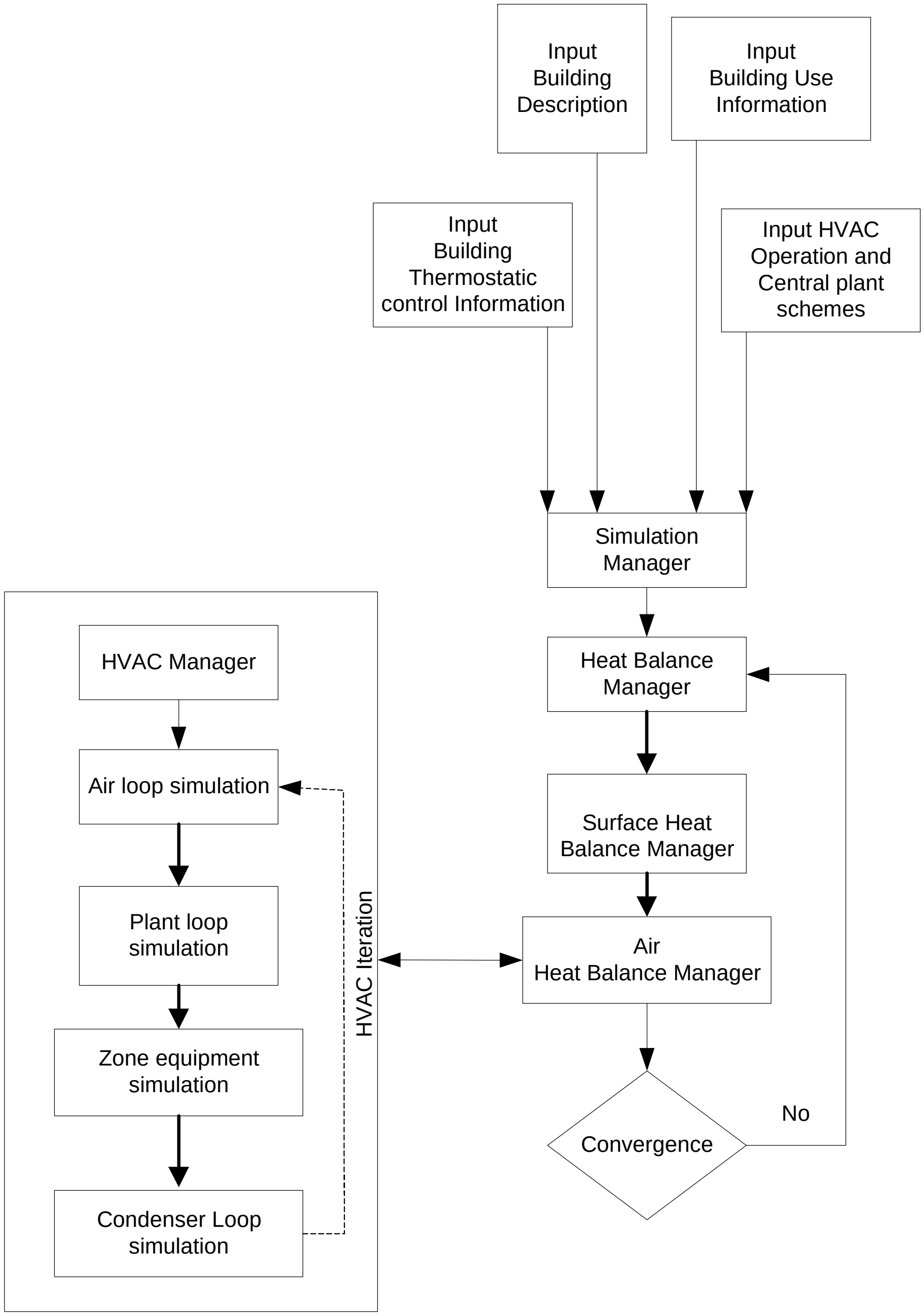

Figure 3.1. *Overall hierarchy of EnergyPlus*

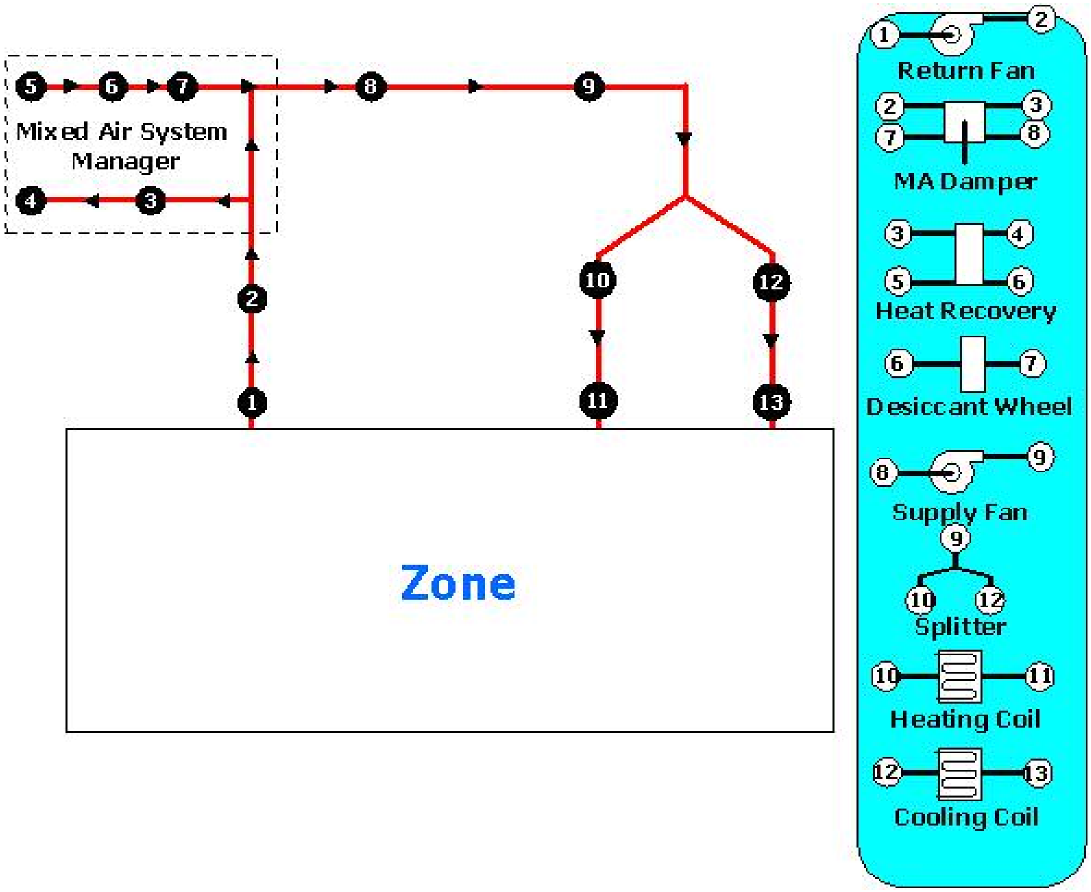

Figure 3.2. *EnergyPlus nodal connections*
(EnergyPlus guide for module developers (EnergyPlus2002)

The HVAC simulation in EnergyPlus is loop based (Fisher, 1999). Equipment such as boilers, chillers, thermal storage tanks and water-to-water heat pumps are simulated on a "Plant loop". Environmental heat exchangers such as cooling towers, ground loop heat exchangers, pavement heat exchangers and ponds are simulated on a "Condenser loop". Heating coils, cooling coils as well as unitary equipment such as water-to-air heat pumps and air-to-air heat pumps are simulated on an "Air loop". The components are connected to the loops by defining nodes at the connections as shown in Fig 3.2 .The nodes are in turn defined in the FORTRAN program as data structures which hold state variable and control information for the node location on the loop.

3.3. Water source heat pumps in EnergyPlus

3.3.1. Water-to-water heat pump simulation

Fig 3.3 shows the HVAC loop connections for a water-to-water heat pump simulation in EnergyPlus. The water-to-water heat pump is coupled both to the plant loop and the condenser loop as shown in the Figure.

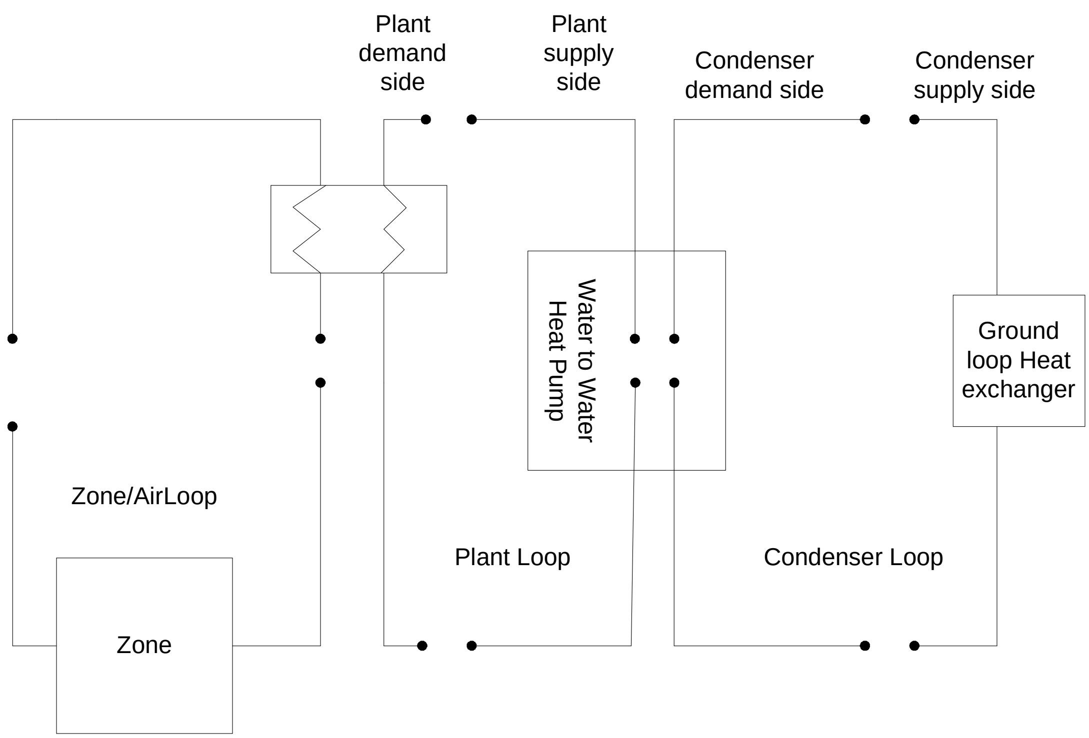

Figure 3.3. *HVAC loop connections for a water-to-water heat pump simulation*

<u>Plant Loop</u>

The plant loop is divided into two sections, so that it can be simulated using a successive substitution solver. The demand side couples coils and other components to the zone/air loop, and the supply side models energy conversion equipment such as boilers, chillers and heat pumps. The plant supply side supplies hot or cold water to meet the demands of the plant demand side and controls the flow rate and the temperature of the loop.

3.3.2. Water-to-air heat pump simulation

Fig 3.4 shows the HVAC loop connections for a water-to-air heat pump simulation in EnergyPlus. The water-to-air heat pump is coupled directly to the condenser loop. The system consists of two loops, the Zone/Air loop and the Condenser loop.

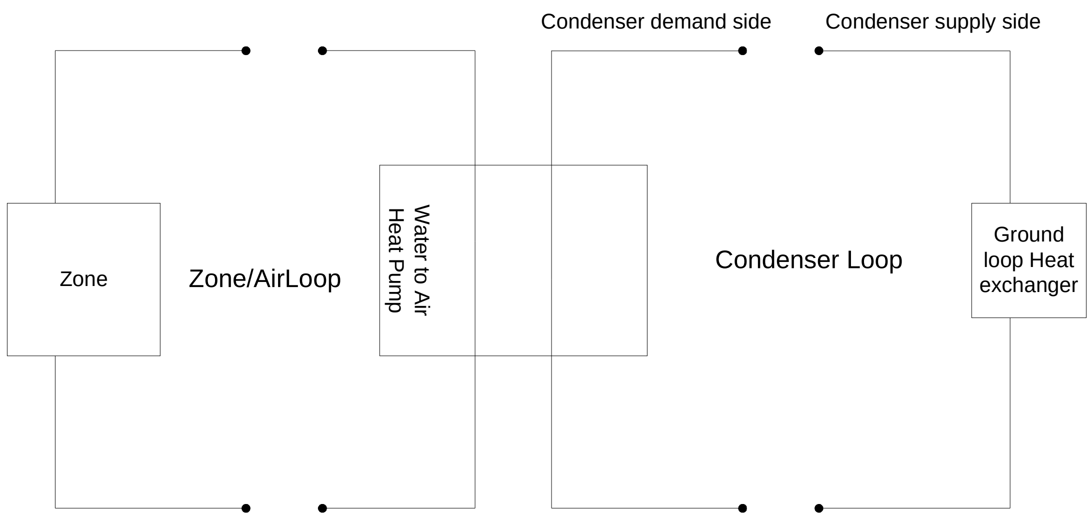

Figure 3.4. *HVAC loop connections for a water-to-air heat pump simulation*

Zone/Air loop

The water-to-air heat pump model is called from the zone air loop manager. The zone thermostat turns the heat pump on or off depending on the sensible demand for that zone. The most recent source side inlet temperature is taken from the condenser loop and the water flow request based on the air loop simulation is passed as a demand to the condenser loop.

Condenser Loop

The condenser loop is divided into two sections, the demand side where energy is transferred to the air stream by various components in the zone/air loop and the supply side where energy is transferred to the environment by various components. The loop attempts to meet the flow request made by the heat pump. The condenser loop temperature is determined by the ground loop heat exchanger simulation. The overall HVAC simulation manager calls the zone/air loop and the condenser loop successively until convergence is achieved.

3.4.Implementing Water-to-air heat pump models in EnergyPlus

In EnergyPlus, the unitary water-to-air heat pump is a virtual component that consists of a single speed fan, a cooling coil, a heating coil and a gas or electric supplementary heating coil as shown in Figure 3.5.

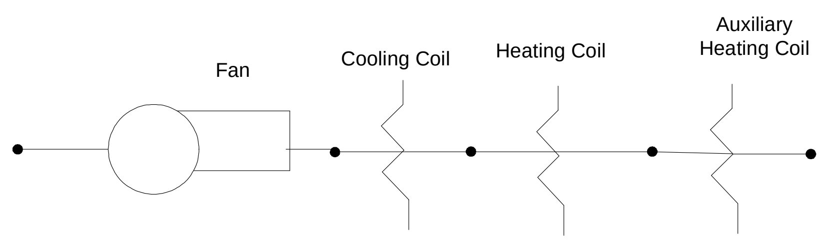

Figure 3.5. *Schematic of Blow thru Water-to-air heat pump virtual component*

The heat pump components are accessed with a single call to the unitary equipment manager from the air loop manager. Air properties are evaluated at nodes between each of the heat pump components. The outlet node of one model forms the inlet node to the next model within the virtual component. The virtual component is connected to the air loop as specified in the user generated input file. This means that the inlet node for the heat pump is also the inlet node for the fan component. In the draw thru configuration, the fan will be located between the coils and the supplemental heating coils. Although it is convenient to cycle the heat pump at the system time step, typically, EnergyPlus time steps will be too long to avoid overheating or overcooling the zone using this control scheme. EnergyPlus, therefore, approximates cycling as a part load by adjusting the system air flow rate to meet the demand of the controlling (thermostat) zone. The fraction of total system volumetric airflow that goes to the controlling zone along with the controlling zone load determines the total load that must be met by the heat pump as:

$$HeatPumpHeatingLoad = \frac{ControlZoneHeatingLoad}{ControlZoneAirFlowFraction} \quad \text{……………………..…(3.1)}$$

$$HeatPumpCoolingLoad = \frac{ControlZoneCoolingLoad}{ControlZoneAirFlowFraction} \quad \text{…………………..…...(3.2)}$$

The “part load” system flow rate is computed on the basis of the heat pump heating or cooling load. The run time fraction of the heat pump in its heating/cooling mode is estimated by the ratio of the sensible heating/cooling demands to the base heating/cooling capacities. The base capacities can be determined from the manufacturers’ catalog data. The equation fit performance model discussed in Chapter 4 is used to determine performance variables such as the heating/cooling capacities, power consumption, energy efficiency ratio and the coefficient of performance. The actual energy extracted or provided is then calculated as the product of the equation fit performance variables for energy extracted or provided and the run time fraction for the respective modes. The power consumption is computed in the same manner. The output data is then moved from the heat pump data structures to the node data structure and are readily available to other computational and reporting modules.

CHAPTER 4

MODEL DEVELOPMENT AND IMPLEMENTATION

4.1. Development of the water-to-air heat pump model

The model implemented in EnergyPlus is based on the water loop heat pump system model developed by Lash (1990). This section describes the development of the model equations and how they were incorporated in the EnergyPlus simulation engine.

4.2. Design basis

Water source heat pumps are required to meet the standards of ARI or ISO 13256-1. The certification program rates water source heat pump performance at specified entering water temperatures. The standard simplifies the use of rating data for heat pump performance modeling in energy analysis calculations and allows for direct rating comparisons across applications. High efficiency water source heat pumps are designed to operate over a range of entering water temperatures in either the heating or the cooling mode. The manufacturer's catalog provides heat pump performance data within a specific range of entering fluid temperatures. The catalog provides sufficient data to develop a correlation for heat pump performance as a function of entering water temperature, entering air temperatures and inlet water mass flow rates. The catalog data were used to fit the coefficients for the water-to-air heat pump described later in this chapter. A typical set of manufacturer's data for the Florida heat pump in the heating and cooling mode is shown in Figure 4.1.

FHP GT030 Data								
All performance at 1000 CFM and 7.5 GPM								
Entering Fluid Temp.	Ent. Air Wet Bulb Temp.	Total Capacity BTUH	Watts Input	Heat Rejection BTUH	Sensible Capacity BTUH Air Dry Bulb Temp.			EER
					75^o	80^o	85^o	
50	61	29514	1209	33639	22460	26946	29514	24.4
50	64	30944	1236	35162	21011	26674	29583	25
50	67	32399	1263	36709	19342	25271	29159	25.7
50	70	33878	1290	38280	16126	22292	29000	26.3
50	73	35381	1318	39877	-	19000	26041	26.9
60	61	28380	1360	33022	21597	25911	28380	20.9
60	64	29756	1391	34501	20204	25650	28447	21.4
60	67	31155	1421	36003	18599	24301	28039	21.9
60	70	32577	1452	37530	15507	21436	27886	22.4
60	73	34022	1482	39080	-	18270	25041	23
70	61	27246	1512	32404	20735	24876	27246	18
70	64	28567	1545	33840	19397	24625	27310	18.5
70	67	29910	1579	35298	17857	23330	26919	18.9
70	70	31276	1613	36779	14887	20580	26772	19.4
70	73	32664	1647	38284	-	17540	24040	19.8
85	61	25546	1739	31478	19441	23324	25546	14.7
85	64	26785	1777	32848	18187	23088	25606	15.1
85	67	28044	1816	34240	16742	21874	25239	15.4
85	70	29324	1855	35654	13958	19295	25101	15.8
85	73	30625	1894	37089	-	16446	22540	16.2
100	61	23846	1966	30552	18147	21771	23846	12.1
100	64	25002	2009	31857	16976	21552	23902	12.4
100	67	26177	2053	33182	15628	20418	23560	12.8
100	70	27372	2097	34528	13029	18011	23431	13.1
100	73	28587	2142	35894	-	15351	21040	13.3

Figure 4.1. *Manufacturer's catalog for Florida Heat Pump GT030*

Using manufacturer's catalog data in the model equations requires conversion since the units vary for the input parameters. For example, in Figure 4.1 British IP units are used for all reported parameters except the power input which is in SI units. The catalog data must be converted to the units required by the model equations—SI units in the case of the EnergyPlus model. The converted catalog data for the Florida heat pump GT030 in the cooling mode is shown in Table 4.1.

Table 4-1.*Manufacturer's catalog for Florida Heat Pump GT030 in the cooling mode*

Entering Fluid	Ent. Air Wet Bulb Temp.	Total Capacity	Power	Heat Rejection	Sensible Capacity (kW) at Ent. Air Dry Bulb Temp			EER
					330.2 K	335.2 K	340.2 K	
(K)	(K)	(kW)	(kW)	(kW)	(kW)	(kW)	(kW)	(-)
283	289.1	8.6	1.2	9.9	6.6	7.9	8.6	24.4
283	290.8	9.1	1.2	10.3	6.2	7.8	8.7	25
283	292.4	9.5	1.3	10.8	5.7	7.4	8.5	25.7
283	294.1	9.9	1.3	11.2	4.7	6.5	8.5	26.3
288.6	289.1	8.3	1.4	9.7	6.3	7.6	8.3	20.9
288.6	290.8	8.7	1.4	10.1	5.9	7.5	8.3	21.4
288.6	292.4	9.1	1.4	10.5	5.4	7.1	8.2	21.9
288.6	294.1	9.5	1.5	11.0	4.5	6.3	8.2	22.4
294.1	289.1	8.0	1.5	9.5	6.1	7.3	8.0	18
294.1	290.8	8.4	1.5	9.9	5.7	7.2	8.0	18.5
294.1	292.4	8.8	1.6	10.3	5.2	6.8	7.9	18.9
294.1	294.1	9.2	1.6	10.8	4.4	6.0	7.8	19.4
302.4	289.1	7.5	1.7	9.2	5.7	6.8	7.5	14.7
302.4	290.8	7.8	1.8	9.6	5.3	6.8	7.5	15.1
302.4	292.4	8.2	1.8	10.0	4.9	6.4	7.4	15.4
302.4	294.1	8.6	1.9	10.4	4.1	5.7	7.4	15.8
310.8	289.1	7.0	2.0	9.0	5.3	6.4	7.0	12.1
310.8	290.8	7.3	2.0	9.3	5.0	6.3	7.0	12.4
310.8	292.4	7.7	2.1	9.7	4.6	6.0	6.9	12.8
310.8	294.1	8.0	2.1	10.1	3.8	5.3	6.9	13.1

Since it is extremely cumbersome to obtain data at every single point within the range specified, manufacturers often assume rated flow rates or temperatures for convenience. Different manufacturers assume different input variables as rated constants. For example, the Florida heat pump catalog assumes rated air flow rates and water flow rates, whereas the ClimateMaster heat pump catalog creates the set of catalog data at constant wet bulb temperatures over a range of water flow rates. The heat pump model was developed to accommodate both data formats. In Table 4.1 All performance data is obtained at a constant air volumetric flow rate of 0.471 m^3/s and a constant water mass flow rate of 0.47 kg/s. A detailed explanation of the methodical approach in using manufacturer's data along with a list of different heat pump manufacturers is provided in Chapter 5.

Following Lash, Heat pump performance based on entering air temperatures, entering water temperatures and inlet water mass flow rate were developed as follows:

4.2.1. Cooling Mode Equations

$$\frac{Q_c}{Q_{base}} = A1 + B1\left[\frac{T_{Win}}{T_{ref}}\right] + C1\left[\frac{T_{ref}}{T_{wb}}\right]\left[\frac{\dot{m}_w}{\dot{m}_{w\text{-}base}}\right] \text{.............................(4-1)}$$

$$\frac{EER}{EER_{base}} = D1 + E1\left[\frac{T_{Win}}{T_{ref}}\right] + F1\left[\frac{T_{ref}}{T_{wb}}\right]\left[\frac{\dot{m}_w}{\dot{m}_{w\text{-}base}}\right] \text{.............................(4-2)}$$

$$\frac{Q_{Sens}}{Q_{Sens\text{-}base}} = G1 + H1\left[\frac{T_{Win}}{T_{ref}}\right] + I1\left[\frac{T_{ref}}{\dot{m}_{w\text{-}base}}\right]\left[\frac{\dot{m}_w}{T_{wb}}\right] + J1\left[\frac{T_{ref}}{T_{db}}\right]\left[\frac{\dot{m}_w}{\dot{m}_{w\text{-}base}}\right] \text{... (4-3)}$$

$$Q_l = Q_c - Q_{Sens} \text{ ....(4-4)}$$

Where:

$A1 - J1$ = Equation fit coefficients for the cooling mode

T_{ref} = 283K or 511 ^oR

T_{Win} = Entering water temperature (K or ^oR)

$\dot{m}_w$ = Mass flow rate of water through the heat pump

$\dot{m}_{w-base}$ = Base mass flow rate of water through the heat pump

T_{db},T_{wb} = Entering Dry bulb and wet bulb air temperatures (K or ^oR)

Q_{base} = Base capacity of the heat pump unit(W)

Qc = Cooling capacity (W)

Ql = Latent cooling capacity (W)

Qsens = Sensible cooling capacity (W)

In cooling mode the heat pump rejects heat to the ground loop. The power input and the heat rejected by the heat pumps are functions of the entering water temperature, the water mass flow rate and the entering air temperature. The power input to the heat pump is computed from the energy efficiency ratio(EER), which is defined as the ratio of net cooling capacity to the total input rate of electric energy, as follows:

$$P = \frac{Q_c}{EER} \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\text{(4-9)}$$

The amount of heat rejected to the loop is given by Equation(4-10)

$$\mathrm{Q_{Reject}} = \mathrm{Q_c} + \frac{\mathrm{Q_c}}{\mathrm{EER}} \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\text{..(4-10)}$$

Equations 4.1 -4.3 take into account critical heat pump performance parameters and are cast in a form that accommodates both variable wet bulb and variable water flow rate data. By defining a reference temperature and a 'base case', the equations are cast in non-dimensional form.

For the FHP cooling mode catalog data shown in Figure 4.1, the sensible capacity is a function of wet bulb temperatures and the dry bulb temperatures while the total capacity is a function of the wet bulb temperature only.

On a psychrometric chart, the lines of constant air enthalpy follow almost exactly the lines of constant wet bulb temperature. Therefore wet bulb temperature variation, which can be easily measured, provides a relatively accurate measurement of the enthalpy difference:

$$\Delta h \propto \Delta WB \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\text{.(4.5)}$$

The air ('load side') coil heat transfer rate is directly proportional to this change in enthalpy. The inlet air wet bulb temperature is therefore an appropriate scale for the air side heat transfer rate in the total capacity equation (4-1).

For a given heat pump (specified coil geometry, compressor and expansion device), the air and water inlet conditions as shown in Fig 4-2, completely determine the total operating capacity of the unit. The inlet water temperature is therefore an appropriate scale for the source side heat transfer rate.

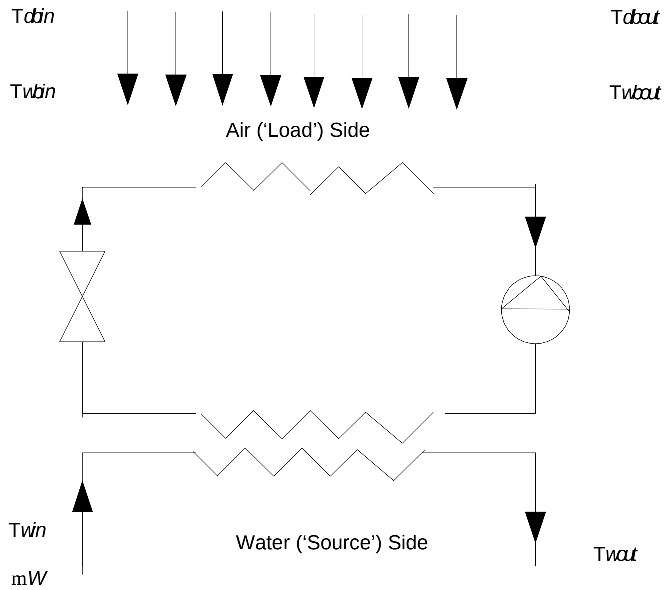

Figure 4.2. *Water to air heat pump cycle (cooling mode)*

When the load increases, the entering wet bulb temperature increases. If the heat pump has to meet this increase in load, with a constant source side water mass flow rate, the water temperature has to be reduced. This establishes an inverse relationship between the entering water temperature and the entering wet bulb temperature as shown in equation 4.6.

$$T_{win} \propto \frac{1}{T_{wb}} \quad \ldots\ldots(4.6)$$

The C1 term in equation (4-1) shows the expected relationship between the water mass flow rate and the entering wet bulb temperature as illustrated in Table 4-2. As the mass flow rate decreases the total cooling capacity and the sensible cooling capacity also decrease. This establishes a direct relationship between the heat pump capacities and the water mass flow rate as shown by equation 4.7.

$$TC \propto \dot{m}_w \quad \ldots\ldots(4.7)$$

Table 4-2. *Manufacturer's catalog data for cooling mode*

		0.9 GPM				1.1 GPM				1.7 GPM			
EWB	EWT	TC	SC	KW	HR	TC	SC	KW	HR	TC	SC	KW	HR
60	60	6.52	5.42	0.46	9	6.69	5.42	0.45	9.1	6.86	5.52	0.43	9.2
65	60	7.33	4.47	0.47	9.1	7.52	4.47	0.46	9.2	7.71	4.56	0.43	9.3
66.2	60	7.49	4.19	0.47	9.2	7.69	4.19	0.46	9.3	7.89	4.27	0.44	9.4
67	60	7.60	4.00	0.48	9.3	7.80	4.00	0.47	9.4	8.00	5.20	0.44	9.5
70	60	7.97	3.21	0.49	9.4	8.18	3.21	0.48	9.5	8.39	3.28	0.45	9.6

Where:

EWB= Entering wet bulb temperature (^{o}F)

EWT= Entering water temperature (^{o}F)

TC = Total cooling capacity (kBtu/hr)

SC = Sensible cooling capacity (kBtu/hr)

KW= Kilowatts input

HR = Heat rejected to water (kBtu/hr)

Table 4-2 also shows that as the mass flow rate decreases the sensible cooling capacity decreases. This establishes a direct relationship between the heat pump capacities and the water mass flow rates as shown by equation 4.8.

$$SC \propto \dot{m}_w \qquad (4.8)$$

Since most data sets show either a range of entering wet bulb temperatures or a range of mass flow rates, but not both, the C1 term may introduce significant error in simulation applications where either the water mass flow rate or the entering wet bulb temperature will vary beyond the range of the single point shown in the catalog data. This concern is discussed in detail in Chapter 5.

4.2.2. Heating Mode Equations

The Heating mode equations are analogous to the cooling mode equations. The dry bulb temperature is used in place of the wet bulb temperature and the COP is used in place of the EER as shown in equation (4-11) through (4-13).

$$\frac{Q_h}{Q_{base}} = A2 + B2\left[\frac{T_{Win}}{T_{ref}}\right] + C2\left[\frac{T_{ref}}{T_{db}}\right]\left[\frac{\dot{m}_w}{\dot{m}_{w\text{-}base}}\right] \text{.............................(4-11)}$$

$$\frac{COP}{COP_{base}} = D2 + E2\left[\frac{T_{Win}}{T_{ref}}\right] + F2\left[\frac{T_{ref}}{T_{db}}\right]\left[\frac{\dot{m}_w}{\dot{m}_{w\text{-}base}}\right] \text{.............................(4-12)}$$

The amount of heat transferred to the ground loop is given by:

$$Q_{absorb} = Q_h - \frac{Q_h}{COP} \text{...(4-13)}$$

The power consumption is given by equation (4-14)

$$P = \frac{Q_h}{COP} \text{...(4-14)}$$

Where:

$A2 - F2$ = Equation fit coefficients for the heating mode

T_{ref} = 283K or 511 ^oR

T_{Win} = Entering water temperature (K or ^oR)

$\dot{m}$ = The mass flow rate of water through the heat pump

T_{db},T_{wb} = The dry bulb and wet bulb air temperatures (K or ^oR)

Q_{base} = The base capacity of the heat pump unit(W)

Qh = Heating capacity (W)

The base capacity is typically selected as the maximum capacity at which the heat pump unit can operate. Hence it can be sorted easily by looking at the peak values of the capacity in the manufacturer's catalog. The base mass flow rate is the water flow rate at the maximum capacity. Similarly, the base EER and COP are the respective values at the maximum capacity.

4.2.3. Accommodating Variable air flow rates

Manufacturers rate the heat pump assuming a "constant volume" fan which means that the fan runs at a constant rpm and delivers a relatively constant volumetric flow rate over a range of air temperatures for a given duct configuration. For rating purposes, the manufacturers assume a zero pressure drop across the fan which is an approximated minimum for an independent water source unit without any ductwork. However, if the heat pump is simulated with rated air flow, predicted performance of the heat pump will be high. With no measured data readily available, air flow correction factors provided by the manufacturer are the only means of estimating the effect of off-design air flow rates. These correction factors may be used to generate extra catalog points as discussed in the next chapter. In order to investigate the sensitivity of the performance variables at variable air flow rates, equations 4.1-4.3 are proposed and implemented in the modified form by including the air mass flow rates as follows:

Cooling Mode:-

$$\frac{Q_c}{Q_{base}} = A1 + B1\left[\frac{T_{Win}}{T_{ref}}\frac{\dot{m}_{a-base}}{\dot{m}_a}\right] + C1\left[\frac{T_{ref}}{T_{wb}}\right]\left[\frac{\dot{m}_w}{\dot{m}_{w\text{-}base}}\right] \text{..................(4-15)}$$

$$\frac{EER}{EER_{base}} = D1 + E1\left[\frac{T_{Win}}{T_{ref}}\frac{\dot{m}_{a-base}}{\dot{m}_a}\right] + F1\left[\frac{T_{ref}}{T_{wb}}\right]\left[\frac{\dot{m}_w}{\dot{m}_{w\text{-}base}}\right] \text{................ .(4-16)}$$

$$\frac{Q_{Sens}}{Q_{Sens\text{-}base}} = G1 + H1\left[\frac{T_{Win}}{T_{ref}}\frac{\dot{m}_{a-base}}{\dot{m}_a}\right] + I1\left[\frac{T_{ref}}{\dot{m}_{w\text{-}base}}\right]\left[\frac{\dot{m}_w}{T_{wb}}\right] + J1\left[\frac{T_{ref}}{T_{db}}\right]\left[\frac{\dot{m}_w}{\dot{m}_{w\text{-}base}}\right] \text{.........(4-17)}$$

Heating Mode:-

$$\frac{Q_h}{Q_{base}} = A1 + B1\left[\frac{T_{Win}}{T_{ref}}\frac{\dot{m}_{a-base}}{\dot{m}_a}\right] + C1\left[\frac{T_{ref}}{T_{db}}\right]\left[\frac{\dot{m}_w}{\dot{m}_{w\text{-}base}}\right] \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\text{(4-18)}$$

$$\frac{COP}{COP_{base}} = D1 + E1\left[\frac{T_{Win}}{T_{ref}}\frac{\dot{m}_{a-base}}{\dot{m}_a}\right] + F1\left[\frac{T_{ref}}{T_{db}}\right]\left[\frac{\dot{m}_w}{\dot{m}_{w\text{-}base}}\right] \ldots\ldots\ldots\ldots\ldots\text{(4-19)}$$

Since only a few data points are available to verify this form of the model, it is proposed for future consideration, but was not implemented in EnergyPlus.

4.2.4. Determining the performance Ccefficients

Performance coefficients in the heating and cooling modes for the equation fit model are determined by implementing a generalized least square equation fitting method. The method uses a minimal sum of the deviations squared from a given set of data. The performance coefficients A1-J1 and A2-F2 in equations (4-1) (4-2) (4-3) (4-11) and (4-12) are generated by this method.

The coefficients are generated by using the available catalog data for the heating mode and the cooling mode. Since water-to-air heat pump systems seldom operate at the catalog specified water temperature, a correlation for the heat pump performance as a function of the entering water temperature, wet bulb and dry bulb air temperature and mass flow rate must be derived from manufacturer’s data as discussed in the previous section. An information flowchart showing all the input parameters is shown in Figure 4.3.

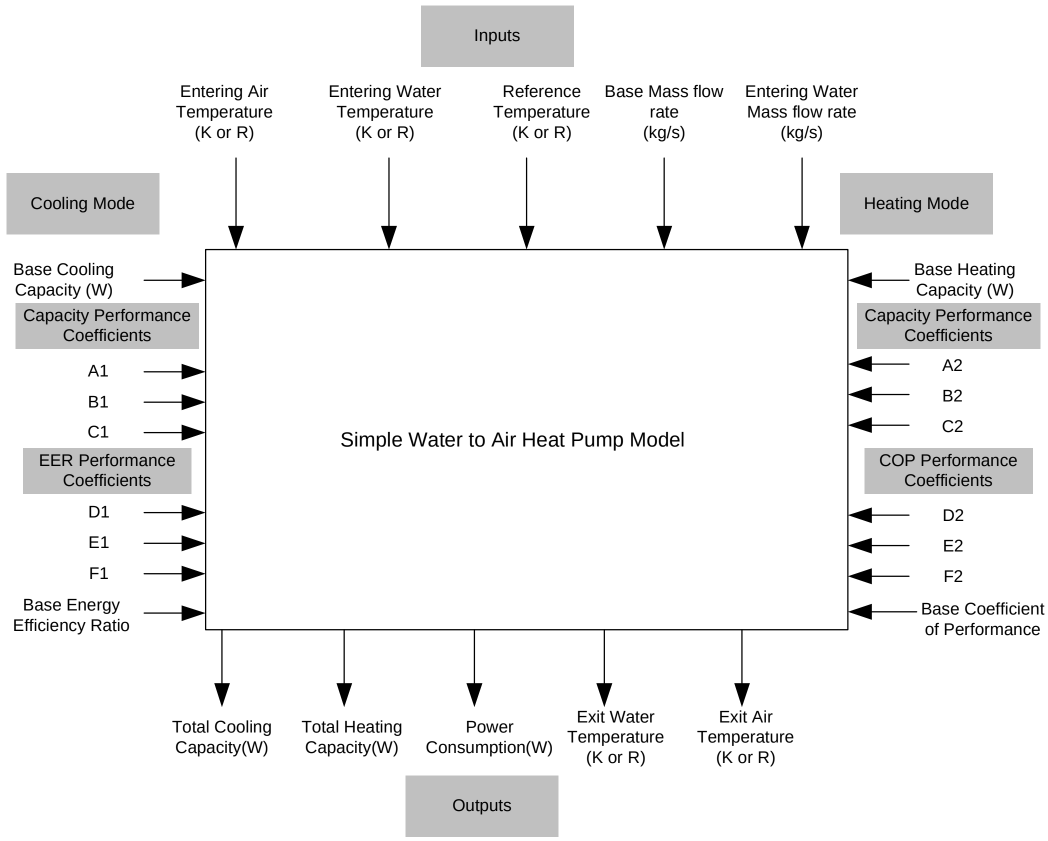

Figure 4.3. *Information flow chart*

4.2.5. Performance coefficients calculation procedure

For each unit at the specified operating conditions, the following catalog and input data is needed:

- Sensible cooling capacity and Latent cooling capacity (Cooling Mode)
- Heating capacity (Heating mode)
- Energy efficiency ratio
- Coefficient of performance
- Nominal capacity of the heat pump unit
- Nominal water mass flow rate
- Inlet water mass flow rate
- Entering air wet bulb and dry bulb temperature

- Entering water temperature
- Nominal coefficient of performance of the heat pump unit
- Nominal energy efficiency ratio of the heat pump unit

Once the data at each operating point is obtained, they are input into a subroutine performing the LU decomposition of the n data points in the matrix. A procedure for decomposing a N x N matrix formed from n data points into a product of lower triangular matrix L and upper triangular matrix U is called the LU decomposition. The upper triangular matrix results from reducing the original matrix by way of elementary row operations excluding row interchange. The lower triangular matrix is created by storing a representative of each elementary row operation .The standard procedure for solving X for the linear system AX=B is shown in Appendix A.

4.3. Model implementation

The model is implemented using the equations (4-1) and (4-4) for the cooling mode and (4-11) and (4-12) for the heating mode. The input/output reference specified in the EnergyPlus guide for module developers (2002) lists guidelines for new module implementation. The water-to-air heat pump model was implemented as a single module with both heating and cooling subroutines. The single module approach with cooling/heating mode switching within the module makes the execution computationally less intensive than the detailed model.

The zone sensible demand tells the model which set of coefficients (heating mode or cooling mode) and which input parameters must be used. The state variable information is read by the model from the loop nodes

Table 4-3 shows the modifications made in accordance with the EnergyPlus standards to implement the water-to-air heat pump simulation. EnergyPlus psychometric property routines were used to calculate the enthalpy, humidity ratios etc.

Table 4-3 *Implementation of the simple water-to-air heat pump model*

Source file/Program	Modifications	Purpose
IDF IDD	**Added** 2 Keywords UNITARYSYSTEM:HEATPUMP:SIMPLE COIL:WATERTOAIRHP:SIMPLE	Defines the inputs for the simple water-to-air heat pump model.
EnergyPlus	**Added** 1 Module WATERTOAIRHPSIMPLE	Encapsulates the data and algorithms required to manage the water-to-air heat pump component
	5 Subroutines	
	SimsimpleWatertoAirHP	Main calling subroutine at the top of the module hierarchy
	GetSimpleWatertoAirHPInput	Obtains input data for the heat pump and stores it in heat pump data structures
	InitSimpleWatertoAirHP	Initializes the water-to-air heat pump components
	CalcSimpleWatertoAirHP	Simulates the cooling and the heating mode of the water-to-air heat pump
	UpdateSimpleWatertoAirHP	Updates the water-to-air heat pump outlet nodes

4.3.1. Input configuration

Energyplus is characterized by two formats of the input specification file, the input data dictionary file (IDD) and the input data file (IDF). The IDD defines a list of all possible EnergyPlus objects (‘keywords’). The IDD has an ‘.idd’ extension. The IDF provides the user description of a specific building and its underlying HVAC system components. The IDF has an ‘.idf’ extension.

4.3.2. Input specification

The keywords may appear in the IDF in any order, but the data under each keyword must follow the format specified in the IDD. Each data value in the IDF must go hand in hand with the keyword fields of the IDD. This means that the type of data (Alpha or Numeric) and the order of the data must match the specifications of the IDD. Figure 4.4 provides a pictorial view of the correspondence required between the IDD and the IDF.

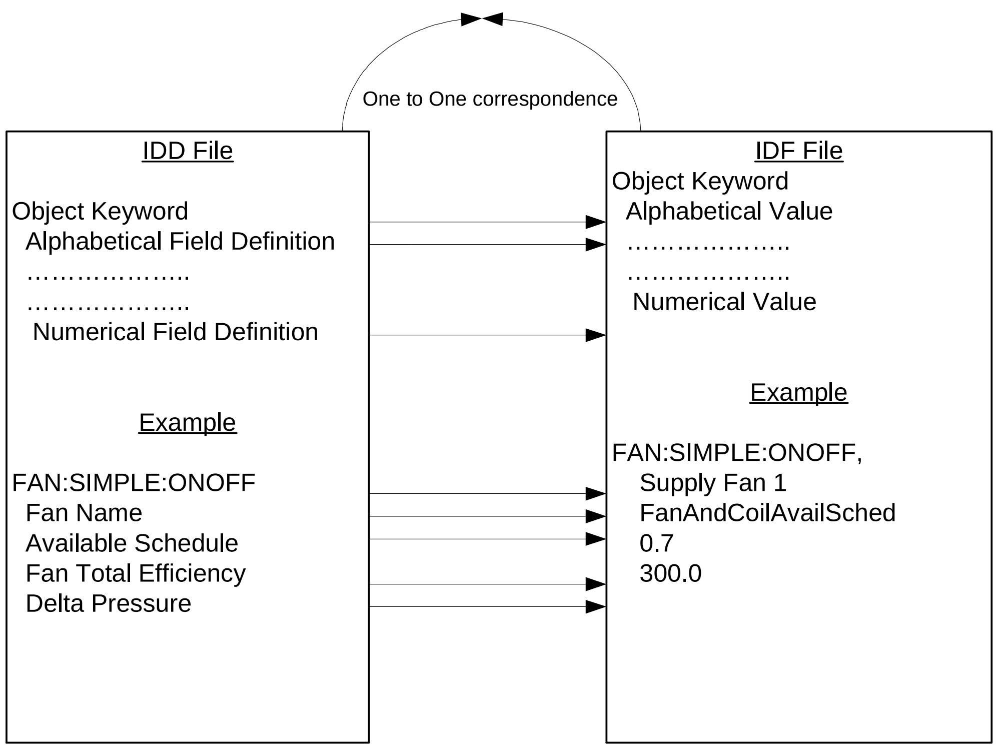

Figure 4.4. *Correspondence required between the IDD and the IDF*

Accordingly, two new objects UNITARYSYSTEM:HEATPUMP:SIMPLE and COIL:WATERTOAIRHP:SIMPLE are created for the model of the water-to-air heat pump as shown in Figure 4.5 and 4.6. The first 15 fields in UNITARYSYSTEM:HEATPUMP:SIMPLE (A1-A15) are alphabetic strings representing the heat pump name, schedule, air nodes, control zone etc. In Figure 4.6

the numeric fields (N1-N18) indicate various numeric inputs that the coil requires to meet the zone demands. N7-N18 are the performance coefficients of the model equations discussed in Section 4.1. The coefficients are generated using the performance coefficient calculator described in detail in chapter 7. Some of the fields carry optional values in them such as the Fan placement which can be a blow-through or a draw-through configuration. The inputs in the Fortran environment are handled by the EnergyPlus Input processor. The Input processor reads the IDD and IDF and supplies the simulation routines with the data contained in the input files. During the course of processing the inputs, the input processor validates keyword formats and node connections.

```
UNITARYSYSTEM:HEATPUMP:SIMPLE,
 A1,  \field Name of Water to air heat pump
 A2,  \field Availability schedule
 A3,  \field Air Side Inlet Node
 A4,  \field Air Side Outlet Node
 A5,  \field Controlling zone or thermostat location
 A6,  \field Supply air fan type
 A7,  \field Supply air fan name
 A8,  \field Fan placement
 A9,  \field Simple Heating or Cooling coil type
 A10, \field Simple Heating or Cooling coil name
 A11, \field Supplemental heating coil type
 A12, \field Supplemental heating coil name
 A13, \field Fan Side Inlet Node
 A14, \field Fan Side Outlet Node
 A15, \field Heat pump operating mode
 N1,  \field Fraction of the total volume flow that goes through controlling zone
 N2,  \field Air Volumetric Flow Rate
 N3,  \field Supplemental heating coil capacity
 N4,  \field Maximum supply air temperature from heat pump supplemental heater
 N5;  \field Maximum outdoor dry-bulb temperature for supplemental heater operation
```

Figure 4.5. *Input object for the simple water-to-air heat pump model defined in the IDD*

```
COIL:WATERTOAIRHP:SIMPLE,
 A1,   \field Name of Coil
 A2,   \field Water Side Inlet Node
 A3,   \field Water Side Outlet Node
 A4,   \field Air Side Inlet Node
 A5,   \field Air Side Outlet Node
 N1,   \field Design Water mass flow rate
 N2,   \field Base water Mass Flow Rate
 N3,   \field Base EER
 N4,   \field Base COP
 N5,   \field Base Capacity for the Heating mode
 N6,   \field Base Capacity for the Cooling mode
 N7,   \field Capacity Coefficient Heating 1
 N8,   \field Capacity Coefficient Heating 2
 N9,   \field Capacity Coefficient Heating 3
 N10,  \field COP Coefficient 1
 N11,  \field COP Coefficient 2
 N12,  \field COP Coefficient 3
 N13,  \field Capacity Coefficient cooling 1
 N14,  \field Capacity Coefficient cooling 2
 N15,  \field Capacity Coefficient cooling 3
 N16,  \field EER Coefficient 1
 N17,  \field EER Coefficient 2
 N18;  \field EER Coefficient 3
```

Figure 4.6. *Input object for the heating and cooling coils as defined in the IDD*

4.3.3. Flow of control / implementation algorithm

The water-to-air heat pump simulation is located in the module SIMPLEWATERTOAIRHP.. Once the heating/cooling latent and sensible demands are determined, the water-to-air heat pump simulation routine is called. SIMSIMPLEWATERTOAIRHP is the main driver routine for the water-to-air heat pump. The sensible and latent coil demands are passed as arguments to the driver routine. The driver routine, as shown in Figure 4.7, calls specific service subroutines to execute various tasks discussed below.

- GetSimpleWatertoAirHPInput

This subroutine obtains input data for the water-to-air heat pump and stores it in the data structures. The routine also reads the data, checks whether they conform to IDD and IDF specifications, allocates arrays, initializes data structures and sets up report variables for the heat pump.

- InitSimpleWatertoAirHP

This subroutine initializes the water-to-air heat pump components at the beginning of each environment, day, hour or time step. It uses status flags to trigger appropriate initializations. It also updates local simulation variables with the latest node data.

- CalcSimpleWatertoAirHP

This subroutine simulates the cooling and the heating mode of the water-to-air heat pump. It simulates the model for the inlet conditions using the calculated performance coefficients and predicts the outlet conditions. The main subroutine employs 3 routines called HEAT, COOL and OFF which simulate the three operating modes of the heat pump. The check for the type of mode depends on the sign associated with the coil

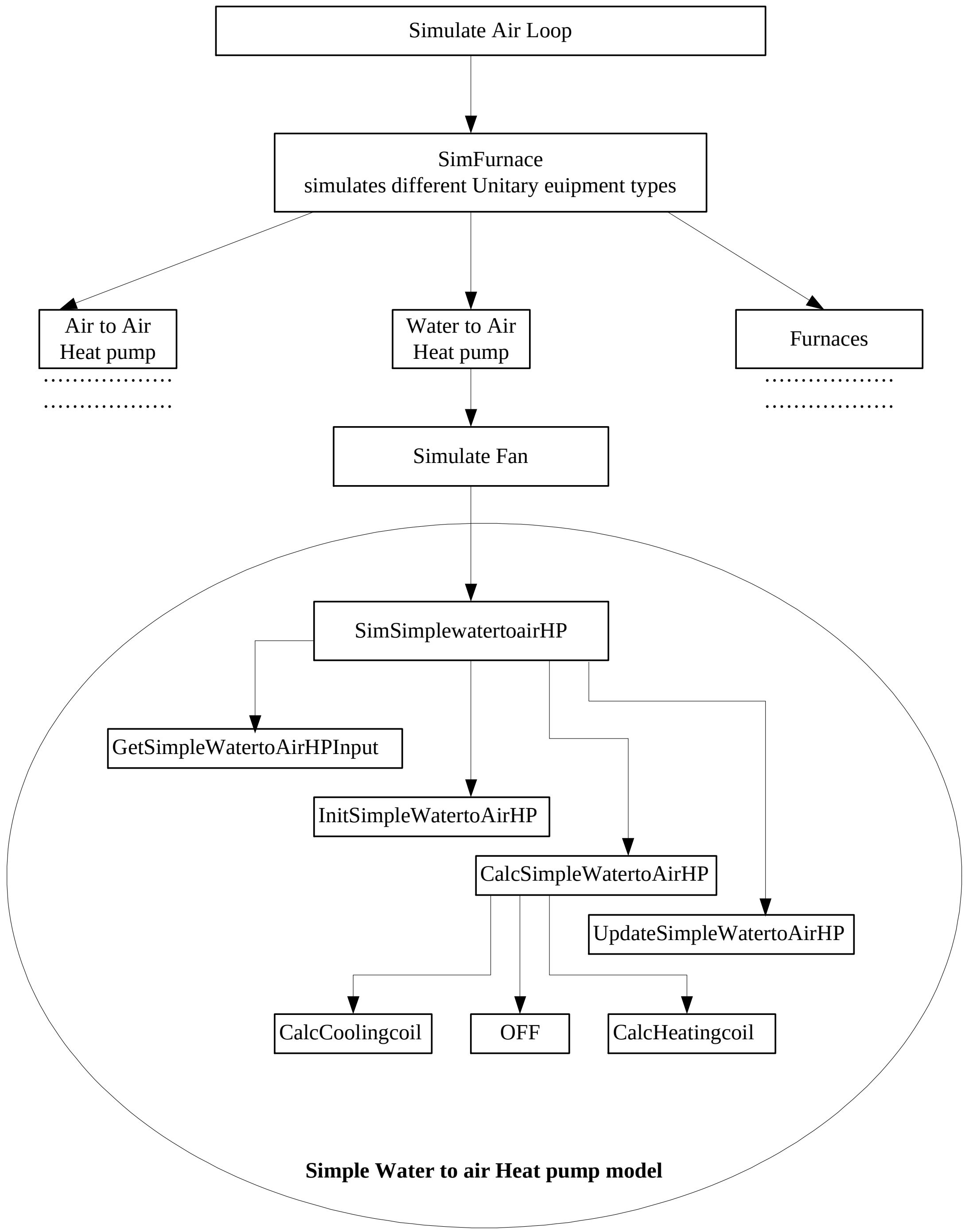

Figure 4.7. *Hierarchical flow of simulation*

demand, which has been discussed earlier as the thermostatic signal. The duty factor is calculated as the ratio of the zone load to the nominal capacity of the heat pump. Once all the outlet variables are computed, the control is transferred to the update routines.

- UpdateSimpleWatertoAirHP

This subroutine updates the water-to-air heat pump outlet nodes. Data is moved from the heat pump data structure to the heat pump outlet nodes. Output variables like the heating and cooling capacities, power consumption and the load side and the source side temperatures are computed.

A detailed flowchart of the computational algorithm is shown in Figure 4.8.The algorithm uses the building hourly loads and the base heat pump capacity to compute the runtime fraction. The building hourly loads are equated to the heat pump hourly loads. Depending on the hourly load (Cooling/Heating), the algorithm switches between the operating modes. Then the model equations and the generated performance coefficients are used to calculate the performance variables such as capacities, COP and EER. The actual power, is the product of the calculated capacity and the runtime fraction. If the heat pump capacity exceeds the demand, the demand is met and the heat pump is switched off. However, if the demand exceeds the capacity, the duty factor of the heat pump equals 1, and and the heat pump runs at its maximum capacity.

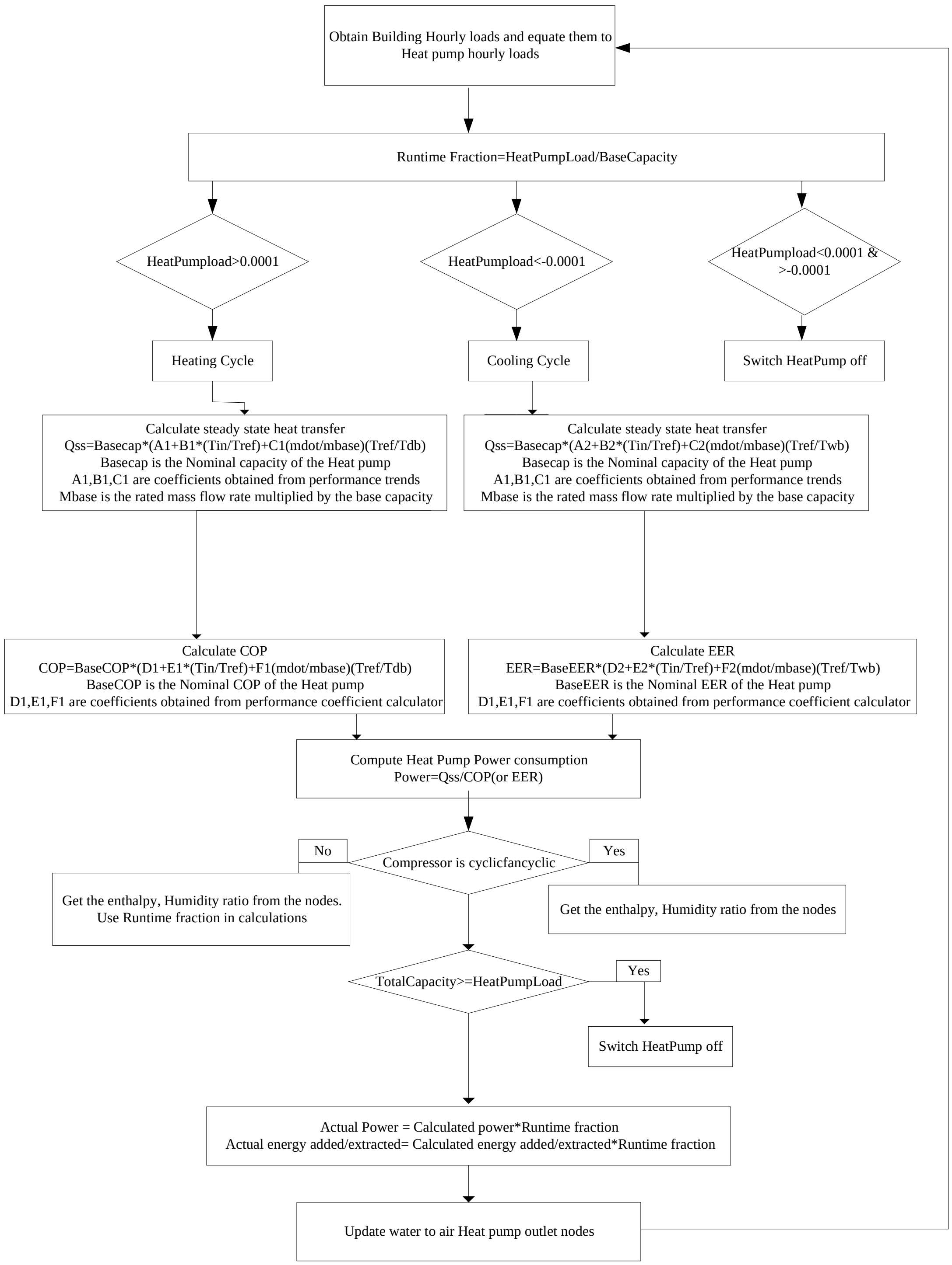

Figure 4.8. *Flow of code for the simple water-to-air heat pump model*

4.3.4. Output arrangement

EnergyPlus has an efficient way of handling outputs. The output formats of the files are simple and text based. The output variables of the water-to-air heat pump can be reported at every HVAC time step. EnergyPlus automatically saves all node data for each timestep. This information is available upon user request in the IDF. In addition, any simulation variable can be specified as a report variable in the computational module and included in the output reports. Output reporting is enhanced by the READVARS.EXE application which converts the text format of the output file to a readable excel table format.

CHAPTER 5

MODEL VERIFICATION

5.1. Introduction

The simple water-to-air heat pump is verified by using catalog data from two different heat pump manufacturers. The model has been verified for both heating and cooling modes. The verification exercise is intended to check both the form of the model and the generation of the performance coefficients. The performance coefficients for the model have been generated using the performance coefficient calculator, a graphical user interface written in visual basic and described in detail in Chapter 7.

5.2. Cooling mode verification

Table 5.1 summarizes the results of the cooling mode verification exercise. The RMS error was less than 1.3% for the cooling capacity. The RMS error for the model prediction of power consumption in comparison with the catalog data ranged from 0.2% to 1.2%. The comparison shows good agreement between the data predicted by the model and the catalog data. The model also predicts sensible and latent capacity splits with good accuracy. The RMS error is higher when the nominal capacity of the heat pump increases drastically. This may be due to small inconsistencies in the catalog data.

Table 5.1. *List of heat pumps in cooling mode for model verification*

No	Manufacturer	Nominal Cooling Capacity		RMS Error for Capacity	RMS Error for Power	RMS Error for Energy Added to ground
		W	Btu/h			
1	FHP024	7000	24000	0.42%	0.21%	0.33%
2	ClimateMasterGSH024	7000	24000	1.28%	1.15%	1.23%

Also, in the case of heat pump#1(Florida Heat Pump), with all data reported for a single water flow rate, all the calculated points lie on the diagonal to match the catalog data points as shown in Figures 5.1-5.4. However, in the case of heat pump#2 (ClimateMaster) as shown by Figures 5.5-5.8, some of the points group and cluster away from the diagonal. This error is small and may be a result of either the computer model used to generate the tables from a few measured points or of the testing and rating methods used by the manufacturer. Water source heat pumps are subjected to the rating requirements of ARI standard 320 which allows a tolerance of ±5% from the catalog data. Also, the ARI rated condition forms one data point among the various data points in the catalog. As previously noted, many of the catalog points are generated by computer models. This creates a built in uncertainty in the catalog data.

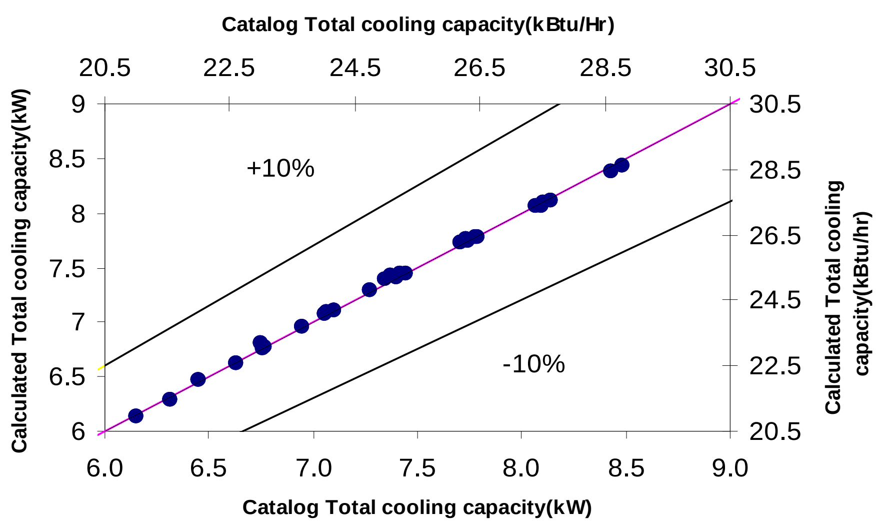

Figure 5.1. *Catalog cooling capacity v/s calculated cooling capacity (heat pump #1)*

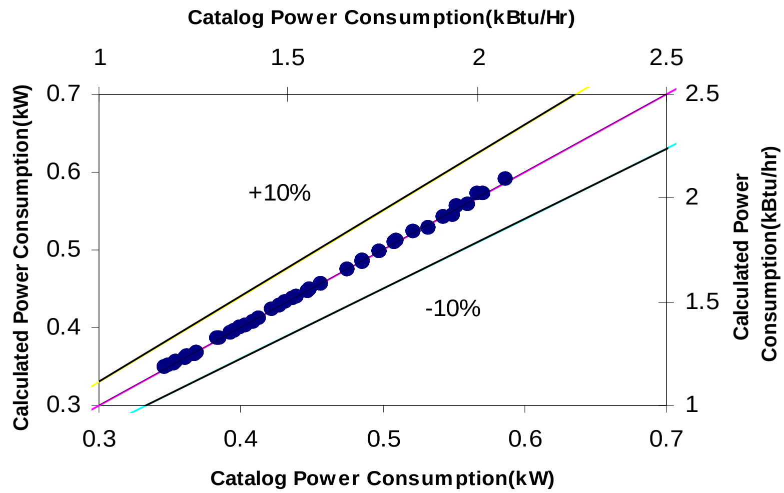

Figure 5.2. *Catalog power consumption v/s calculated power consumption (heat pump #1)*

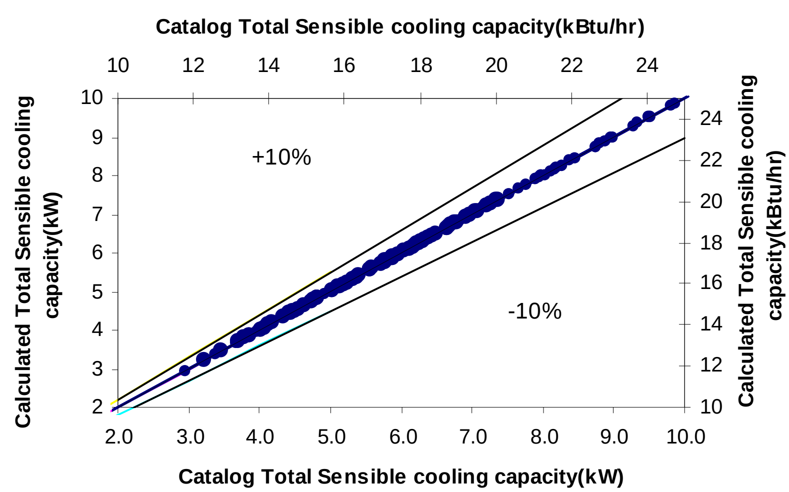

Figure 5.3. Catalog *sensible capacity v/s calculated sensible capacity (heat pump #1)*

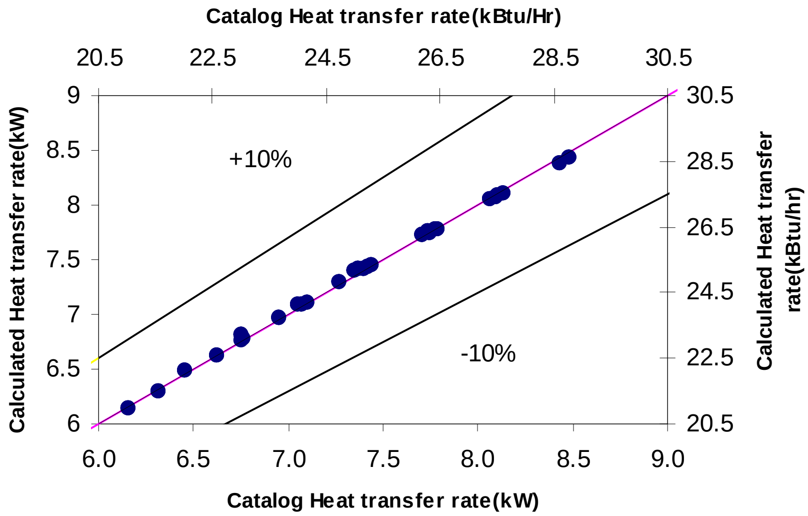

Figure 5.4. *Catalog v/s calculated heat transfer rate (heat pump #1)*

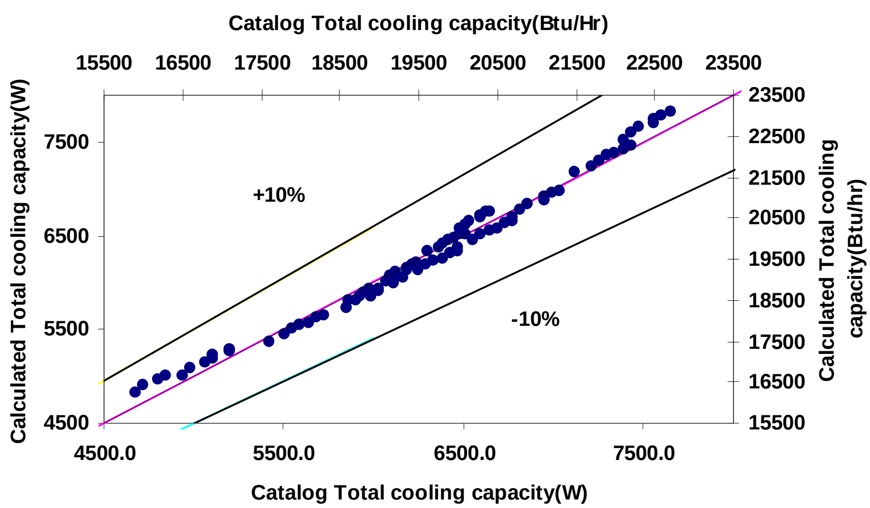

Figure 5.5. *Catalog cooling capacity v/s calculated cooling capacity (heat pump #2)*

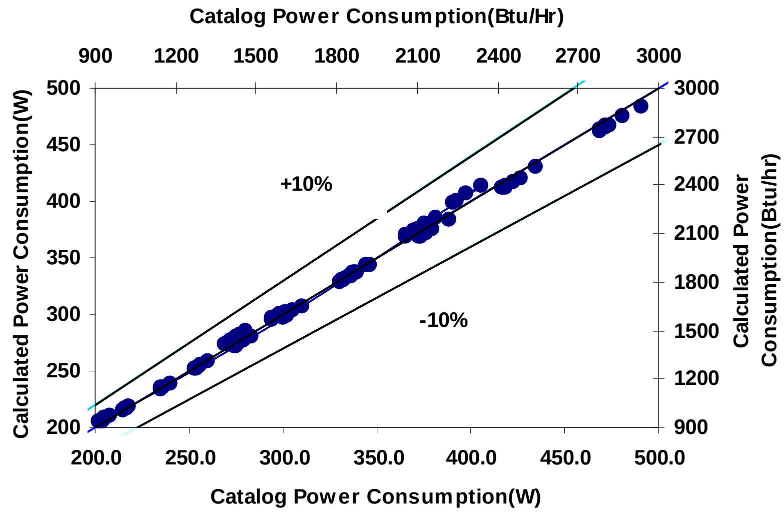

Figure 5.6. *Catalog power consumption v/s calculated power consumption (heat pump #2)*

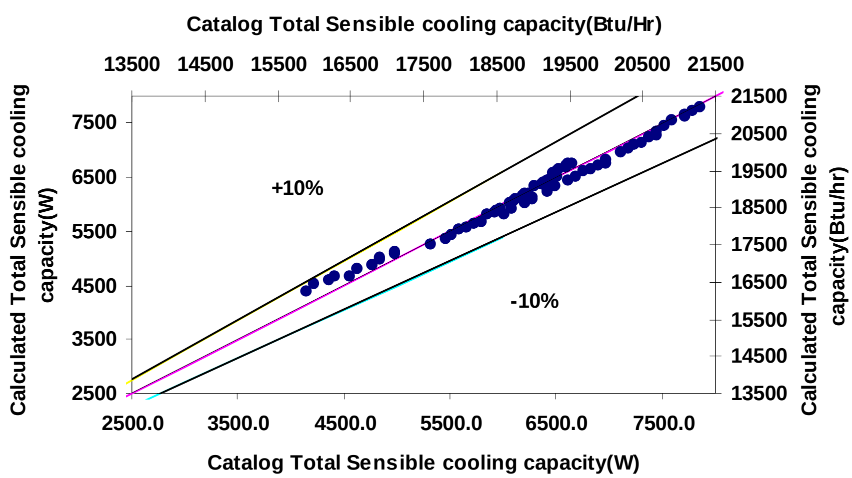

Figure 5.7. *Catalog sensible capacity v/s calculated sensible capacity (heat pump #2)*

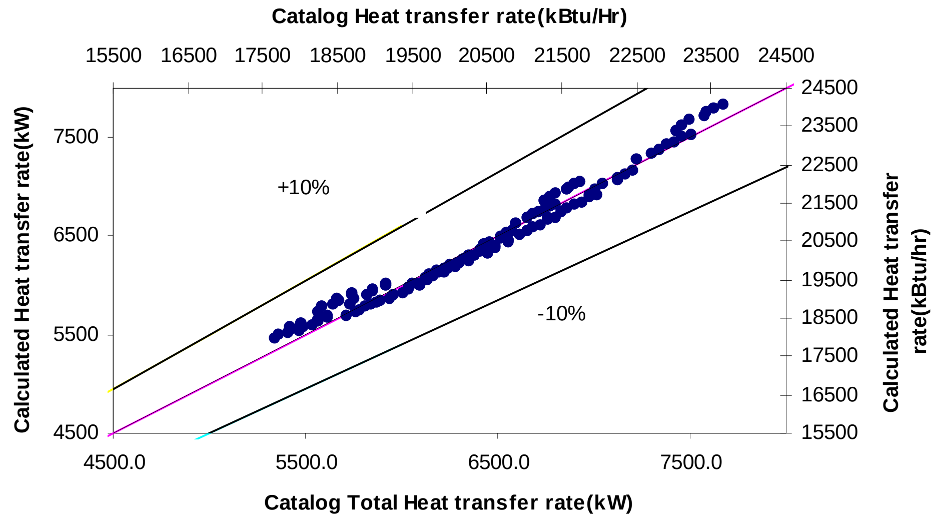

Figure 5.8. *Catalog v/s calculated heat transfer rate (heat pump #2)*

5.3. Heating mode verification

Table 5.2 summarizes the results of the heating mode verification exercise. There is a good agreement between the model predicted data and the catalog data for the heating mode of the simple model. The RMS error was less than 0.8% for the heating capacity. The RMS error for the model prediction of power consumption in comparison with the catalog data ranged from 0.2% to 0.65%. The comparison shows good agreement between the model predicted data and the catalog data. The heating model as shown by the Figures 5.9-5.14 performs better than the cooling model with most of the points lying on the diagonal. The error is small and probably occurs due to the uncertainty in the catalog data as discussed in the previous section.

Table 5-2. *List of heat pumps in heating mode for model verification*

No	Manufacturer	Nominal Heating Capacity		RMS Error for Capacity	RMS Error for Power	RMS Error for Energy extracted from ground
		W	Btu/h			
1	FHP024	7000	24000	0.41%	0.26%	0.31%
2	ClimateMasterGSH024	7000	24000	0.73%	0.62%	0.71%

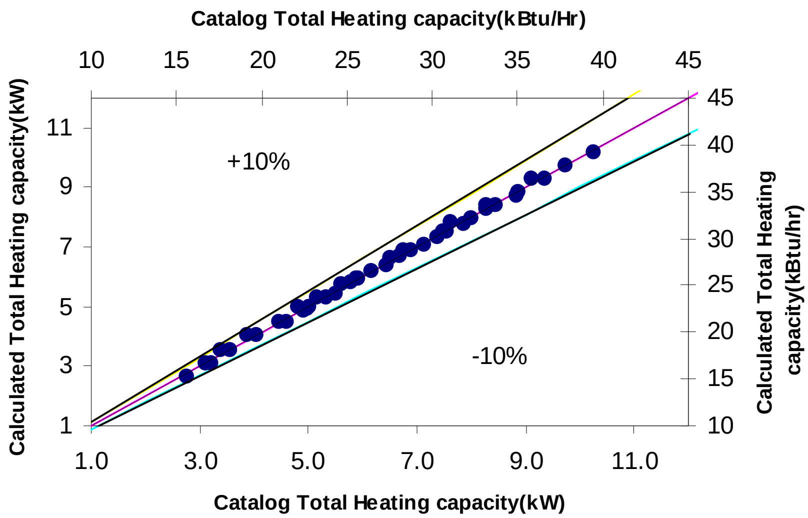

Figure 5.9. *Catalog heating capacity v/s calculated heating capacity (heat pump #1)*

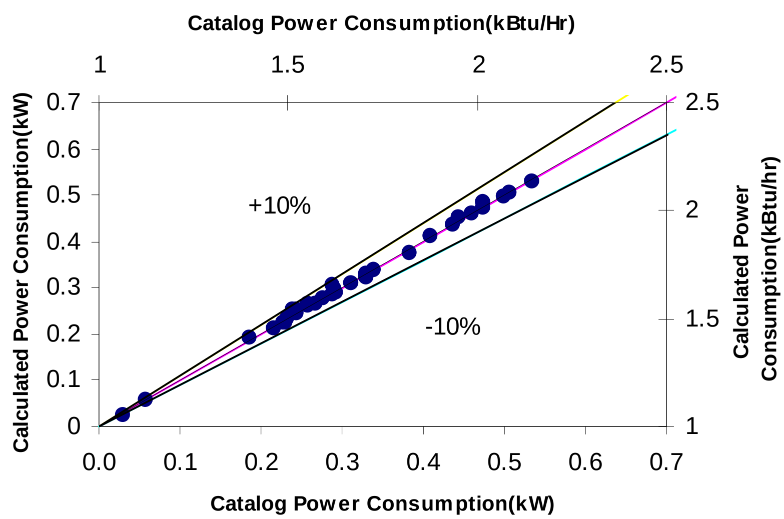

Figure 5.10. *Catalog power consumption v/s calculated power consumption (heat pump #1)*

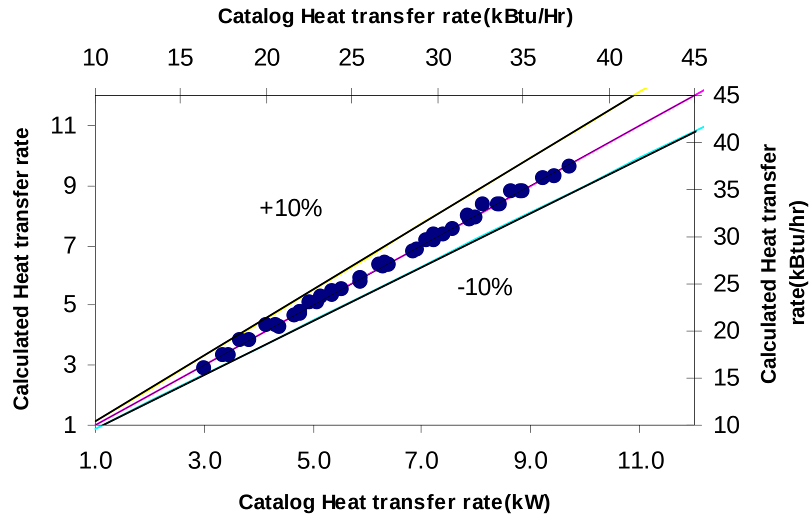

Figure 5.11.*Catalog v/s calculated heat transfer rate (heat pump #1)*

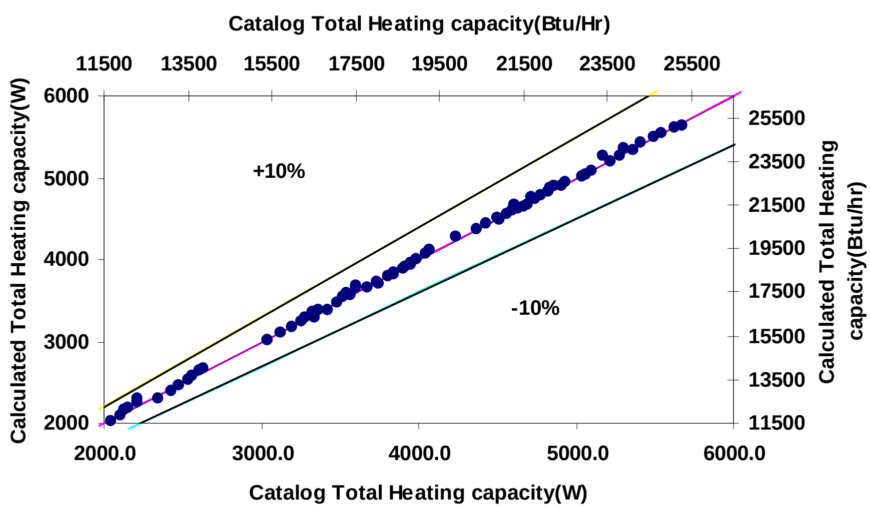

Figure 5.12. *Catalog heating capacity v/s calculated heating capacity (heat pump #2)*

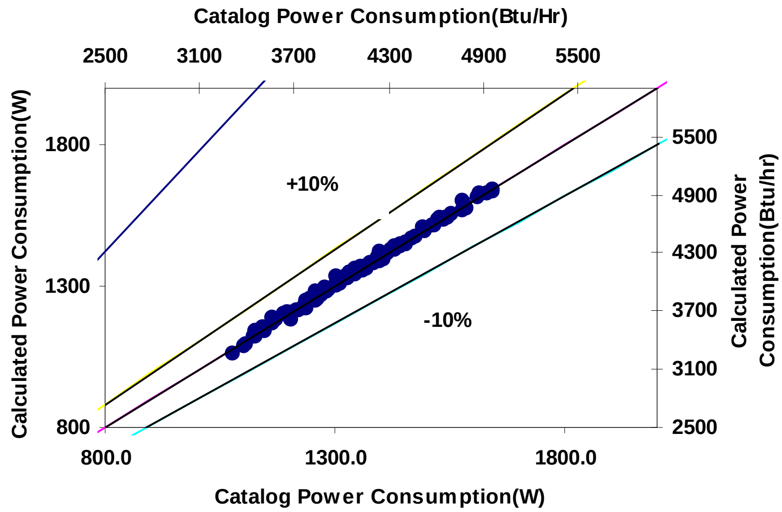

Figure 5.13. *Catalog power consumption v/s calculated power consumption (heat pump #2)*

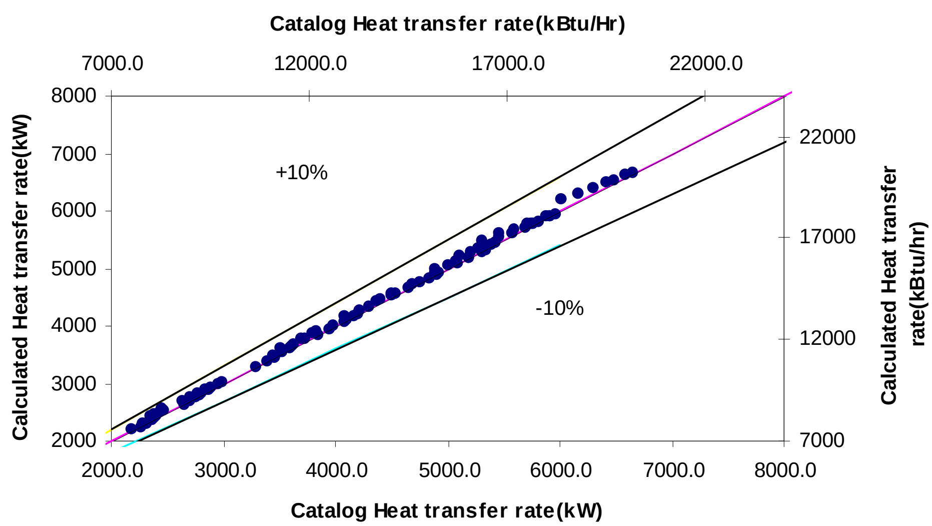

Figure 5.14.*Catalog v/s calculated heat transfer rate (heat pump #2)*

5.4. Verification using Correction Factors

5.4.1. Correction Factors

As noted in Section 4.3, some manufacturers, such as ClimateMaster, provide correction factors to account for the effect of parameters that are held constant in their data sets. These parameters include the air flow rates and the wet bulb temperatures. If the units are operating at conditions different than the rated conditions, the heat pump capacities can be approximated using these correction factors. Table 5.3 and 5.4 show air flow rate(CFM) and wet bulb temperature correction factors for total capacity (TC), sensible capacity (SC), power and heat rejection rate (HR) to the ground for cooling operation of the ClimateMaster GS series units.

Table 5.3. *Correction factors for the air flow rate*

CFM	TC	SC	Power	HR
350	0.957	0.946	0.994	0.964
375	0.979	0.969	0.997	0.982
400	1	1	1	1
425	1.021	1.029	1.003	1.018
450	1.043	1.058	1.006	1.036

Table 5.4. *Correction factors for the wet bulb temperatures*

WB	TC	SC (80DB)	Power	HR
60	0.899	1.192	0.984	0.899
65	0.94	1.106	0.991	0.949
66.2	0.976	1.043	0.997	0.98
67	1	1	1	1
70	1.012	0.933	1.002	1.01
75	1.024	0.8	1.005	1.019

The influence of the correction factors on the performance variables is observed by plotting the correction factors against the air flow rates and the wet bulb temperatures as shown by Figure 5.15 and 5.16.

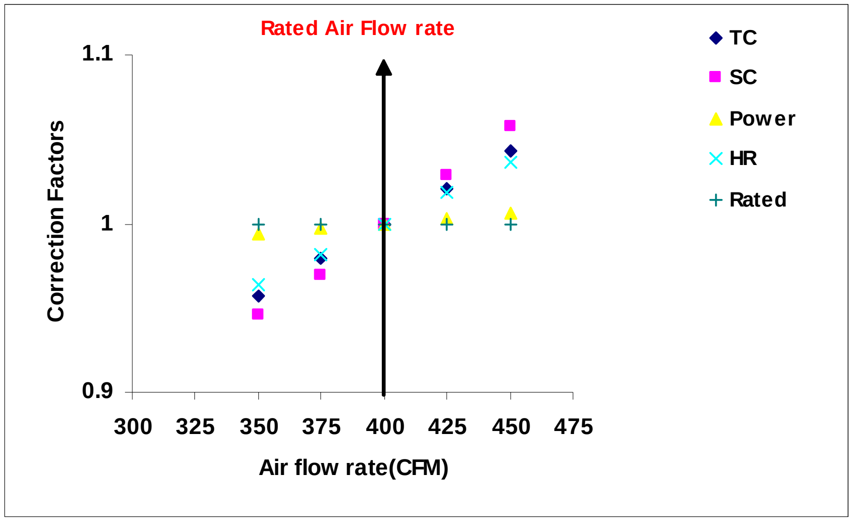

Figure 5.15. *Influence of air flow correction factors on performance*

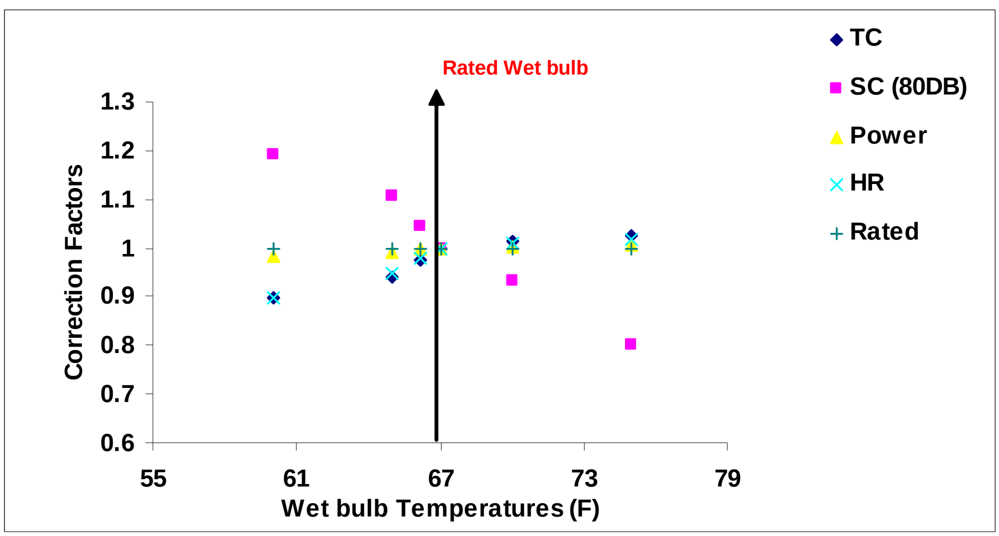

Figure 5.16.*Influence of wet bulb correction factors on performance*

Figure 5.15 shows that for a variation of 50 CFM (12.5%) in the air flow rate, the performance of the GS series heat pump changes by less than 10%. For the same change in air flow rate, the change in power is negligible. Since the sensible cooling capacity is highly dependent on the wet bulb temperature, it shows a significant variation (20%) over a wet bulb range of 15^{o} F (60-75^{o}F) as shown in Figure 5.16. For low wet bulb depression (high relative humidity) a change in the wet bulb affects the latent/sensible split, but not the total capacity. At low relative humidity, Fig. 5-16 shows that a change in wet bulb temperature affects both the total capacity and the latent/sensible split.

The plots suggest that the standard manufacturers' data sets should be extended to include a range of air flow rates and inlet wet bulb temperatures (in addition to water mass flow rates, water inlet temperatures and inlet dry bulb temperatures). Since in the simulation environment, neither the wet bulb temperature nor the air flow rate are expected to always be at the rated condition, using the correction factors to extend the data set is necessary to ensure that the model is not applied outside the range of the catalog data.

5.4.2. Comparison of simulated and measured data using correction factors

In this section, the effect of applying the model outside the range of the manufacturer's standard data set is investigated. Since experimental data is not available, data points generated using correction factors are used.. The procedure is as follows:

- Generate model coefficients using the standard data set (no correction factor data)
- Compare predicted outputs with calculated outputs at non-rated (correction factor) conditions.
- Generate model coefficients using extended data set (standard + correction factor data).
- Compare predicted outputs with calculated outputs at non-rated (correction factor) conditions.

5.4.2.1. Comparison Procedure

ClimateMaster GC series performance and correction data was used as shown in Fig 5.17. Nine data points were generated using the correction factors for the total cooling capacity at different air flow rates. Similarly, thirty one data points were generated using the correction factors for the sensible cooling capacity over the range of wet bulb temperatures. The data points were merged with the catalog data set characterized by constant air flow rate and wet bulb. Then the entire data set was used to generate a new set of coefficients for equations 4-1 through 4-3 and equations 4-15 through 4-17 by applying the least square technique. A set of coefficients based on the regular catalog data set (without the correction factors) and a set of coefficients (with the correction factors) were then used in the model equations to calculate heat pump capacities and power use at variable air mass flow rates and wet bulb temperatures.

PERFORMANCE DATA
CORRECTION TABLES

AIR FLOW CORRECTION TABLE

Airflow	Heating			Cooling			
% of Nominal	Htg Cap	Power	Heat of Ext	Total Cap	Sens Cap	Power	Heat of Rej
75%	0.966	1.051	0.939	0.970	0.899	0.953	0.967
81%	0.976	1.037	0.956	0.979	0.924	0.966	0.976
88%	0.985	1.023	0.973	0.987	0.949	0.979	0.985
94%	0.993	1.012	0.987	0.994	0.975	0.990	0.993
100%	**1.000**	**1.000**	**1.000**	**1.000**	**1.000**	**1.000**	**1.000**
106%	1.006	0.991	1.010	1.005	1.026	1.008	1.005
113%	1.011	0.982	1.020	1.009	1.051	1.016	1.010
119%	1.014	0.975	1.027	1.011	1.077	1.022	1.013
125%	1.017	0.968	1.033	1.013	1.102	1.027	1.016

Rev: 06/12/01 B

ENTERING AIR CORRECTION TABLE

Heating Corrections			
Ent Air DB °F	Htg Cap	Power	Heat of Ext
45	1.044	0.803	1.123
50	1.042	0.847	1.107
55	1.037	0.888	1.086
60	1.028	0.927	1.062
65	1.016	0.965	1.033
68	1.007	0.986	1.014
70	**1.000**	**1.000**	**1.000**
75	0.980	1.033	0.963
80	0.957	1.065	0.921

Cooling Corrections										
		Sens Clg Cap Multiplier - Entering DB °F								
Ent Air WB °F	Total Clg Cap	70	75	80	80.6	85	90	95	Power	Heat of Rej
60	0.858	0.812	1.062	1.217	1.229	*	*	*	0.982	0.886
65	0.964	0.622	0.876	1.076	1.098	1.240	*	*	0.996	0.971
66.2	0.986	0.577	0.822	1.032	1.055	1.214	*	*	0.999	0.989
67	**1.000**	0.547	0.785	**1.000**	1.024	1.192	1.362	1.508	**1.000**	**1.000**
70	1.049		0.630	0.864	0.891	1.086	1.236	1.399	1.004	1.039
75	1.113			0.580	0.609	0.814	1.027	1.218	1.007	1.089

Rev: 06/14/04D

* Sensible capacity equals total capacity

ARI/ISO/ASHRAE 13256-1 uses entering air conditions of Clg- 80.6°F DB/66.2°F WB and Htg- 68°F DB/59°F WB

Bold Print indicates base performance as shown in submittal data tables.

Figure 5.17. *Correction factors for GC ClimateMaster series (used with permission)*

The ClimateMaster GC018 data set was used in the verification process. The total and sensible capacities were calculated over a range of wet bulb temperatures and air mass flow rates as shown in Table 5.5. The 3rd data point was introduced by the manufacturers to meet the European standards of 19^{o}C(66.2^{o}F) wet bulb and 27^{o}C(80.6^{o}F) dry bulb.

Table 5.5. *Range of wet bulb temperatures and air flow rates for GC series*

WB(F)	Air flow rate(kg/s)	Air flow rate (% of Nominal)
60	0.25	75
65	0.28	81
66.2	0.30	88
67	0.32	94
70	0.34	100
75	0.36	106
-	0.38	113
-	0.40	119
-	0.42	125

5.4.2.2. Results

Figure 5.18-5.23. summarizes the complete set of coefficients generated for the study. Two data sets were used to generate coefficients for six model equations: total capacity, sensible capacity and power for both forms of the model presented in section 4.3..By merging the extra data points generated using correction factors with the existing data set, coefficients for the total cooling capacity, sensible capacity and the power consumption equations are calculated. The coefficients generated using constant wet bulb temperature of 67 ^{o}F,(rated condition) diverge from the catalog data as shown in Figure 5.18-5.20. The one point that lies on the diagonal is the rated condition. With the inclusion of the corrected wet bulb temperature points, the total cooling capacity, sensible capacity and the power consumption were in good agreement with the catalog data. As expected for all three figures, the two models (equations 4-1 – 4-3 and equations 4-15-4-17) show identical results for coefficients generated with and without correction factor data.

As shown in figure 5-18, the total capacity is accurately predicted within 10% by the water mass flow rate alone. That is, generating model coefficients at the rated wet bulb temperature then applying the model beyond the data set (over a 15 ^{o}F range of wet bulb temperatures) results in less than 1% error in the predicted total capacity.

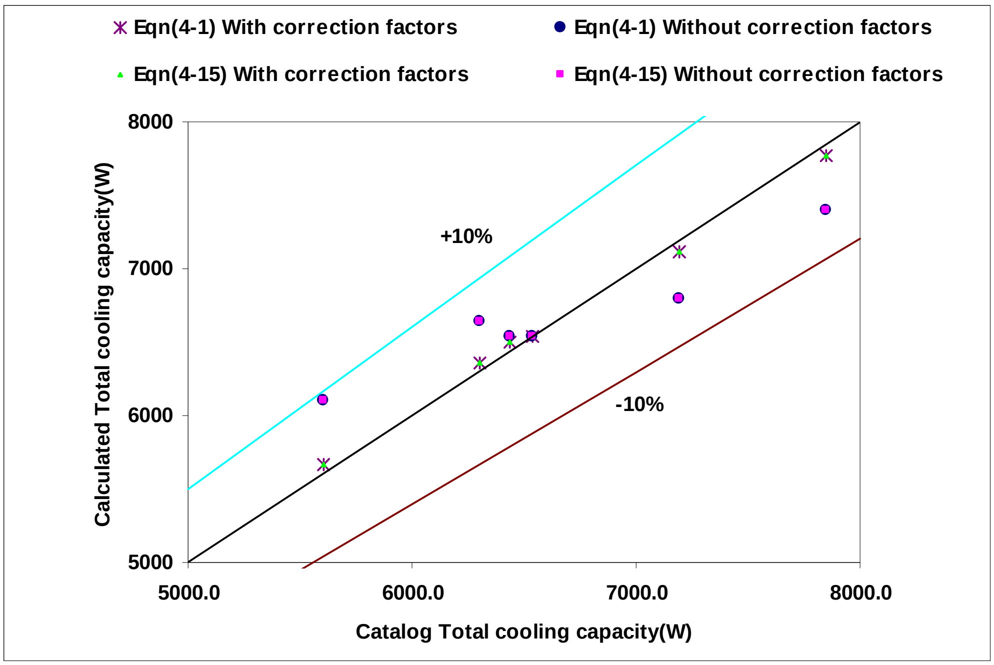

Figure 5.18. *Comparison of catalog v/s calculated cooling capacity with and without the correction factors for wet bulb temperatures*

Figure 5-19 shows that the sensible/latent split cannot be accurately predicted without including a range of wet bulb temperatures in the data set used to generate the coefficients. This is the expected result based on Figure 5.16 and 5.17. As shown in Figure 5.19 the error in the predicted sensible heat transfer rate is in the range of 30% to 40%. The error in the predicted power on the other hand is small (less than 2%) as shown in Figure 5.20. This result is also expected from Figure 5.16 and 5.17.

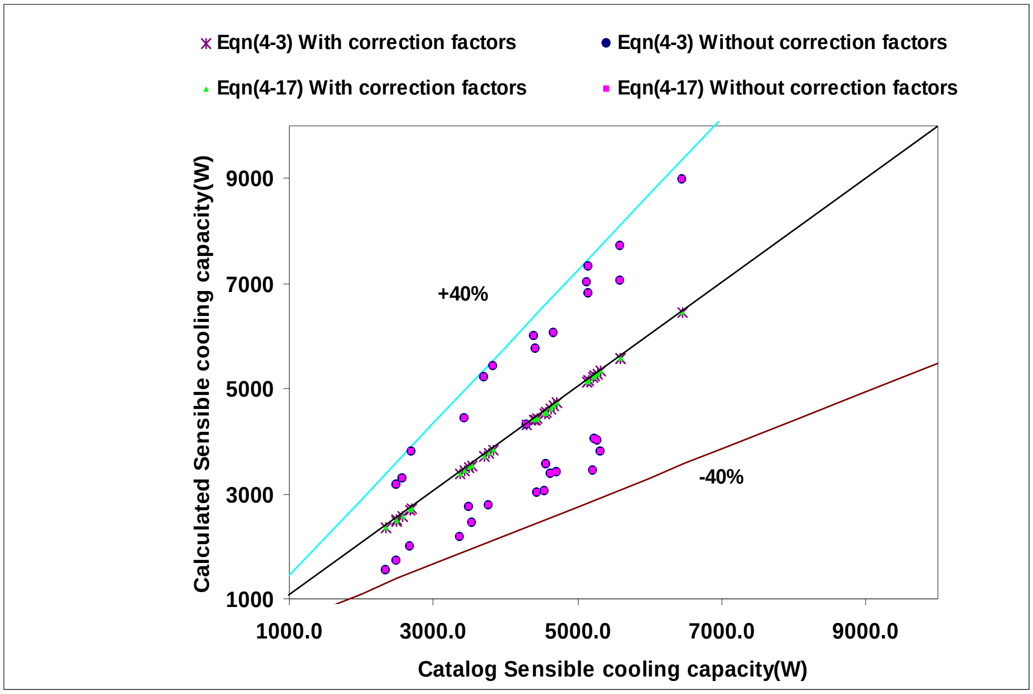

Figure 5.19. *Comparison of catalog v/s calculated sensible cooling capacity with and without the correction factors for wet bulb temperatures*

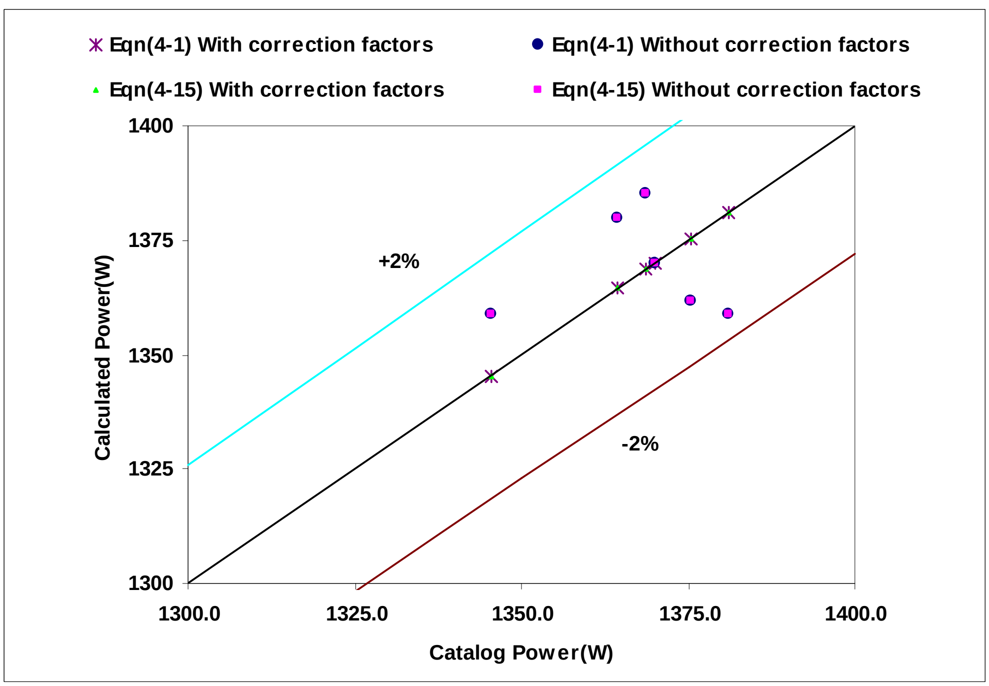

Figure 5.20. *Comparison of catalog v/s calculated power consumption with and without the correction factors for wet bulb temperatures*

Equations 4.1-4.3 are independent of the air mass flow rate and hence will not respond to any load side flow variation. However, at rated conditions (600 CFM), the calculated point coincides with the catalog data point as shown by Figures 5.21-5.23. Equation (4.15)-(4.17) developed in section 4.3 accounts for air flow rate variation and the correction factors make possible a few data points to verify this form of the model. At constant air mass flow rate, $\frac{\dot{m}_{base-a}}{\dot{m}_a} = 1$ and Equations 4-15-4.17 match Equation 4.1-4.3 at the rated condition. Figure 5-22 shows that the sensible/latent split cannot be accurately predicted without including a range of air flow rates in the data set used to generate the coefficients. This is the expected result based on Figure 5.15 and 5.17. As shown in Figure 5.22 the error in the predicted sensible heat transfer rate is about 27%. This is because the sensible heat transfer is more susceptible to a variation in the wet bulb temperature than the air flow rate as seen in Figure 5.15 and 5.17. The error in the predicted power is small (less than 3%) as shown in Figure 5.23. This result is also expected from Figure 5.15 and 5.17.

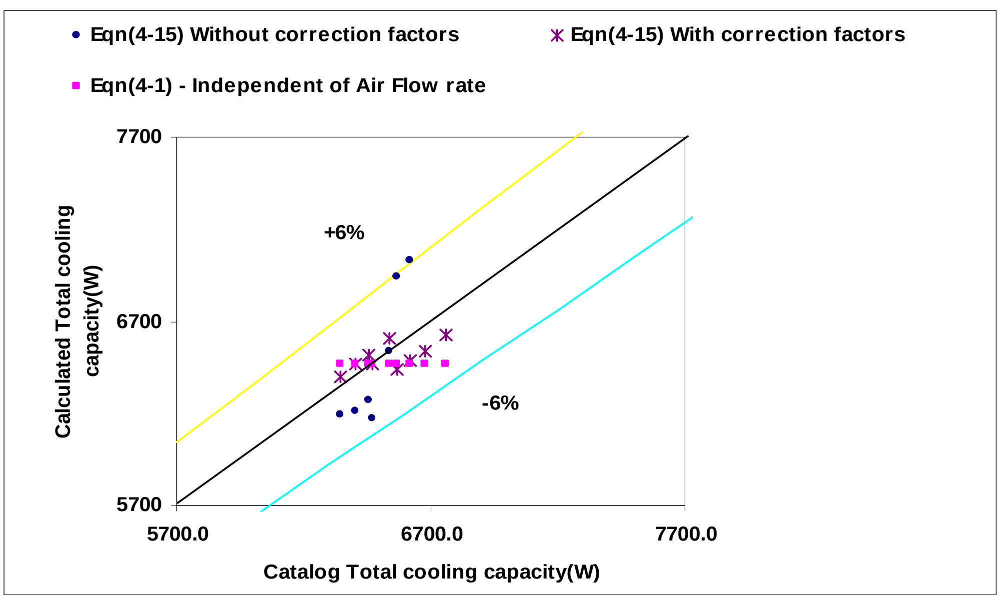

Figure 5.21. *Comparison of catalog v/s calculated cooling capacity with and without the correction factors for variable air flow rates*

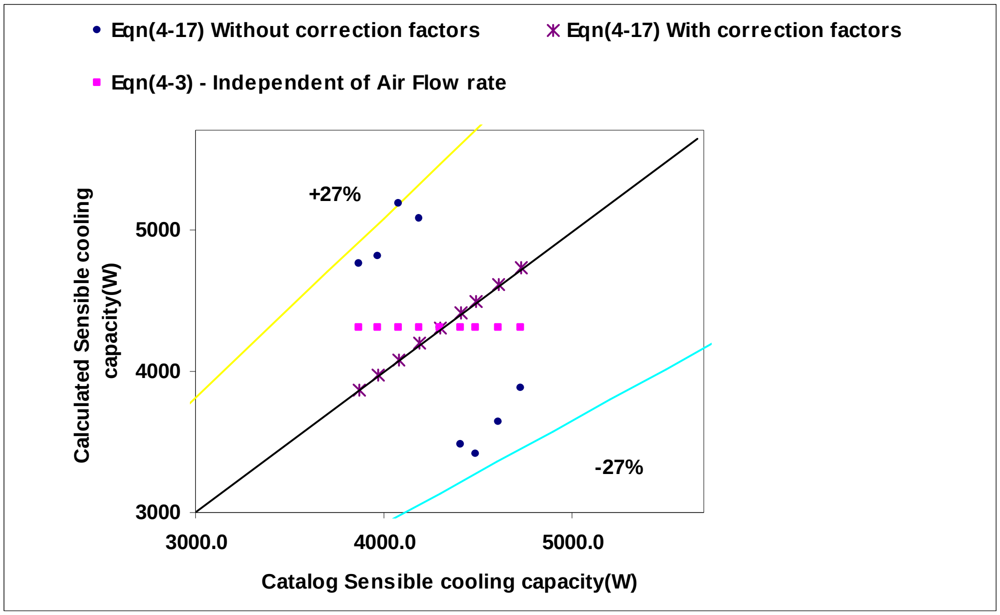

Figure 5.22. *Comparison of catalog v/s calculated sensible capacity with and without the correction factor for variable air flow rates*

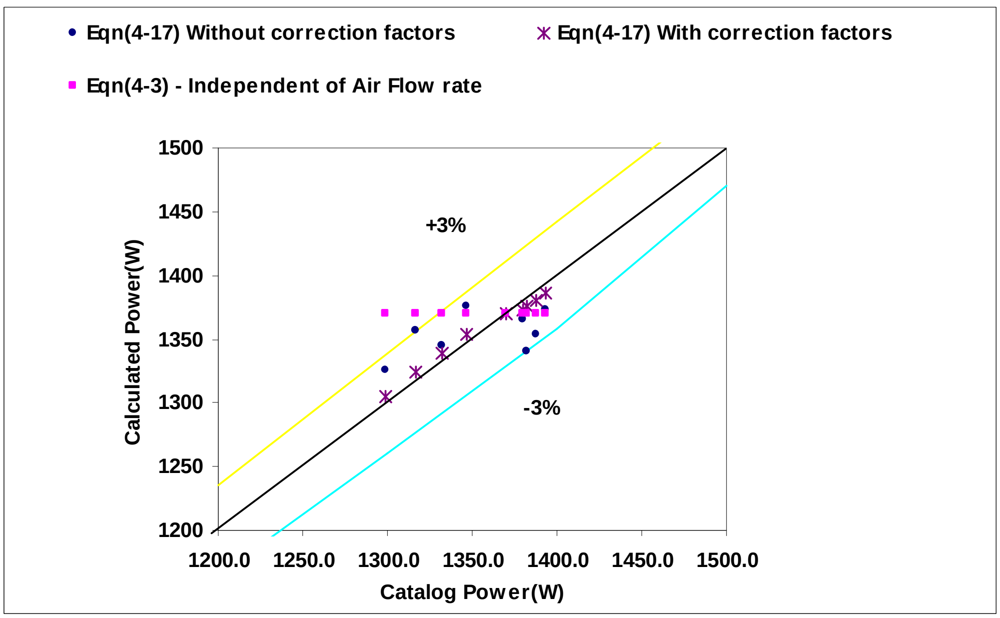

Figure 5.23. *Comparison of catalog v/s calculated power consumption with and without the correction factors for variable air flow rates*

The results recorded in the above comparison plots indicate this mathematical influence on calculations to determine the stability of the implemented model simulation environment.

5.4.3. Application of the model outside the catalog data sets

Jin(2002) validated the detailed model using ClimateMaster HS006 performance data. In the model validation, the HS series correction factors were used to expand the original data over a range of air temperatures and flow rates. The RMS errors from Jin's work are shown for comparison table 5.6.

The simplified model has been validated under identical conditions. The nominal capacity of the heat pump is 7400 Btu/h or 2170 W. ClimateMaster HS006 performance data is merged with the HS series correction factors for the air temperatures and flow rate. Since each correction factor is applied to multiple operating points (i.e. different entering water temperature, water flow rate and air flow rate), these data are presumed to have lower accuracy than the regular tabulated data. 2981 points are generated by applying correction factor to all the operating points. The performance is also calculated using the model equations (4-15) and (4-17). As shown in Figure 5.24 the error in the predicted sensible heat transfer rate is less than 13%.and the error in the predicted total cooling capacity is less than 10% The error in the predicted power is relatively small (less than 8%) as shown in Figure 5.25.

The root-mean-square (RMS) error for the total cooling capacity, sensible cooling capacity and power are compared with the detailed model in Table 5.6. The RMS errors between model prediction and catalog data for the total and sensible capacities are 6.4% and 7.7% respectively. This is slightly larger than the RMS error reported for the detailed model, but still quite reasonable. The RMS error in the predicted power is nearly the same for two models. Overall, the simplified model does a reasonable job of extrapolating beyond the data set.

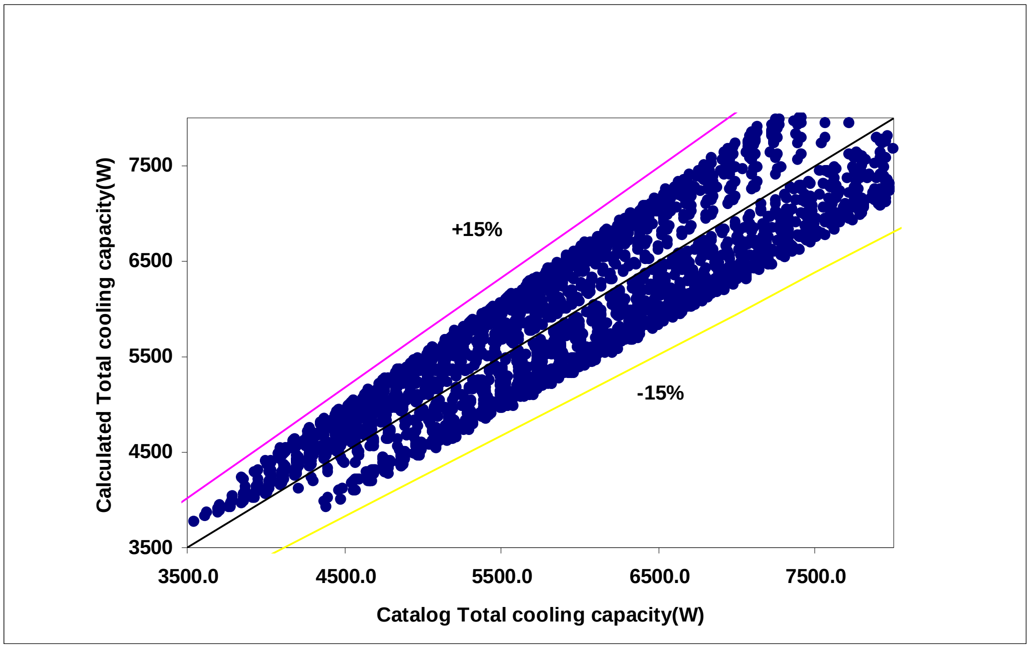

Figure 5.24. *Comparison of catalog v/s calculated cooling capacity for all points with the correction factors(simplified model- equation(4-15))*

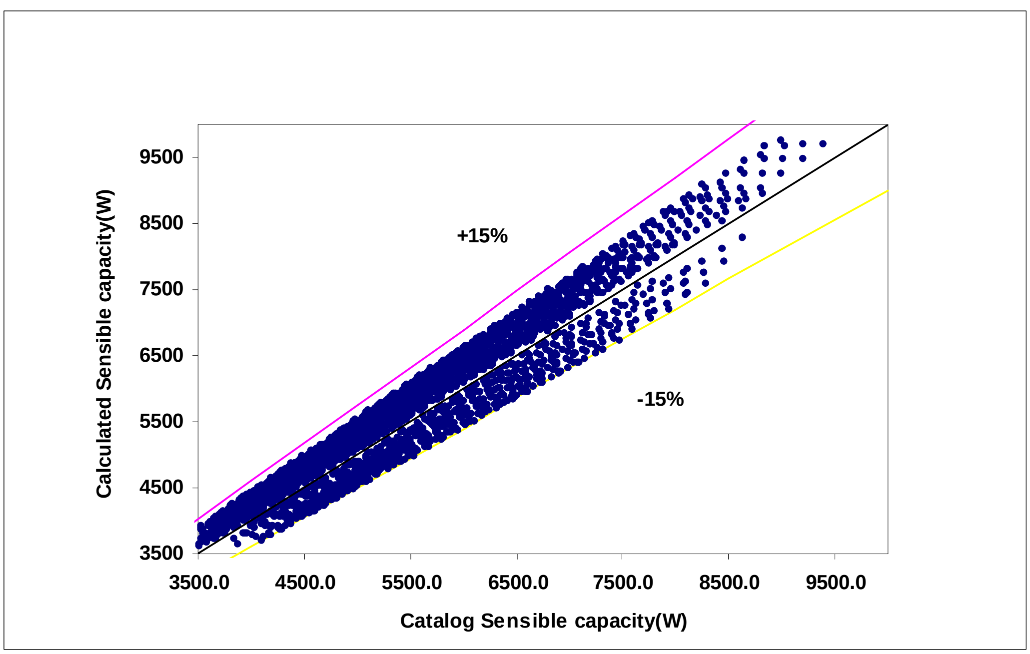

Figure 5.25. *Comparison of catalog v/s calculated sensible capacity for all points with the correction factors(simplified model- equation(4-17))*

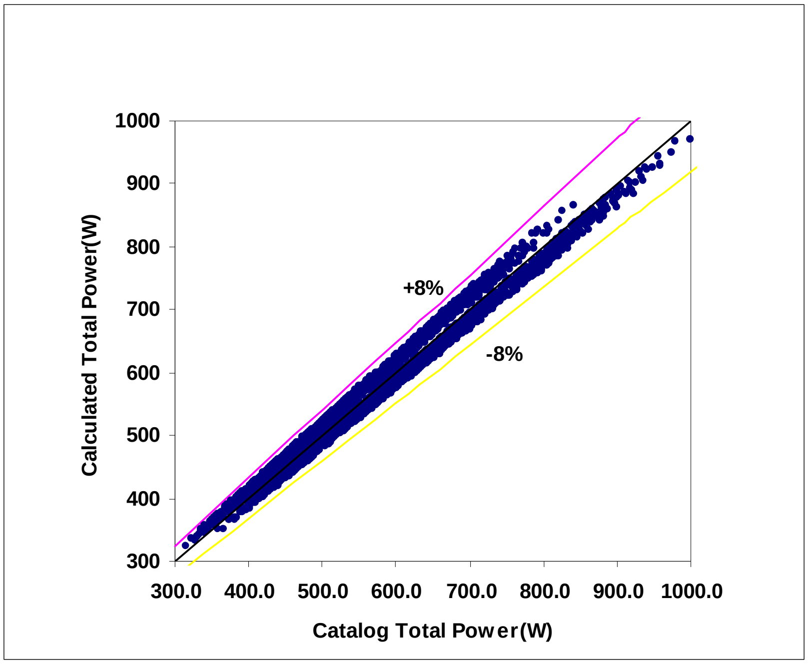

Figure 5.26. *Comparison of catalog v/s calculated power consumption for all points with the correction factors(simplified model)*

Table 5.6. *Comparison of model RMS errors for ClimateMaster HS006*

Performance Variable	RMS Error	
	Detailed Model	**Simple Model**
Total Capacity	4.72%	6.38%
Sensible Capacity	6.33%	7.72%
Power Consumption	6.38%	6.61%

CHAPTER 6

MODEL APPLICATION

6.1.Using manufacturers' data

As previously stated, manufacturers are required to certify the performance of their water-to-air heat pumps using the Air Conditioning and Refrigeration Institute (ARI) Standard 320 and the ISO Standards 13256-1. The standard developed by ASHRAE Standard 90.1, references both ARI and ISO standards, with the ISO standard designated as the executive standard. The following performance metrics are verified by test for a limited set of conditions, usually 3 or 4 points.

- Cooling Capacity, Btu/h [W]
- EER, Btu/w.h [W/W]
- Heating Capacity, Btu/h [W]
- EER, Btu/w.h [W/W]

As discussed in chapter 5, manufacturers only measure a few data points. They use the measured data to generate parameters for their own models. These models are then used to generate tables of catalog data. This catalog data is not necessarily accurate and often consists of physically inconsistent data. Most heat pump manufacturers use air temperatures or the mass flow rates as the basis for measuring the power and the capacities. Florida Heat Pump focuses on keeping the water and air flow rates constant while varying the entering water and air temperatures. ClimateMaster and Trane keep the flow rates constant. Section 5.4 used available correction factors to partially evaluate the effect of data sets with constant parameters (air mass flow rates and wet bulb temperatures. This section evaluates the basic model (equation 4-1 through 4-14) implemented in the EnergyPlus program with coefficients generated from both ClimateMaster and FHP data sets. The results of a 24 hour simulations are compared with results from detailed Jin (2002) model with coefficients generated from the same data sets.

6.1.1. Example 1 - Florida Heat pump (FHP)

The units follow the ISO 13256-1 standards. The nominal capacities of the units can be determined simply by looking at the identification number of each unit. For instance, FHP GT018 is designed with a nominal cooling and heating capacities of 18000 Btu/h and a FHP GT054 with 54000 Btu/h. The units are designed to operate with entering fluid temperatures between 50°F (10°C) and 110°F (43.3°C) in cooling and temperatures between 25°F (-3.9°C) and 80°F (27°C) in heating. All catalog points are at constant air flow and water flow rates as shown in Figure 6.1. For every entering fluid temperature data point there are five different entering air wet bulb temperature data points in the cooling operation of the unit. Moreover, for every wet bulb temperature data point there are three sensible capacity data points at 75°F, 80°F and 85°F. This means that for the Florida Heat Pump, the total cooling capacity is only a function of the dry bulb temperatures and the sensible cooling capacity is a function of both the dry and the wet bulb temperatures. Q_{base} and $\dot{m}_{base}$ are selected at the peak capacities. For the FHPGT018, the peak cooling capacity 20900 Btu/h occurs at an entering water temperature of 50^{o}F and an entering air temperature of 70^{o}F. The water mass flow rate, $m_{base,}$ occurs at a constant flow rate of 4 GPM. This will result in the coefficients of the capacity equations that depend only on the load side and source side temperatures since $\frac{\dot{m}_w}{\dot{m}_{w-base}}$=1 for all data points.

Identification code indicates the Nominal capacity GT018 ---18000 Btu/h

FHP GT018 Data								
All performance at 550 CFM and 4.0 GPM								
Entering	Ent. Air	Total		Heat	Sensible Capacity BTUH			EER
Fluid	Wet Bulb	Capacity	Watts	Rejection	Air Dry Bulb Temp.			
Temp.	Temp.	BTUH	Input	BTUH	75°	80°	85°	
50	61	18259	713	20692	13895	16670	18259	23.4
50	64	19144	729	21632	12999	16502	18302	24
50	67	20044	745	22586	11966	15634	18040	24.6
50	70	20959	761	23556	9976	13791	17941	25.2
50	73	21889	777	24541	-	11754	16110	25.7
60	61	17375	776	20021	13222	15863	17375	19.9
60	64	18217	793	20922	12370	15703	17416	20.4
60	67	19074	810	21838	11387	14878	17167	20.9
60	70	19945	827	22768	9494	13124	17073	21.4
60	73	20830	845	23713	-	11186	15331	21.9
70	61	16464	852	19372	12529	15032	16464	17.7
70	64	17262	871	20234	11721	14880	16503	18.2
70	67	18074	890	21111	10790	14098	16267	18.6
70	70	18899	909	22001	8996	12436	16178	19.1
70	73	19738	928	22905	-	10599	14527	19.5
85	61	14701	994	18091	11187	13422	14701	15.3
85	64	15413	1016	18879	10466	13286	14735	15.7
85	67	16138	1038	19680	9634	12588	14524	16.1
85	70	16875	1060	20493	8032	11104	14445	16.5
85	73	17623	1083	21318	-	9464	12971	16.9
100	61	13318	1125	17156	10135	12159	13318	13.1
100	64	13963	1150	17887	9481	12037	13349	13.4
100	67	14620	1175	18629	8728	11404	13158	13.8
100	70	15287	1200	19383	7277	10059	13086	14.1
100	73	15966	1226	20148	-	8574	11751	14.4

Figure 6.1. *Florida Heat Pump (GT018) cooling catalog data*

6.1.2 Example 2 - ClimateMaster

ClimateMaster holds the air temperature and the air mass flow rate constant to generate their data. Unlike the FHP unit in which the water mass flow rate remains constant through out the specification data, the water mass flow rate is varied between 2.2 and 4.5 gpm. Again, the identification code indicates the nominal capacity of the heat pump unit that is GSV018 indicates a nominal capacity of 18000Btu/h. The design conditions for the heat pump unit occur at 85°F and 4.5 gpm as shown in Figure 6.2. The entering air temperature is 80°F dry bulb and 67°F wet bulb for cooling and 70°F dry bulb for heating as per the ISO standards. Again, Q_{base} and m_{base} are selected at the peak cooling capacities. For the GSH/GSV018, the peak cooling capacity 22400 Btu/h occurs at an entering water temperature of 30^{o}F, an entering air dry bulb temperature of 80^{o}F and wet bulb temperature of 67^{o}F. The volumetric flow rate is 4.5 GPM occurs at the peak capacity. This will result in coefficients of the capacity equations that depend only on source side mass flow rate and temperature since the load side wet bulb temperature is a constant for all data points.

Table does not reflect fan or pump power ISO corrections

600 CFM Nominal Airflow

Performance capacities shown in thousands of Btuh

Base flow rate

Base Capacity

Design Conditions

EWT °F	GPM	WPD		COOLING - EAT 80°F DB/67°F WB						HEATING - EAT 70°F				
		PSI	FT	TC	SC	Sens/Tot Ratio	KW	HR	EER	HC	KW	HE	LAT	COP
20	2.2	0.5	1.2	Operation Not Recommended										
	3.5	1.2	2.7											
	4.5	1.8	4.2							12.1	1.12	8.2	88.6	3.17
30	2.2	0.5	1.2	22.1	14.7	0.66	0.80	24.0	27.5	13.2	1.14	9.3	90.4	3.40
	3.5	1.1	2.6	22.3	14.7	0.66	0.76	24.1	29.5	13.9	1.16	9.9	91.4	3.52
	4.5	1.8	4.1	22.4	14.7	0.66	0.75	24.2	30.0	14.1	1.16	10.2	91.8	3.57
40	2.2	0.5	1.1	21.7	14.7	0.68	0.88	23.6	24.6	15.2	1.18	11.2	93.4	3.77
	3.5	1.1	2.5	22.1	14.7	0.67	0.81	23.7	27.1	16.0	1.20	11.9	94.7	3.92
	4.5	1.7	3.9	22.2	14.7	0.66	0.79	23.8	27.9	16.4	1.21	12.2	95.2	3.98
50	2.2	0.5	1.1	21.0	14.6	0.69	0.98	23.3	21.4	17.2	1.22	13.1	96.6	4.13
	3.5	1.1	2.5	21.6	14.7	0.68	0.90	23.3	24.0	18.2	1.24	14.0	98.1	4.31
	4.5	1.7	3.8	21.7	14.7	0.68	0.87	23.3	25.0	18.6	1.25	14.4	98.8	4.38
60	2.2	0.5	1.0	20.2	14.3	0.71	1.09	23.2	18.4	19.3	1.26	15.0	99.9	4.49
	3.5	1.0	2.4	20.9	14.5	0.70	1.00	23.2	20.9	20.5	1.28	16.1	101.6	4.69
	4.5	1.6	3.7	21.1	14.6	0.69	0.97	23.2	21.8	21.0	1.29	16.6	102.4	4.77
†70	2.2	0.4	1.0	19.3	13.9	0.72	1.22	23.0	15.8	21.5	1.30	17.0	103.1	4.84
	3.5	1.0	2.3	20.0	14.2	0.71	1.12	23.1	17.9	22.8	1.32	18.3	105.2	5.06
	4.5	**1.5**	**3.6**	20.3	14.3	0.71	1.08	23.1	18.8	**23.4**	**1.33**	**18.8**	**106.0**	**5.14**
80	2.2	0.4	1.0	18.3	13.5	0.74	1.36	22.6	13.5	23.6	1.34	19.1	106.5	5.18
	3.5	1.0	2.2	19.1	13.8	0.73	1.25	22.6	15.3	25.1	1.36	20.5	108.8	5.41
	4.5	1.5	3.5	19.4	14.0	0.72	1.21	22.5	16.0	25.7	1.37	21.1	109.7	5.51
†85	**4.5**	**1.5**	**3.4**	**18.8**	**13.7**	**0.73**	**1.28**	**22.3**	**14.8**					
90	2.2	0.4	0.9	17.2	12.9	0.75	1.49	22.1	11.5	25.8	1.37	21.1	109.8	5.51
	3.5	0.9	2.1	18.0	13.3	0.74	1.39	22.1	13.0	27.4	1.40	22.6	112.3	5.76
	4.5	1.4	3.3	18.3	13.5	0.74	1.35	22.0	13.6	28.1	1.40	23.3	113.3	5.85
100	2.2	0.4	0.9	16.1	12.2	0.76	1.63	21.5	9.8	Operation Not Recommended				
	3.5	0.9	2.1	16.9	12.7	0.75	1.53	21.4	11.0					
	4.5	1.4	3.2	17.2	12.9	0.75	1.49	21.3	11.6					
110	2.2	0.4	0.9	14.9	11.5	0.77	1.77	20.9	8.4					
	3.5	0.9	2.0	15.7	12.0	0.76	1.67	20.8	9.4					
	4.5	1.3	3.1	16.1	12.2	0.76	1.63	20.7	9.8					

†ARI/ASHRAE/ISO 13256-1 (WLHP applications) certified conditions are 86°F EWT, 80.6°F DB/66.2°F WB EAT in cooling and 68°F EWT, 68°F DB/59°F WB EAT in heating. Data in **bold print** is shown for application performance only, as these conditions are typical for WSHP systems.

Rev: 06/14/04D

Interpolation is permissable, extrapolation is not.
All entering air conditions are 80°F DB and 67°F WB in cooling and 70°F DB in heating.
ARI/ASHRAE/ISO 13256-1 (GLHP application) certified conditions are 77°F EWT, 80.6°F DB and 66.2°F WB in cooling and 32°F EWT, 68°F DB and 59°F WB in heating.
All performance data is based upon the lower voltage of dual voltage rated units.
See Performance Correction Tables for operating conditions other than those listed above.
Operation below 60°F EWT requires optional insulated water circuit.
Operation below 40°F EWT is based upon 15% antifreeze solution.

Figure 6.2. *ClimateMaster heat pump (GSH/GSV018) catalog data (used with permission)*

6.1.3 Example 3 – Trane

Trane is similar to ClimateMaster in its approach to following the standard rating conditions. The performance data is based on constant air dry bulb and wet bulb temperatures with varying water mass flow rates and entering water temperatures. GEH/V006 indicates a nominal capacity of 6000Btu/h. The entering air temperature is 80.6°F dry bulb and 67°F wet bulb for cooling and 68°F dry bulb for heating as per the ISO standards. For the GEH/V006, the peak cooling capacity or Q_{base} is 9200 Btu/h and occurs at an entering water temperature of 45^{o}F ,an entering air dry bulb temperature of 80^{o}F and wet bulb temperature of 67^{o}F. The volumetric flow rate is 1.2 GPM at the peak capacity.

TRANRE GEH006 Data						
Rated GPM 1.8						
Rated CFM 242						
	Entering Fluid Temp. EWT	Ent. Water Mass flow rate GPM	Total Capacity BTUH	Watts Input	COP	Heat Absorption BTUH
	25	1.2	5.8	0.52	3.3	4
	25	1.5	5.8	0.52	3.3	4
	25	1.7	5.8	0.53	3.2	4
	25	1.8	5.8	0.53	3.2	4
	25	1.9	5.9	0.53	3.3	4.1
	25	2	5.9	0.53	3.3	4.1
	25	2.2	5.9	0.53	3.3	4.1
	35	1.2	6.2	0.55	3.3	4.3
	35	1.5	6.3	0.55	3.3	4.4
	35	1.7	6.4	0.56	3.3	4.5
	35	1.8	6.4	0.56	3.4	4.5
	35	1.9	6.4	0.56	3.4	4.5
	35	2	6.4	0.56	3.4	4.5
	35	2.2	6.5	0.56	3.4	4.6
	45	1.2	7	0.57	3.6	5.1
	45	1.5	7.2	0.57	3.7	5.3
	45	1.7	7.3	0.57	3.7	5.3
	45	1.8	7.4	0.58	3.7	5.4
	45	1.9	7.4	0.58	3.7	5.4
	45	2	7.4	0.58	3.7	5.4
	45	2.2	7.5	0.58	3.8	5.5
	55	1.2	8	0.59	4	6
	55	1.5	8.2	0.6	4	6.1
	55	1.7	8.3	0.6	4	6.2
	55	1.8	8.4	0.6	4.1	6.3
	55	1.9	8.4	0.6	4.1	6.4
	55	2	8.4	0.6	4.1	6.4
	55	2.2	8.5	0.6	4.2	6.5
	68	1.2	9.2	0.63	4.3	7.1
	68	1.5	9.5	0.63	4.4	7.3
	68	1.7	9.6	0.63	4.5	7.4
	68	1.8	9.7	0.64	4.5	7.5
	68	1.9	9.7	0.64	4.5	7.5
	68	2	9.7	0.64	4.5	7.6
	68	2.2	9.7	0.64	4.5	7.6
	75	1.2	9.9	0.64	4.6	7.8
	75	1.5	10.1	0.64	4.6	7.9
	75	1.7	10.2	0.64	4.6	8
	75	1.8	10.3	0.64	4.7	8.1
	75	1.9	10.3	0.64	4.7	8.1
	75	2	10.4	0.64	4.7	8.2
	75	2.2	10.4	0.65	4.7	8.2
	86	1.2	10.8	0.66	4.8	8.5
	86	1.5	11	0.66	4.9	8.7
	86	1.7	11	0.66	4.9	8.8
	86	1.8	11.1	0.66	4.9	8.9
	86	1.9	11.2	0.67	4.9	8.9
	86	2	11.2	0.67	4.9	8.9
	86	2.2	11.3	0.67	4.9	9

.**Figure 6.3.** *TRANE heat pump (GEH/V 006) heating catalog data*

Identification code indicates the Nominal capacity GEH/V 006 ---6000 Btu/h

TRANE GEH006 Data							
Rated GPM 1.8							
Rated CFM 242							
	Entering Fluid Temp. EWT	Ent. Water Mass flow rate GPM	Total Capacity BTUH	Sensible Capacity BTUH	Watts Input	EER	Heat Rejection BTUH
	45	1.2	9.2	6.4	0.49	18.9	10.9
	45	1.5	9.2	6.4	0.47	19.6	10.8
	45	1.7	9.3	6.4	0.46	20.2	10.9
	45	1.8	9.3	6.4	0.45	20.6	10.9
	45	1.9	9.3	6.4	0.45	20.6	10.9
	45	2	9.3	6.4	0.45	20.6	10.9
	45	2.2	9.3	6.4	0.45	20.6	10.9
	55	1.2	8.8	6.2	0.49	18	10.4
	55	1.5	8.9	6.2	0.47	18.9	10.5
	55	1.7	8.9	6.2	0.46	19.2	10.4
	55	1.8	8.9	6.2	0.45	19.6	10.4
	55	1.9	8.9	6.2	0.45	19.6	10.4
	55	2	8.9	6.2	0.45	19.6	10.4
	55	2.2	8.9	6.2	0.45	19.6	10.4
	68	1.2	8.2	5.9	0.54	15.2	10
	68	1.5	8.3	6	0.52	15.8	10
	68	1.7	8.3	6	0.51	16.1	10
	68	1.8	8.3	6	0.5	16.4	10
	68	1.9	8.3	6	0.5	16.7	10
	68	2	8.3	6	0.5	16.7	10
	68	2.2	8.3	6	0.5	16.7	10
	75	1.2	7.8	5.8	0.58	13.4	9.8
	75	1.5	7.9	5.8	0.57	14	9.8
	75	1.7	7.9	5.8	0.55	14.4	9.8
	75	1.8	7.9	5.8	0.54	14.7	9.8
	75	1.9	8	5.8	0.54	14.8	9.8
	75	2	8	5.9	0.54	14.8	9.8
	75	2.2	8	5.9	0.54	14.8	9.8
	86	1.2	7.4	5.6	0.66	11.2	9.7
	86	1.5	7.4	5.7	0.64	11.5	9.6
	86	1.7	7.4	5.7	0.64	11.6	9.6
	86	1.8	7.5	5.7	0.63	11.9	9.6
	86	1.9	7.5	5.7	0.62	12.1	9.6
	86	2	7.5	5.7	0.62	12.1	9.6
	86	2.2	7.5	5.7	0.62	12.1	9.6
	95	1.2	7	5.4	0.73	9.5	9.5
	95	1.5	7	5.5	0.71	9.9	9.5
	95	1.7	7	5.5	0.7	10	9.5
	95	1.8	7	5.5	0.7	10.1	9.4
	95	1.9	7	5.5	0.7	10.1	9.4
	95	2	7	5.5	0.69	10.3	9.4
	95	2.2	7	5.5	0.69	10.3	9.4
	105	1.2	6.5	5.2	0.81	8.1	9.3
	105	1.5	6.5	5.2	0.8	8.2	9.3
	105	1.7	6.6	5.2	0.78	8.4	9.3
	105	1.8	6.6	5.2	0.77	8.5	9.3
	105	1.9	6.6	5.2	0.77	8.5	9.3
	105	2	6.6	5.2	0.77	8.6	9.2
	105	2.2	6.6	5.2	0.77	8.6	9.2
	115	1.2	5.9	5	0.88	6.7	8.9
	115	1.5	5.9	5	0.87	6.8	8.9
	115	1.7	6	5	0.85	7	8.9
	115	1.8	6	5	0.84	7.1	8.9
	115	1.9	6	5	0.84	7.1	8.9
	115	2	6	5	0.84	7.1	8.9
	115	2.2	6	5	0.84	7.2	8.9
	120	1.2	5.4	4.9	0.9	6	8.5
	120	1.5	5.5	4.8	0.9	6.1	8.5
	120	1.7	5.5	4.8	0.89	6.2	8.5
	120	1.8	5.6	4.8	0.88	6.3	8.6
	120	1.9	5.6	4.8	0.87	6.4	8.5
	120	2	5.6	4.8	0.87	6.4	8.5
	120	2.2	5.6	4.8	0.87	6.4	8.5

.**Figure 6.4.** *TRANE heat pump (GEH/V 006) Cooling catalog data*

6.2.Case study

The simplified water-to-air heat pump model that was implemented in EnergyPlus as described in the previous chapters is evaluated by comparing and analyzing its performance in the simulation environment. The case study was carried out on a typical office building assumed to be located at Chanute AFB, Illinois. An annual building loads simulation along with simulations for the summer design day (21st July) and the winter design day (21st January) were run for this region. Results obtained from the simplified model are compared with detailed model (Jin,2002) results for the same building, system and environmental conditions.

6.2.1. Example building and system description

The example building shown in Figure 6.5 has an area of 108 m^2. The zones are served by a water-to-air heat pump unit that includes a supplemental heating coil.

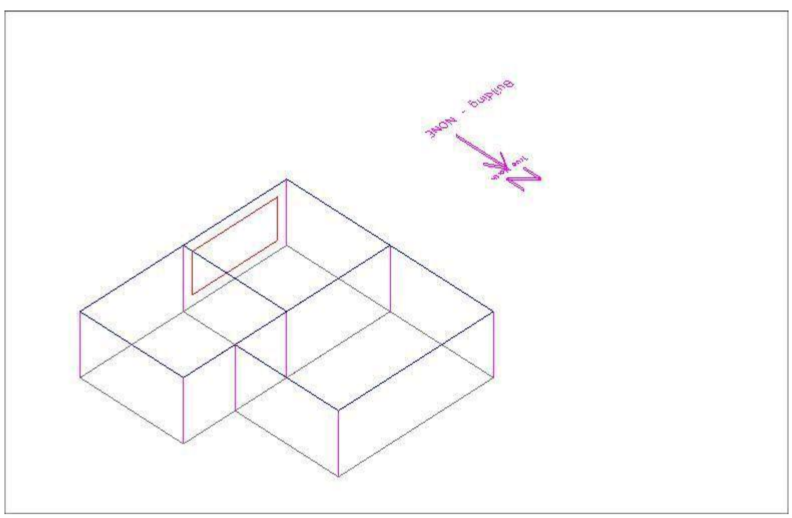

Figure 6.5. *Isometric north east view of the building plan*

A summary of additional assumptions and a brief description of the building and system are shown below.

- Typical building with 3 thermal zones.
- No ground contact (all floors are adiabatic).
- Roofs are exposed to the outdoor environment
- There is one large south facing single pane window located in the southwest zone.
- The air handling system is modeled as a blow through system.
- The lighting loads are 7.6 W/m^2. The electric equipment plug load is 12.1W/m^2.
- The office occupancy is assumed to be one person per 10.7 m^2 with a total heat gain of 131.7 Watts/Person of which 30% is assumed to be radiant heat gain.
- A single water-to-air heat pump serves all zones.
- The controlling zone is the east zone.
- The heating set point during winter months is 20°C during occupied hours, 15°C set-back otherwise. Cooling set point is at 24°C during occupied hours only.
- The heat pump is scheduled to be unavailable during unoccupied hours.

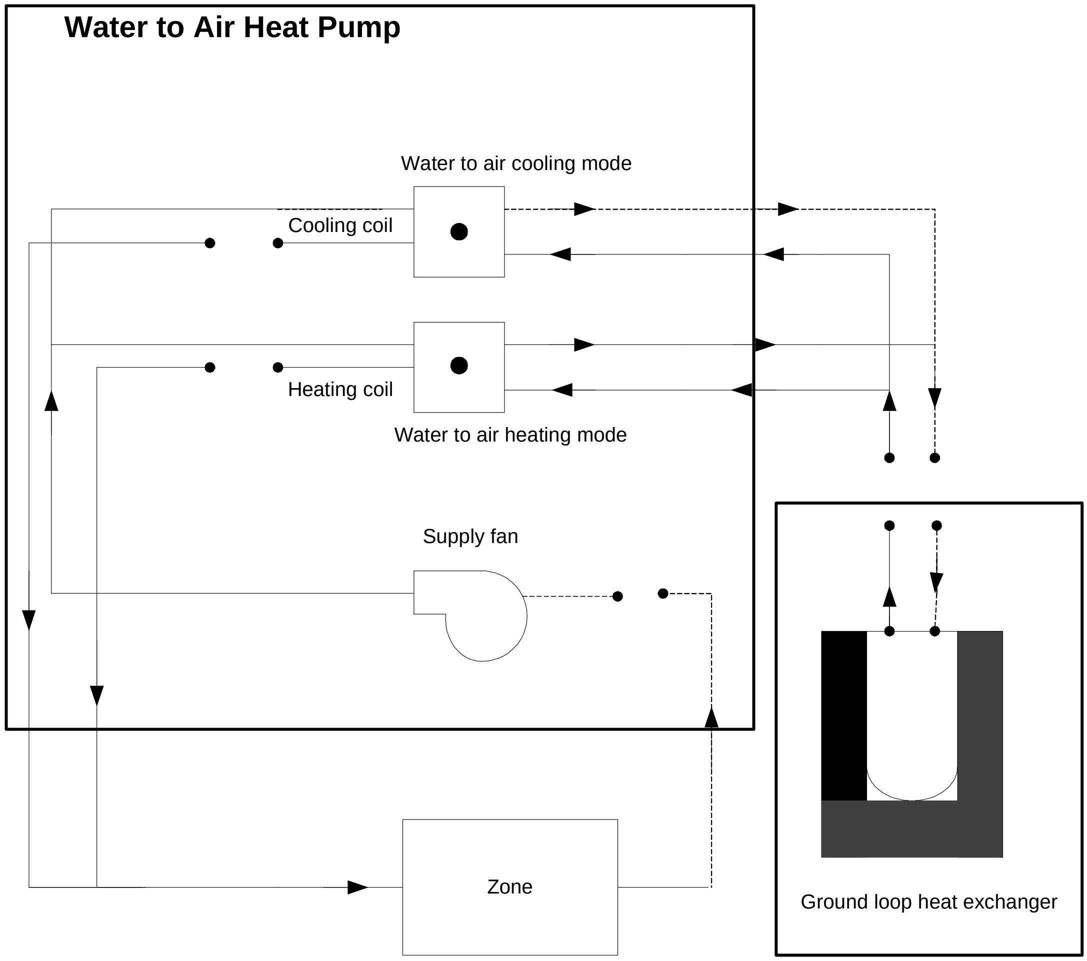

Figure 6.6. *Schematic of the building and system plan implemented in EnergyPlus*

6.2.2. System connections and configuration

In the case of a water-to-air ground source heat pump as shown in Figure 6.6, the condenser loop is directly connected to the air loop. The condenser loop has a constant speed circulating pump that operates continuously. The heat pumps runs with design flow rates on both the load and the source sides. The design water mass flow rate is set at 1.7 kg/s and the design load side mass flow rate is set at 2 kg/s. The heating coil of the single reversible packaged unit is available for operation during winter and the cooling coil in summer. The supplemental gas heating coil is assumed to have a nominal capacity of 32000W. A simple ON-OFF supply fan with a total efficiency of 70% is used. The fan outlet node forms the inlet node to the heating/cooling coils with in the heat pump. The outlet node of the heating coil forms the inlet node to the supplemental coil. The outlet node of the supplemental coil becomes the outlet node of the heat pump to complete the ground loop configuration.

6.2.3 Annual and design day building load profiles

Fig 6.7 shows the hourly zone cooling and heating loads for the example building when simulated with Chanute AFB, Illinois weather data. Heating loads are shown as positive loads and cooling as negative loads. The peak heating load is around 15KW and the peak cooling load is around 16KW. The building is well balanced in terms of the heating and cooling loads. In order to verify the correctness of results generated by the simulation, the heat pump was operated in the heating and the cooling mode. For the winter design day, that is Jan 21st, the heating mode of the heat pump is triggered. For the summer design day, that is July 21st, the cooling mode of the heat pump is activated. The schedule is an active day schedule for both design days, which means that the design day is a typical Monday. The load profile for the winter design day is shown in Fig 6.8. Although typically, a winter design day includes no scheduled load, some scheduled loads were included to exercise the model.

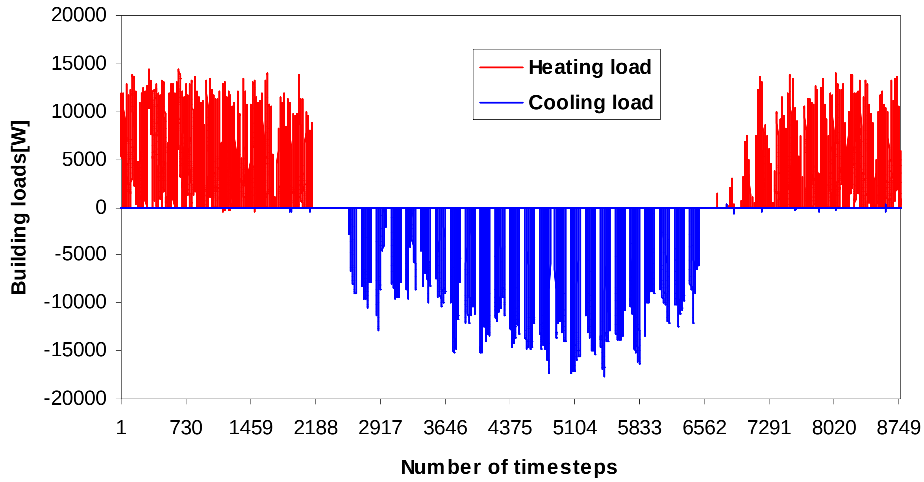

Figure 6.7. *Annual hourly loads for the example building in Chanute AFB,IL*

Figure 6.8 shows the winter design day heating load. From the 1st time step through the 42nd time step, all electric equipment is scheduled OFF and the building is unoccupied. At 7 a.m. (the 43rd time step), the building experiences a high pickup load as the system comes out of setback. Starting at 7a.m. (timestep 43) office occupancy, lighting and all electric equipment schedules begin to ramp up. This along with the solar heat gain largely offset, the heating load which steadily decreases until the activity comes to an end at 5p.m. At 5p.m the building suddenly experiences a major drop in the heating load as the system returns to setback. The load is zero until the sun sets and the thermal mass of the building cooling and which point the heating load steadily increases until the system comes out of setback.

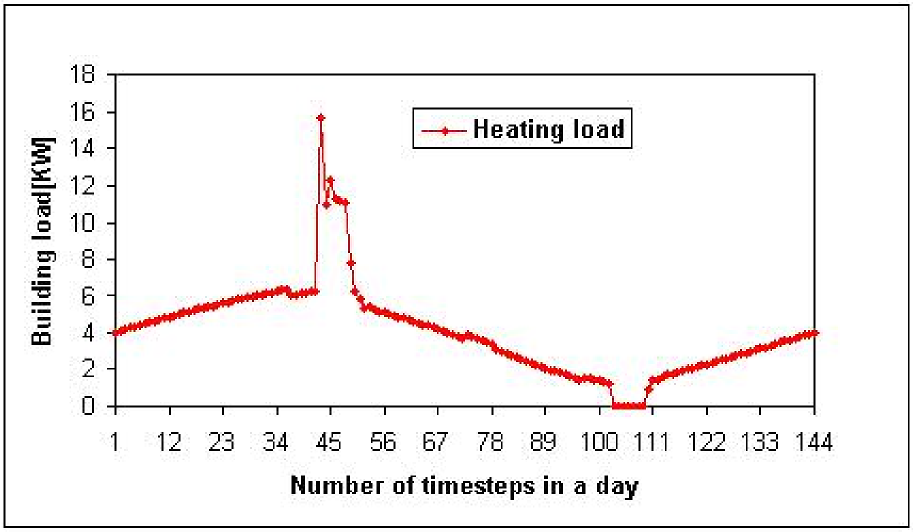

Figure 6.8. *Building load profile for the winter design day*

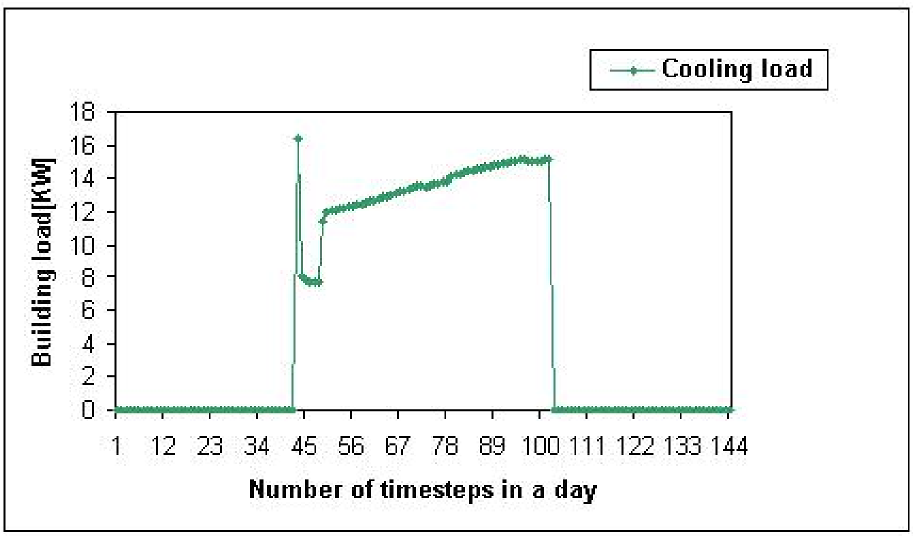

Figure 6.9. *Building load profile for the summer design day*

The load profile for the summer design day is shown in Fig 6.9 and may be analyzed as follows. The peak cooling load is about 17KW. The pick up load again occurs at 7 a.m. as activity and system schedules are initiated. The maximum dry bulb temperature of 32°C and solar radiation contributes to the higher peak load. Once the system stabilizes, the building load steadily increases due to the effect of increasing dry bulb temperature, solar heat gains and scheduled loads until the system returns to setback.

6.2.4 Selection of heat pump

A Florida heat pump GT120 having a nominal capacity of 38KW is selected for the typical office building. The heat pump is modeled as a single heat pump which switches the control to the heating or cooling coils depending on the mode in which the heat pump is required to operate. Although the heat pump is modeled to operate throughout the year, a schedule is maintained to handle the demands appropriately. The cooling mode is made available only in summer and the heating mode is available only in winter. The nominal capacity of the heat pump is set at 38KW with a nominal COP of 4.5 in the heating mode and a nominal EER of 20 in the cooling mode. The nominal values are characteristic of the GT120 Florida heat pump used in this study. Using the nominal values above and the catalog data for the GT120 unit, 17 coefficients were generated as shown in Table 6.1 by using the performance coefficient calculator discussed in the next chapter.

Table 6-1. *Distribution of coefficients in model equations*

Performance Variables	No. of Coefficients in the Performance Equations
Cooling Capacity	3
Heating Capacity	3
COP	3
EER	3
Sensible Cooling Capacity	5

6.2.5. Cooling mode operation and analysis

Based on the control zone sensible load , represented by the east zone in this case study and the fraction of air flow through the control zone, the total cooling sensible load demanded by all the zones being served is given by equation (6.1).

$$HeatPumpCoolingLoad = \frac{ControlZoneCoolingLoad}{ControlZoneAirFlowFraction} \ldots\ldots\ldots\ldots\ldots\ldots\ldots.. (6.1)$$

On the condenser demand side all the components are already simulated and controlled by the air loop. The condenser demand side manager is responsible for supplying the specified flow rate through the source side coil.

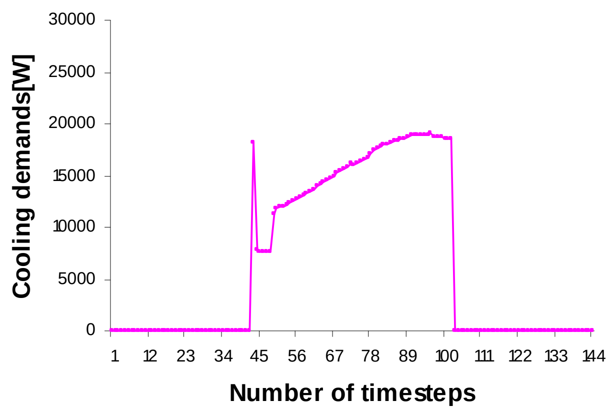

Figure 6.10. *Demands for the summer design day*

The load side coil demand, shown in Figure 6.10, peaks late in the afternoon, unlike the building load (Figure 6.9) which reaches a high point as the office building begins its operation at 7a.m.This is due to the effect of outside air which increasingly adds to the coil demand as the outside temperature rises. To meet this demand, the heat pump coils and fans must be configured and scheduled appropriately. The heat pump may be configured for 'blow-through' or 'draw-through' operation with the fan operating in either of two modes: 'cycling' or 'continuous'.

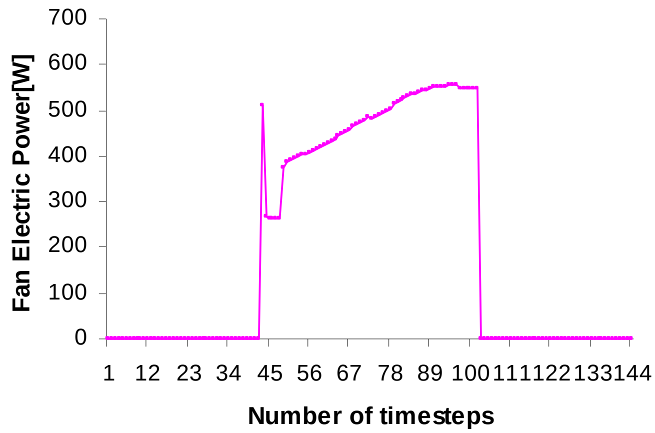

Figure 6.11. *Fan electric power consumption (cooling mode)*

The profile of the fan electric power consumption as shown in Figure 6.11 is identical to the demand that must be met by the heat pump. A simple algorithm switches the fan OFF when the cooling load to be met by the heat pump is zero and switches the fan ON when it is greater than zero. The fan power consumption is directly proportional to the air mass flow rate. The flow rate is resolved as the program iterates through each node and determines what the flow requests and flow limits are.

Figure 6.12 demonstrates point to point synchronization between the cooling demand and the demand met. The heat pump is operated at full load and its capacity under full load conditions is determined. The run time fraction in cooling mode is calculated by using a simple control strategy as shown by equation 6.2.

$$RuntimeFractionCool = \frac{TotalCoolingdemand}{BaseCoolingCapacity} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \; (6.2)$$

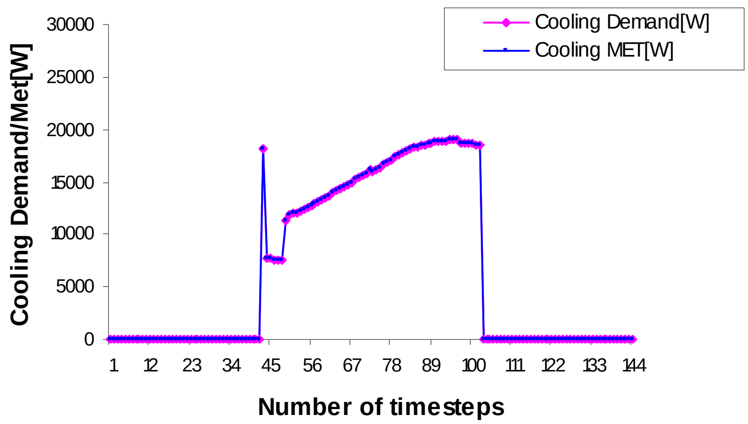

Figure 6.12. *Cooling demands met by the heat pump*

Once the runtime fraction is computed, the total capacity under full load conditions is multiplied by the runtime to compute the actual capacity. The other performance variables such as heat pump power consumption and energy added to the air stream by the heat pump are calculated in the same manner as shown in the following equations.

$$ActualCapacity = TotalCapacity_{fullload} * RuntimeFractionCool \ldots\ldots\ldots\ldots \text{(6.3)}$$

$$ActualPower = TotalPower_{fullload} * RuntimeFractionCool \ldots\ldots\ldots\ldots \text{(6.4)}$$

$$ActualGroundHeatTrans = TotalGroundHeatTrans_{fullload} * RuntimeFractionCool \text{ (6.5)}$$

The performance variables depend on the requested water mass flow rate. In the case study, the east zone temperatures are controlled within the bounds of the set point at 24°C when the building is occupied and the heat pump is operating. The temperature peaks high for a few time steps when the system is switched off and then gradually drops to the set point at 30°C. The sudden rise and drop in the temperatures, which occur at 7 a.m and 5 p.m occur as the system comes out of setback or goes into setback, are shown in Figure 6.13.

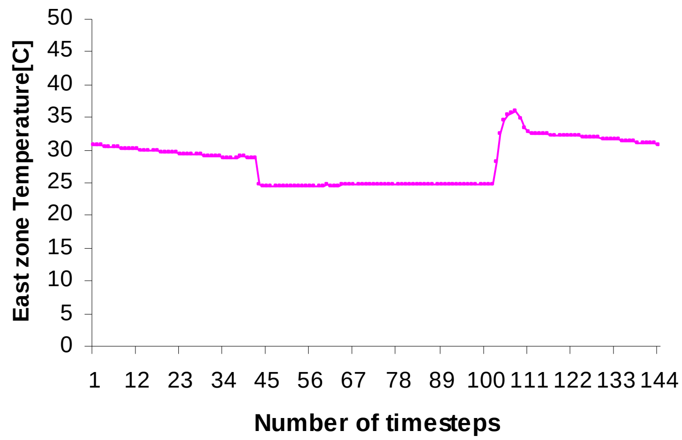

Figure 6.13. *Controlled zone temperatures (cooling mode)*

6.2.6. Heating mode operation and analysis

Analogous to the cooling mode operation, the total heating sensible load demanded by all the zones being served is given by Equation (6.6).

$$HeatPumpHeatingLoad = \frac{ControlZoneHeatingLoad}{ControlZoneAirFlowFraction} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots (6.6)$$

The demand picks up at 7a.m as the building begins its operation as shown in Figure 6.14. Once the building stabilizes the daytime heating demand decreases gradually due to the presence of scheduled internal heat gains. Analogous to the cooling mode, the fan switches OFF when the heating load to be met by the heat pump is zero and switches ON when it is greater than zero. The profile of the fan electric power consumption as shown in Figure 6.15 is identical to the demands needed to be met by the heat pump.

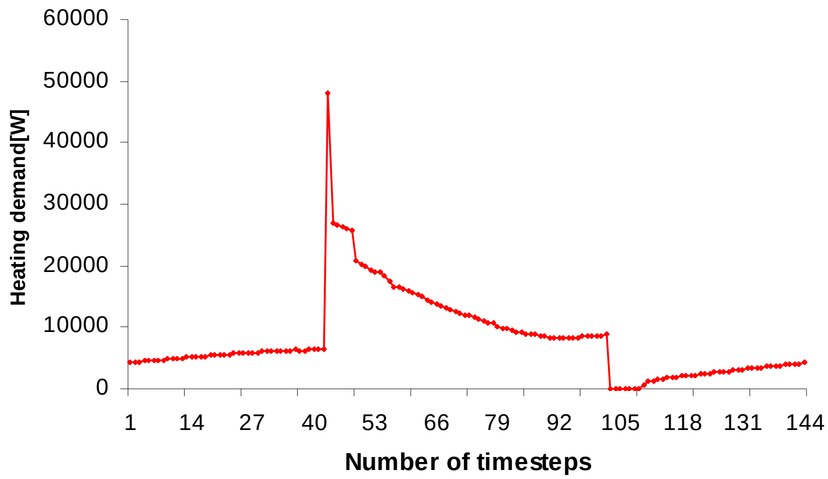

Figure 6.14. *Demands for the winter design day*

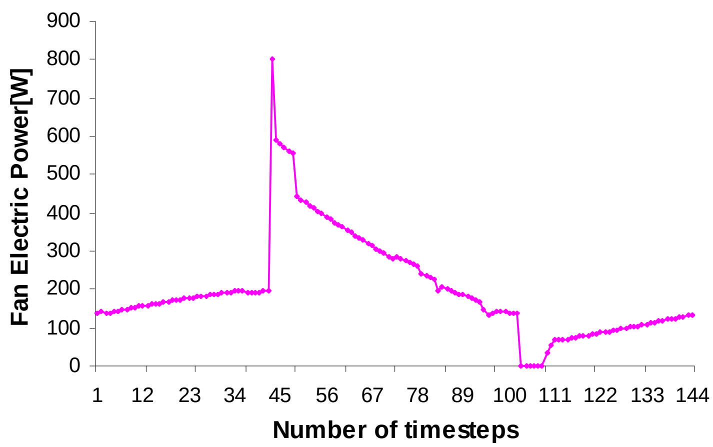

Figure 6.15. *Fan electric power consumption (heating mode)*

Figure 6.16 shows excellent agreement between the heating demand and the demand met. The heat pump is operated at full load and its heating capacity under full load conditions is determined.

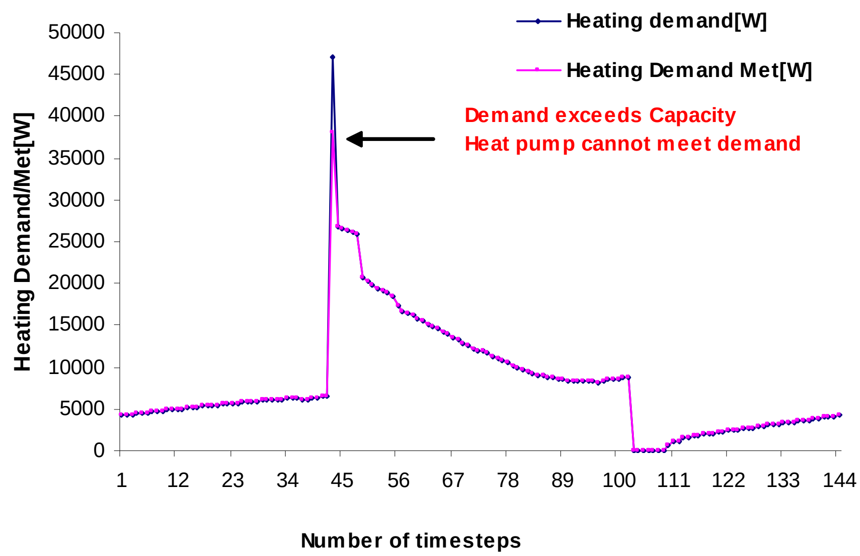

Figure 6.16. *Heating demand v/s demand met*

The run time fraction, actual heating capacity and the actual power for the heating load is calculated using equation 6.3-6.6. In the case study, the east zone temperature is controlled within the bounds of the set point at 20°C when the building is occupied and the heat pump is operating. The temperature peaks high for a few time steps when the system is switched off and then gradually drops to the set point at 15°C as shown in Figure 6.17.

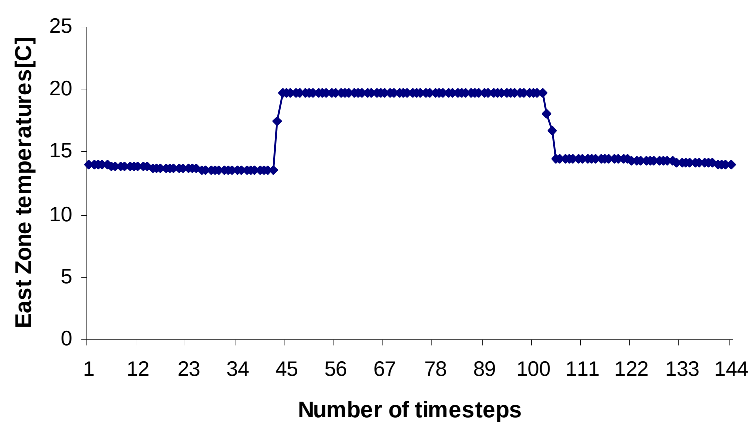

Figure 6.17. *Controlled zone temperatures (heating mode)*

The sudden rise and drop in the temperatures occur at 7 a.m and 5 p.m occur as the system comes out of setback or goes into setback.. A few stand alone points show up between the 43rd and the 48th time step and then again between the 108th time step and the 111th time step as the building system tries to approach a steady state configuration. Some other important plots which are a significant part of this case study are discussed and compared with the detailed model in the next section.

6.2.7. Comparison with the Detailed Model

The building discussed in the preceding sections is simulated again in EnergyPlus using the existing detailed model (Jin, Spitler; 2002). It is a parameter estimation based model which incorporates a multivariable unconstrained optimization algorithm to estimate model parameters. The detailed model is discussed in section 2.2.1.

Comparison of the detailed and the simplified model shows that the new model is in good agreement with the detailed model when the heat pump operates at the rated air mass flow rate and wet bulb temperature. The demands, demand met, duty factor and the power consumption calculated using the detailed model match the simplified model at every point as shown by Figures 6.18 and 6.19.

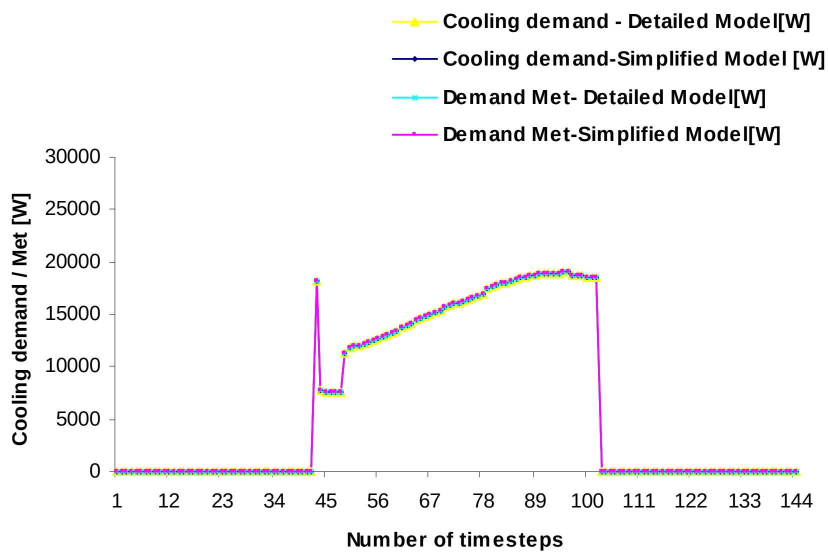

Figure 6.18. *Comparison of demand v/s met in the cooling mode*

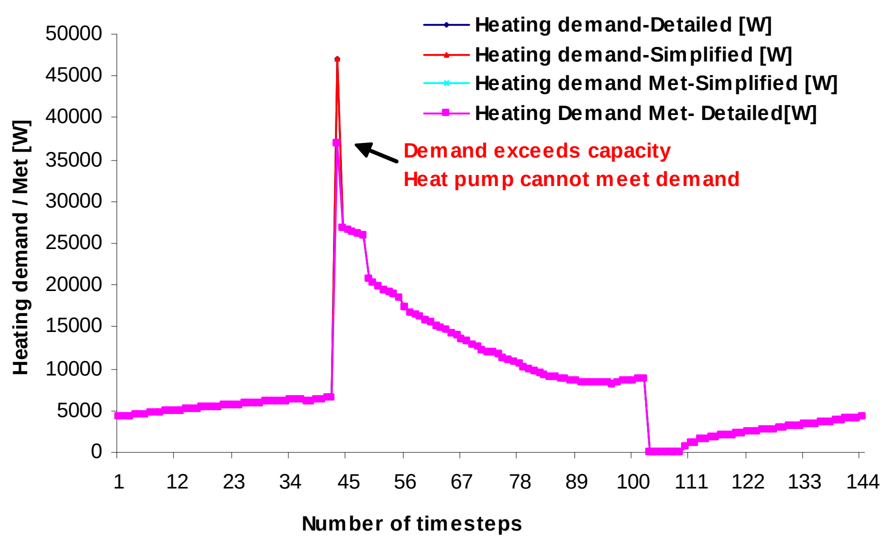

Figure 6.19. *Comparison of demand v/s met in the heating mode*

In cooling mode the heat pump operates with the highest duty factor of 0.6 in the 43rd time step due to the pick-up load as shown in Figure 6.20. At this point the heat pump has to meet a capacity of about 20000W with a power consumption of about 5000W. In the heating mode the heat pump has to operate at full capacity to meet a demand of about 45000W as shown in Figure 6.21. At this point the duty factor is 1 which means that the heat pump is operating at its peak. The points that do not occur along the smooth curve represent the time steps in which the system attempts to attain a steady state after experiencing a huge pick up load.

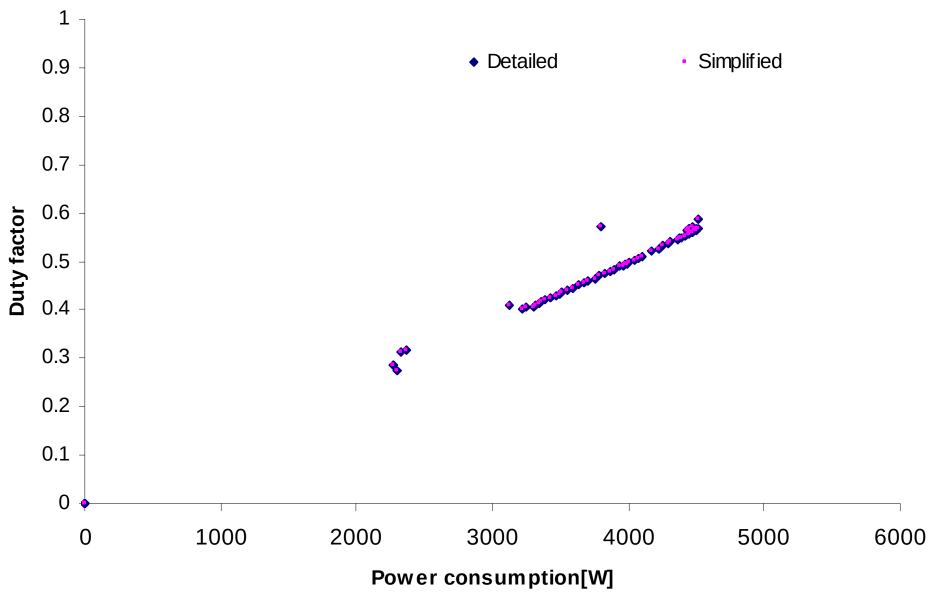

Figure 6.20. *Comparison of duty factor v/s power consumption in the cooling mode*

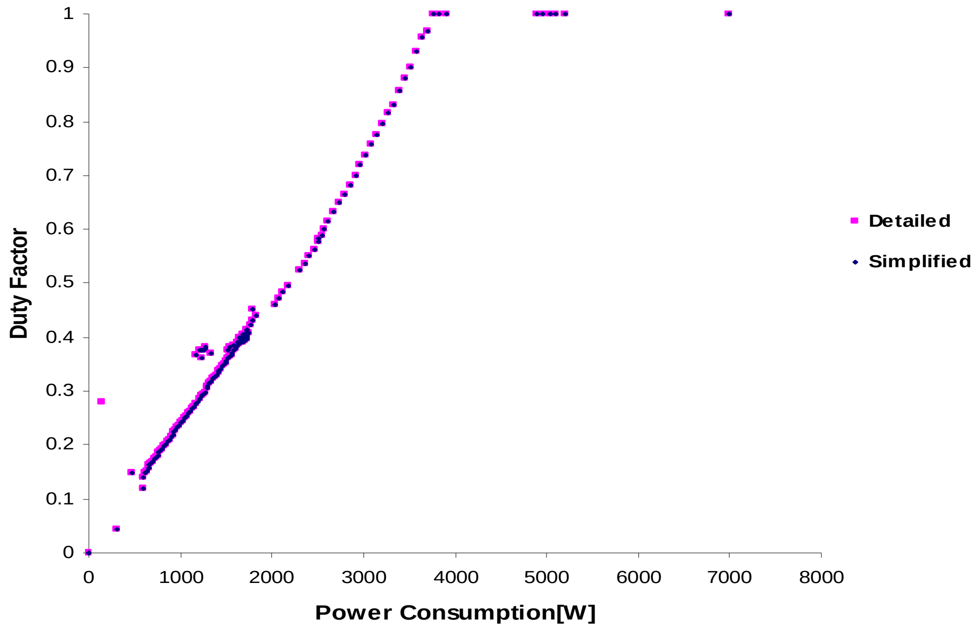

Figure 6.21. *Comparison of duty factor v/s power consumption in the heating mode*

6.2.8. Model performance under off design conditions

The design water flow rate at which the heat pump operates is 4.5 GPM. The heat pump off-design performance is observed by changing the design flow rate to 4 GPM. The profile for both the heating and the cooling modes under off design conditions matches the design conditions as shown in Figures 6.22 and 6.23. This happens because the capacity of the heat pump is directly proportional to the water mass flow rate through it. However at points where the demands exceed capacity, the heat pump fails to meet the demand.

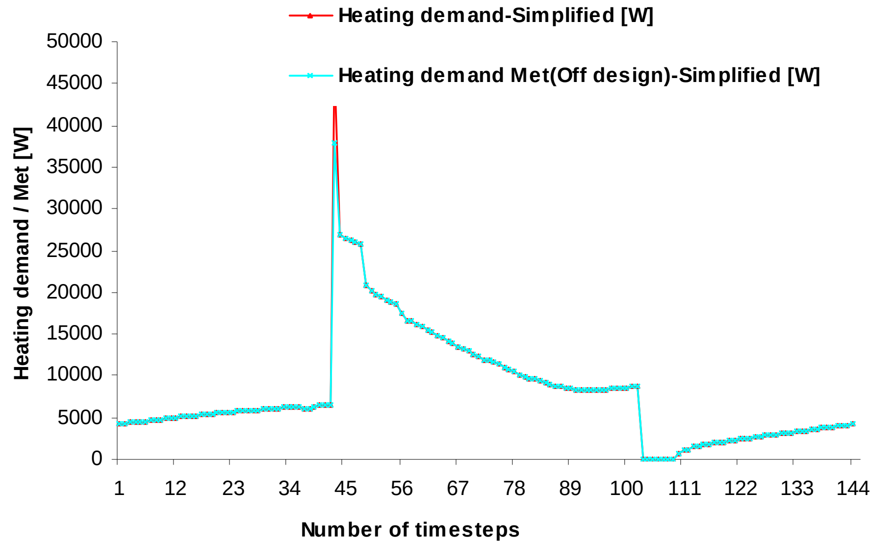

Figure 6.22. *Heating mode under off design conditions*

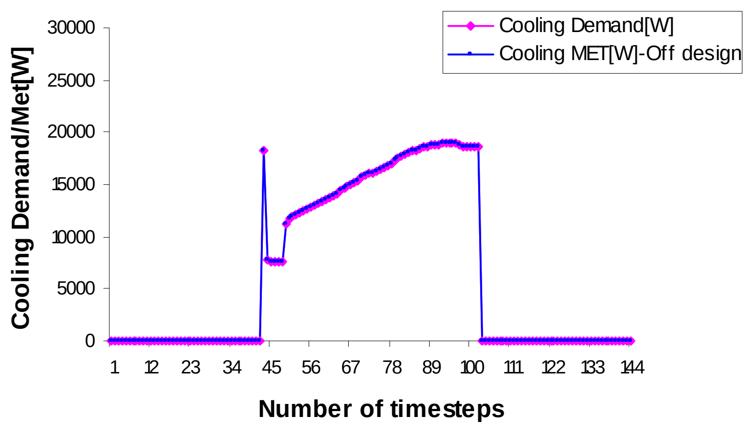

Figure 6.23. *Cooling mode under off design conditions*

As shown in section 5.4, the model's sensitivity to wet bulb temperature is negligibly small for the case when the C1 term in equation 4.1 has been fit for constant wet bulb and variable flow rate (i.e. ClimateMaster data). Based on this analysis, off-design wet bulb conditions are not shown in this section.

6.2.9. Annual Simulation Computation time

For the case study input file, the parameter estimation model running in EnergyPlus takes approximately 35 minutes computing time for an annual simulation on a PC with Pentium 4, 2.4GHz CPU. The simple model takes approximately 7 minutes under identical conditions, which is about 80% faster than the parameter estimation model.

CHAPTER 7

PERFORMANCE COEFFICIENTS CALCULATOR

7.1. Need for an interface

Computation of the model performance coefficients is an important prerequisite to executing accurate simulations. Although a generalized least square method will serve the purpose, calculation of every coefficient for different heat pump units becomes tedious without a graphical user interface. An interface was developed that not only calculates the performance coefficients, but also compares every calculated data point with the corresponding catalog data point in a graphical environment to demonstrate the consistency of the model coefficients with the catalog data.

7.2. Front end architecture

The front end is a visual basic graphical user interface (GUI), the backend application is the generalized least square method. The program interacts with the user through a combination of menu driven event handling sub-functions. Event driven programming determines the sequence of operations for an application by the user's interaction with the application interface (forms, menus, buttons, graphical components etc). Here, the user picks up the process to be performed and the event driven application remains in the background. The advantage of this design is that the application logic that processes events is clearly separated from the user interface logic that generates these events.

7.2.1. Workspace

The key elements in designing an interface are deciding what the user sees, what data he will enter, what kind of warning boxes the application will use and how the application will handle inputs and outputs. Figure 7.1 shows the application window. The application begins by automatically asking the user to select the mode in which the heat pump is operating. Once the user clicks on the command button to select the mode, the application window interacts with the user again prompting him to click the type of performance variable coefficients to be generated. In the cooling mode, the user may click on the Cooling capacity, EER or Sensible capacity coefficients as shown in Figure 7.2. In the heating mode, the user selects heating capacity or COP coefficients. Once the user makes his selection, the application automatically loads the interface window. The interface window consists of a comprehensive menu that supports and conducts all important operations. Figure 7.3 shows the interface window. Operations such as opening a new file, clearing all the values on the interface window, saving the file and aborting the application can be performed under the FILE menu. The window accepts input parameters for the model equations and reports the values of the coefficients after back end processing. The window also calculates the model outputs and compares them with respective catalog data points to demonstrate the correctness of the coefficients.

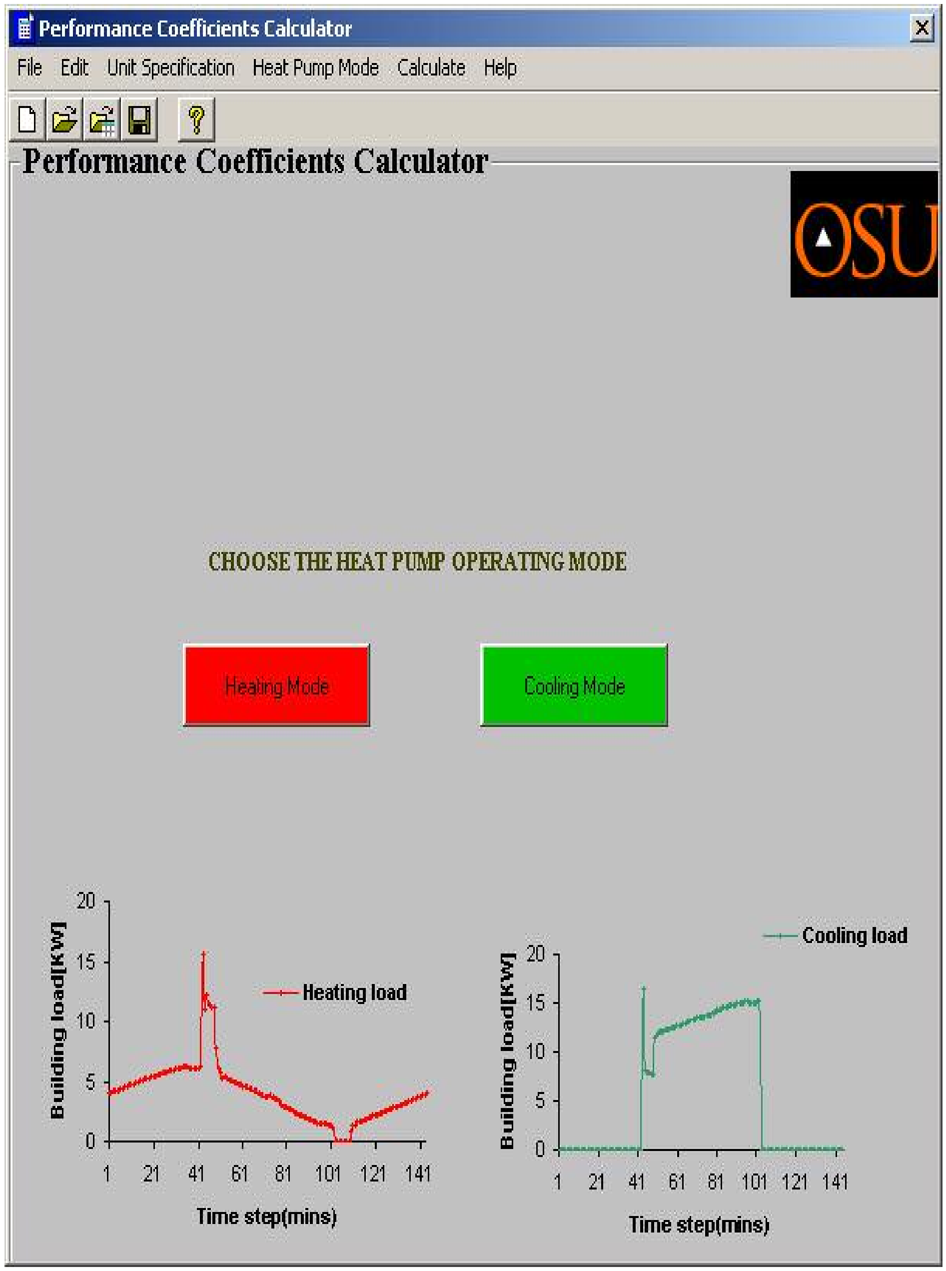

Figure 7.1. *Application window of the performance coefficient calculator*

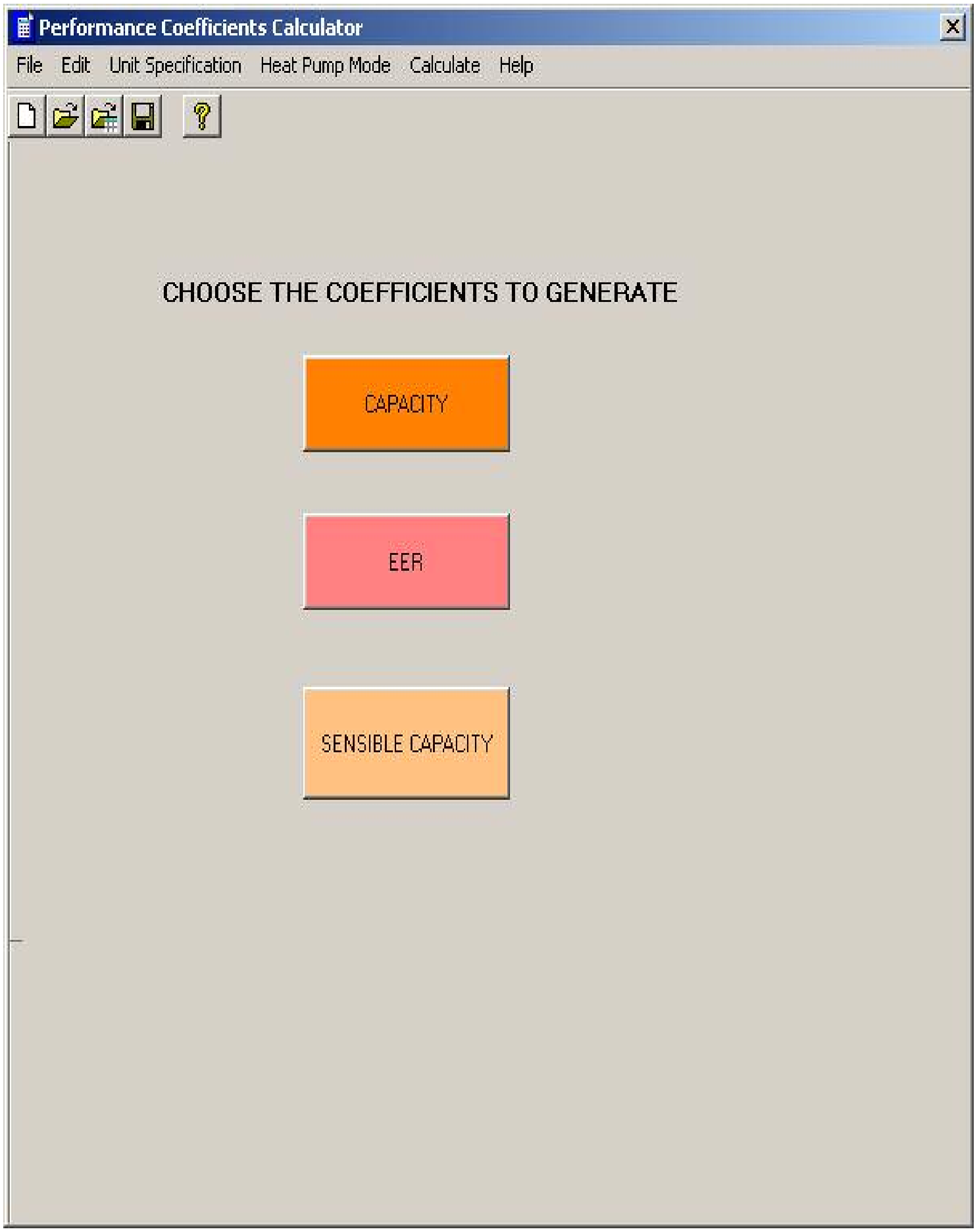

Figure 7.2. *Window when the user clicks on cooling mode*

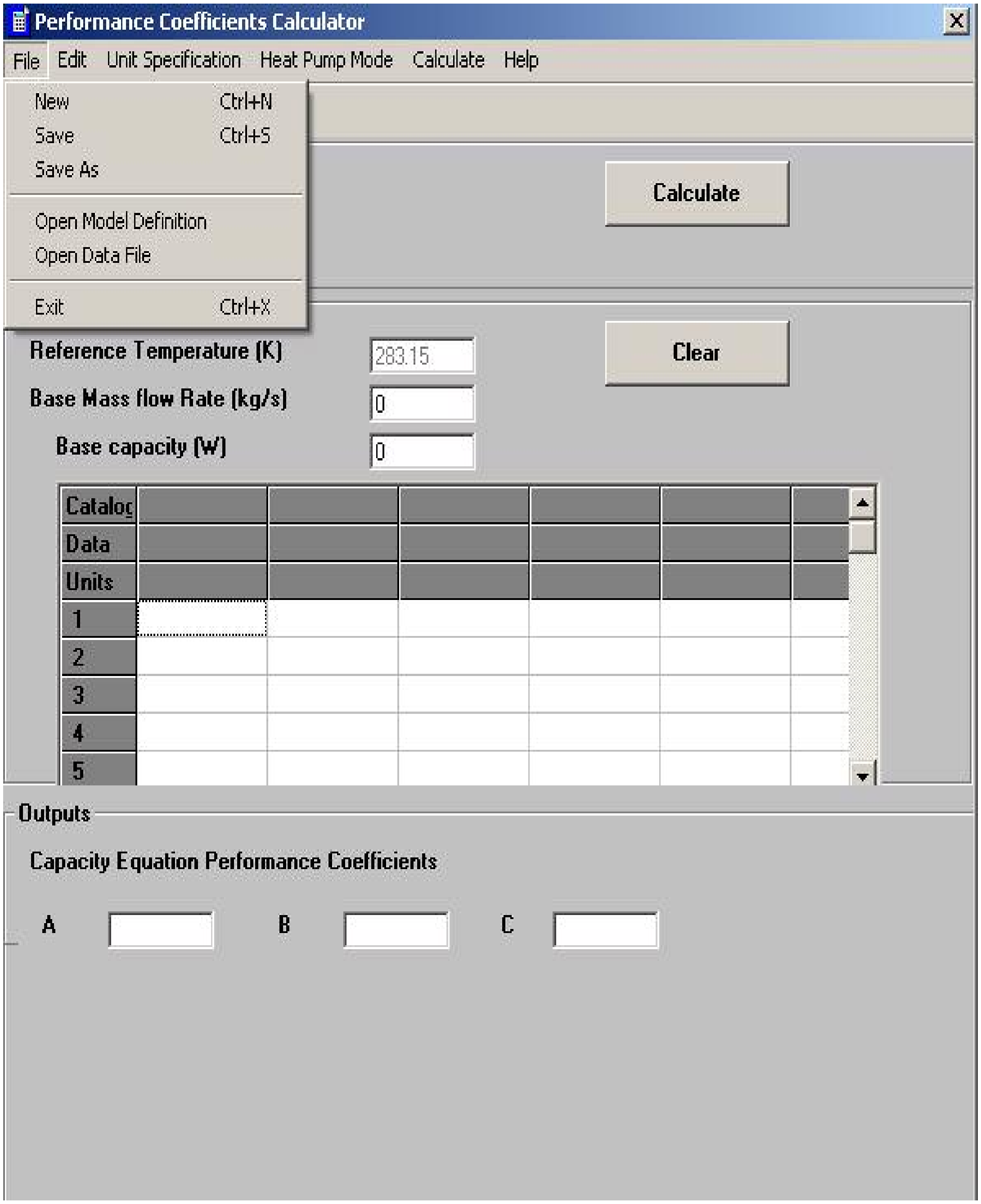

Figure 7.3. *The main interface window*

7.2.2. Input format

For each unit at the specified operating conditions, the following catalog and input data is needed

- Total cooling capacity (Cooling Mode)
- Sensible cooling capacity (Cooling Mode)
- Heating capacity (Heating mode)
- Energy efficiency ratio(EER)
- Coefficient of performance(COP)
- Nominal capacity of the heat pump unit
- Nominal water mass flow rate
- Inlet water mass flow rate
- Entering air temperature
- Entering water temperature
- Nominal coefficient of performance of the heat pump unit
- Nominal energy efficiency ratio of the heat pump unit

The nominal values, which are determined from the manufacturer's catalog are passed to the back end processor by using textboxes. Since the catalog data is the heat pump performance data at multiple points, it would be tedious to have the user enter every single operating point into the text box. Hence a grid component has been introduced as shown in Figure 7.3. The grid component gets information from a text file with the extension ".DAT". The text file contains all the catalog input parameters with their values at the corresponding operating points as shown in Figure 7.4.

```
/* Model Name */
WATERTOAIR COOLING MODE;
/* Data Reference - model number */
FloridaHeatPumpGT018;
/* No. of data points per data set */
25;

/* Input values */
 /*  Entering Air temp, Water mass flow rate, Entering Water temp, Total Capacity*/
/* K     kg/S    K      W - */
283      0.25    289.1  5400;
283      0.25    290.8  5600;
283      0.25    292.4  5900;
283      0.25    294.1  6100;
283      0.25    295.8  6400;
288.6    0.25    289.1  5100;
288.6    0.25    290.8  5300;
288.6    0.25    292.4  5600;
288.6    0.25    294.1  5800;
288.6    0.25    295.8  6100;
294.1    0.25    289.1  4800;
294.1    0.25    290.8  5100;
294.1    0.25    292.4  5300;
294.1    0.25    294.1  5500;
294.1    0.25    295.8  5800;
302.4    0.25    289.1  4300;
302.4    0.25    290.8  4500;
302.4    0.25    292.4  4700;
302.4    0.25    294.1  4900;
302.4    0.25    295.8  5200;
310.8    0.25    289.1  3900;
310.8    0.25    290.8  4100;
310.8    0.25    292.4  4300;
310.8    0.25    294.1  4500;
310.8    0.25    295.8  4700;
```

Figure 7.4. *A typical model input file*

The grid component is also associated with a model definition file that understands the format of the input that is being fed into the grid component as shown in Figure 7.5. In other words, there is a correspondence between the model definition file and the input file. The model definition file has a ".pd" extension. The model definition file has to be loaded into the component before feeding the input information so that the grid component assimilates the type of information it needs to load.

```
/* Model Name */
Watertoair Coolingmode;

/* No. of Inputs */
4;
/* No. of Outputs */
3;

/* Input variable names */
Entering Water Temperature,
Entering Water Mass flow rate,
Entering Air Temperature,
Total Capacity;

/* Input variable units */
K,
Kg/s,
K,
W;
```

Figure 7.5. *A typical model definition file*

Once the definition and the inputs are passed into the component, the inputs are displayed on the interface window. Figure 7.6 shows an example of a pre processed interface window when Florida heat pump GT018 is loaded into it.

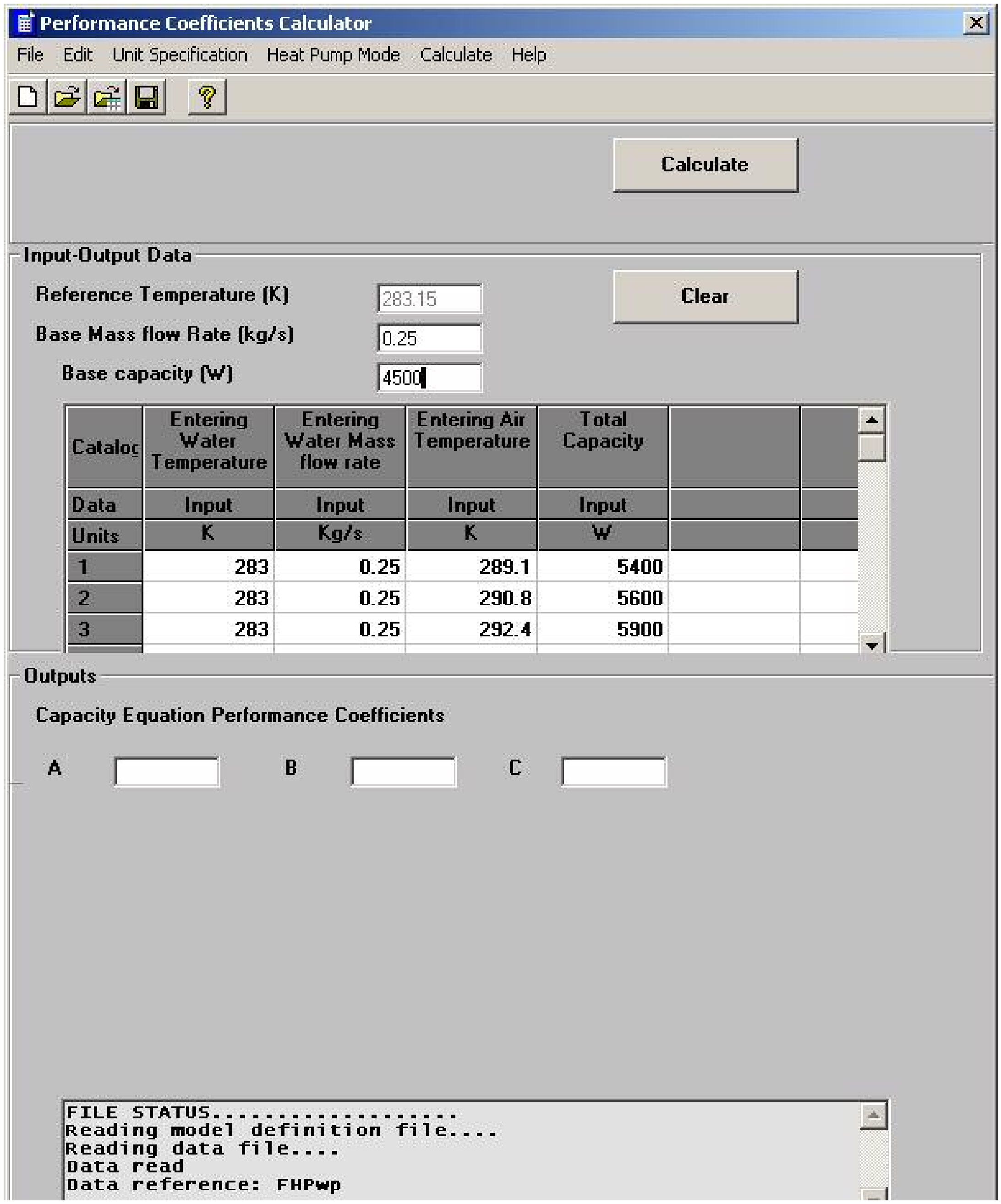

Figure 7.6. *Preprocessed interface window*

7.2.3. Reporting results

After the coefficients are generated by the back end processor, which is the generalized least square algorithm, a validation technique is required to determine if the generated results are reasonable. The interface plots point to point values of the performance variables and then compares every calculated (Model) data point with the catalog data point. This feature triggers automatically as the user clicks on the CALCULATE command button. This event informs the back end processor to calculate the coefficients, use these coefficients in the model equations to generate the respective performance variables and finally plot them in the form of bar charts for point to point comparisons. A typical output generated window is shown in Figure 7.7. A text area is also placed at the bottom indicating the successful reading of the input files.

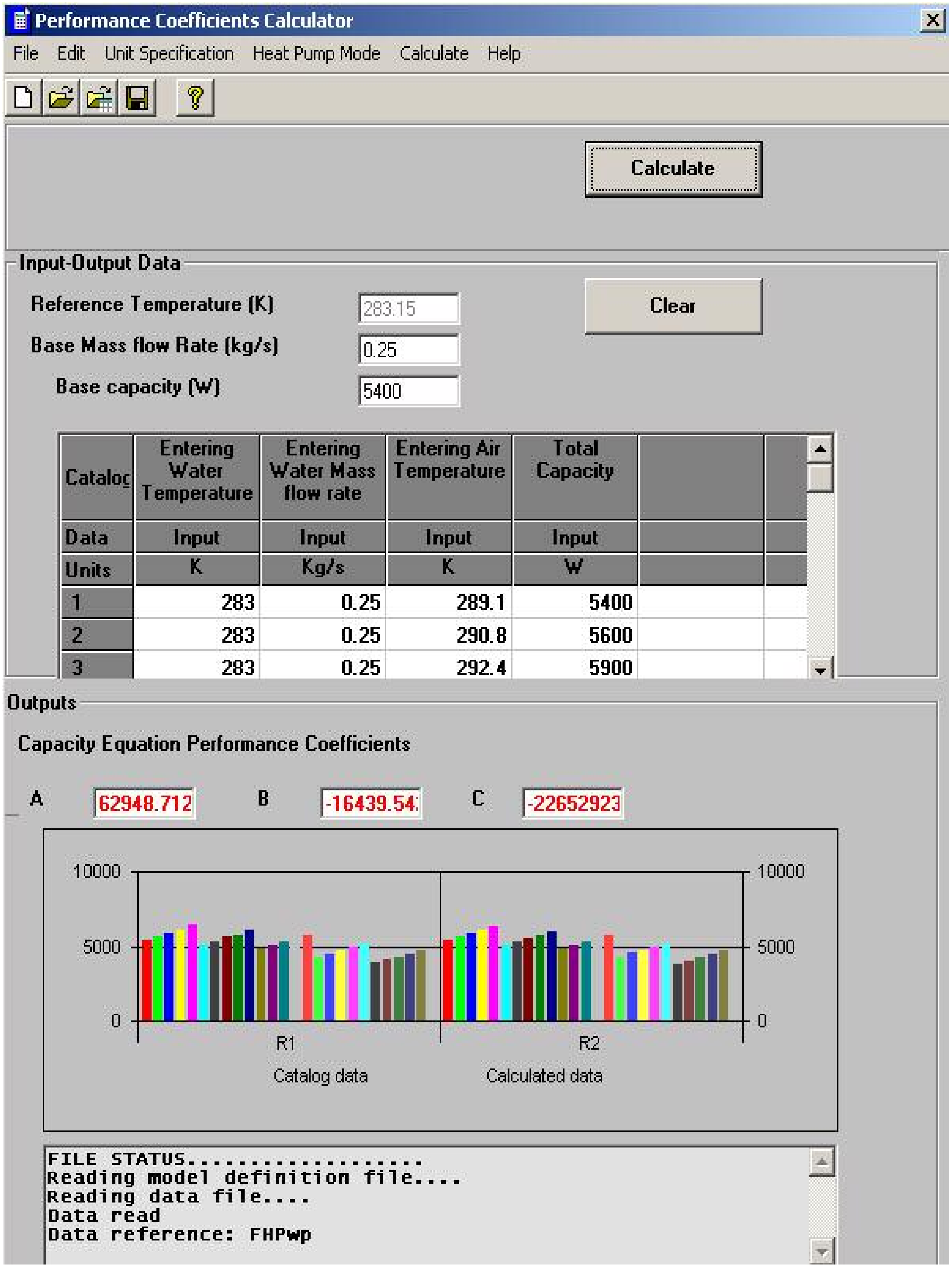

Figure 7.7. *Output window with the coefficients and comparison plots*

7.3. Back end architecture

Performance coefficients in the heating and cooling modes for the equation fit model are determined by implementing a generalized least square equation fitting method. The method uses a minimal sum of the deviations squared from a given set of data. The performance coefficients A1-J1 and A2-F2 in equations (4-1) (4-2) (4-3) (4-7) and (4-8) are generated by this method. The interface provides the facility to fit variable data points. Since the manufacturer's catalog easily contains at least 25 data points and a curve fit needs to be implemented among all these points, the data has to be dealt with in a form of a well defined matrix. Also this matrix may be reduced mathematically by using various numerical methods like the Gauss elimination method or LU decomposition method shown in Appendix A.

CHAPTER 8

CONCLUSIONS AND RECOMMENDATIONS

8.1. Conclusions

This thesis presents the modeling, implementation and verification of a simplified model for water-to-air ground source heat pump system. The existing water source heat pump system model originally developed by Lash (1990) is extended to obtain splits in the latent and sensible cooling capacity of the heat pump. An extra performance coefficient coupled with the temperature and mass flow rate ratios serve the above purpose. In addition, an extension to support variable air flow rate is proposed. This extension is currently supported only through the use of correction factors which are available from some manufacturers.

Within the range of manufacturers' data, the accuracy of the simplified model is equal to that of the detailed model. The accuracy of model coefficients generated from two different heat pump manufacturers' catalog data was evaluated. This evaluation showed that for all cases within the range of the catalog data, the error in the predicted results was less than 1.5%. Performance of the model outside the range of catalog data was also evaluated. Although the results predicted outside of the data set by the detailed model (Jin 2002) were slightly better in all categories as shown in Table 5.6, the simplified model predicted the power, total capacity, and the latent sensible split with RMS errors in the 6-8% range. This is quite reasonable for most applications.

The model is implemented in EnergyPlus as a single component model which can switch between heating and cooling modes. The unitary water-to-air heat pump component is treated as a virtual component which reacts to a thermostatic signal to allow heating or cooling. The simplified model does not carry the high overhead cost associated with accessing the refrigerant data required by the detailed model. This makes the process computationally less intensive and greatly reduces the cost of implementing the model. For an annual simulation in EnergyPlus, the simplified model is approximately five times faster than the detailed model. Implementing the detailed model not only requires a complete library of refrigerant properties but also requires error correction code to keep the property routines 'in bounds' while the model attempts to converge on a solution. Thus, even if refrigerant properties are already available in a simulation environment, the cost of implementing the detailed model is at least twice the cost of implementing the simplified model.

A case study was performed to verify the implementation of the heat pump model in EnergyPlus. The case study was carried on a typical office building assumed to be located at Chanute AFB (Rantoul), Illinois. An annual building loads simulation along with simulations for the summer design day (21^{st} July) and the winter design day (21^{st} January) were run for this region. Results obtained from the simplified model were compared with the detailed model under similar conditions. The following conclusions are based on the case study results.

- The models show the expected trends of the thermal processes correctly. This can be proved by the matching design day profiles and the model demand met profiles.
- The model responds correctly to changing water loop temperatures. With a ground loop configuration coupled with a water source heat pump, the temperature drops during winter making the heat exchanger, a heat source and increases during the summer making the heat exchanger a heat sink.
- The simplified model is in good agreement with the detailed model for the design day simulations. The maximum error obtained from the profile was less than 1%.

Finally, an interface was developed to calculate the performance coefficients. The interface also compares every calculated data point with the respective catalog data point in a graphical environment to show the correctness of the computed coefficients.

8.2. Recommendations

1. Run controlled experiments to obtain a complete set of heat pump operating data including variable air flow rates, water flow rates, water temperatures, air dry bulb temperatures and air wet bulb temperatures. Use this data to validate the extended version of the model (equations 4-15 through 4-19)
2. Implement the extended version of the model in EnergyPlus.
3. Validate the model results against a real working building and system instead of a typical building configuration. This study would involve validation of the entire EnergyPlus simulation with experimental study conducted on an existing building.
4. Investigate the possibility of using a single set of heat pump coefficients for both heating and cooling. This may involve changing the form of the model equations by introducing non-dimensional correction factors. These correction factors may then be a characteristic feature of the individual heat pumps.
5. Run controlled experiments to obtain a complete set of heat pump operating data including variable air flow rates, water flow rates, water temperatures, air dry bulb temperatures and air wet bulb temperatures. Use this data to validate the extended version of the model (equations 4-15 through 4-19)
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Appendix A

Mathematical description with example

Gauss elimination has the disadvantage that all right hand side vectors must be known in advance for the elimination step to initiate. The LU decomposition method can counter this disadvantage and aid in performing modifications to the matrix independent of the right hand side vector. This method breaks down a matrix into two separate matrices. The solution X to the linear system AX=B is found in 4 steps

- Construct the matrices L and U such that A=LU
- Solve LY=B for Y using forward substitution
- Solve UX=Y for X using backward substitution
- Obtain the solution for the coefficients of the equations.

Example

Assume 3 sets of catalog data. It is required to fit 3 coefficients A1, B1, C1 in the model equations formed using the catalog data. For simplicity, also assume that the three equations resulting from the catalog data are as follows

$$6A1-2B1=14$$

$$9A1-B1+C1=21$$

$$3A1-7B1+5C1=9$$

The solution for A1, B1, and C1 by manual calculations is shown below

Upper Triangular Matrix U	Step Explanation	Lower triangular matrix L
$\begin{pmatrix} 6 & -2 & 0 \\ 9 & -1 & 1 \\ 3 & -7 & 5 \end{pmatrix}$	Initial Matrix U Matrix storing elementary row operations L	$\begin{pmatrix} 0 & 0 & 0 \\ 0 & 0 & 0 \\ 0 & 0 & 0 \end{pmatrix}$
$\begin{pmatrix} 1 & -1/3 & 0 \\ 9 & -1 & 1 \\ 3 & -7 & 5 \end{pmatrix}$	Multiply position (1,1) in U by 1/6	$\begin{pmatrix} 6 & 0 & 0 \\ 0 & 0 & 0 \\ 0 & 0 & 0 \end{pmatrix}$
$\begin{pmatrix} 1 & -1/3 & 0 \\ 0 & 2 & 1 \\ 0 & 8 & 5 \end{pmatrix}$	Introducing zeros to positions (2,1) and (3,1) require multiplications by -9 and -3 respectively. So we will store the opposite of these numbers in their respective locations.	$\begin{pmatrix} 6 & 0 & 0 \\ 9 & 0 & 0 \\ 3 & 0 & 0 \end{pmatrix}$
$\begin{pmatrix} 1 & -1/3 & 0 \\ 0 & 1 & 1 \\ 0 & 8 & 5 \end{pmatrix}$	On to the next position in the main diagonal, (2,2). To replace the value in this position with a 1, multiply row 2 by 1/2, thus storing a 2 (the reciprocal) in position (2,2) in the lower triangular matrix.	$\begin{pmatrix} 6 & 0 & 0 \\ 9 & 2 & 0 \\ 3 & 0 & 0 \end{pmatrix}$
$\begin{pmatrix} 1 & -1/3 & 0 \\ 0 & 1 & 1/2 \\ 0 & 0 & 1 \end{pmatrix}$	Replacing the position under the leading 1, position (3,2), with a zero can be done with a multiplication of -8. We will then store 8, the opposite of -8, in the lower matrix at that position.	$\begin{pmatrix} 6 & 0 & 0 \\ 9 & 2 & 0 \\ 3 & -8 & 0 \end{pmatrix}$
$\begin{pmatrix} 1 & -1/3 & 0 \\ 0 & 1 & 1/2 \\ 0 & 0 & 1 \end{pmatrix}$	Only a multiplication of 1 is necessary to introduce a 1 to the next diagonal position. In fact nothing is being done to the upper triangular matrix, but we need the 1 in the lower matrix to demonstrate that.	$\begin{pmatrix} 6 & 0 & 0 \\ 9 & 2 & 0 \\ 3 & -8 & 1 \end{pmatrix}$

Hence, the first step is achieved where a single matrix A is split into 2 separate matrices that is A =LU or in this case

Step 1:-

$$\begin{pmatrix} 6 & -2 & 0 \\ 9 & -1 & 1 \\ 3 & -7 & 5 \end{pmatrix} = \begin{pmatrix} 6 & 0 & 0 \\ 9 & 2 & 0 \\ 3 & -8 & 1 \end{pmatrix} \begin{pmatrix} 1 & -1/3 & 0 \\ 0 & 1 & 1/2 \\ 0 & 0 & 1 \end{pmatrix}$$

Step 2:-

Solving the system of equations AX=B. Making the system LU**x** = **b**.

$$X = \begin{pmatrix} A1 \\ B1 \\ C1 \end{pmatrix}$$

$$\begin{pmatrix} 6 & 0 & 0 \\ 9 & 2 & 0 \\ 3 & -8 & 1 \end{pmatrix} \begin{pmatrix} 1 & -1/3 & 0 \\ 0 & 1 & 1/2 \\ 0 & 0 & 1 \end{pmatrix} \begin{pmatrix} A1 \\ B1 \\ C1 \end{pmatrix} = \begin{pmatrix} 14 \\ 21 \\ 9 \end{pmatrix}$$

Step 3:-

Define a new column matrix **y** so that U**x** = **y**.

$$\begin{pmatrix} 1 & -1/3 & 0 \\ 0 & 1 & 1/2 \\ 0 & 0 & 1 \end{pmatrix} \begin{pmatrix} A1 \\ B1 \\ C1 \end{pmatrix} = \begin{pmatrix} Y1 \\ Y2 \\ Y3 \end{pmatrix}$$

Step 4:-

Rewrite step one with the substitution from step two yielding L**y** = **b**.

$$\begin{pmatrix} 6 & 0 & 0 \\ 9 & 2 & 0 \\ 3 & -8 & 1 \end{pmatrix} \begin{pmatrix} Y1 \\ Y2 \\ Y3 \end{pmatrix} = \begin{pmatrix} 14 \\ 21 \\ 9 \end{pmatrix}$$

Solve for **y** using forward substitution

$Y1 = \frac{7}{3}$ $\qquad Y2 = \frac{29}{6}$ $\qquad Y3 = \frac{33}{2}$

Step 5:-

Using the results from step 4, solve for **X** using back substitution in step 2

$A1 = \frac{43}{36}$ $\qquad B1 = \frac{-41}{12}$ $\qquad C1 = \frac{33}{2}$

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