MODELING, VERIFICATION AND OPTIMIZATION

OF HYBRID GROUND SOURCE HEAT PUMP

SYSTEMS IN ENERGYPLUS

By

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# CHAPTER 1

## INTRODUCTION

### *1.1 Background*

Ground source systems in which supplemental heat rejecters are used together with the ground loop are called hybrid ground source heat pump (HGSHP) systems. The need for the HGSHP system arises in conditions where the heat rejected to the ground over the year is more than the heat extracted from ground during the same period. Such a condition will result in the elevation of ground temperatures over a period of a few years. The elevated ground temperatures result in increased fluid temperatures, which affect the heat pump performance. In such cases, sizing the ground loop to meet the cooling load will yield unreasonably long loop lengths. The first cost for installation becomes high as the loop length increases, thereby reducing the competitiveness of ground source heat pumps compared to conventional HVAC systems.

In order to design for the same capacity with a smaller ground loop length, a HGSHP system, with a supplemental heat rejecter like such as a cooling tower, cooling pond or fluid coolers etc can be used to reject any excess heat. These types of systems reduce the cooling load that has to be met by the ground loop, thereby reducing the size and

first cost of the ground heat exchanger. An ideal ground loop system will have the

heating and cooling loads on the ground loop balanced over a year. The major basis of

design for any HGSHP system is thus to have balanced operation of the ground loop. The

excess cooling load is met by supplemental heat rejecters. The design procedures for the

hybrid systems are discussed in more detail in chapter 4. A typical configuration of a

HGSHP system would be as shown in figure 1-1. Other configurations possible are

discussed in chapter 4.
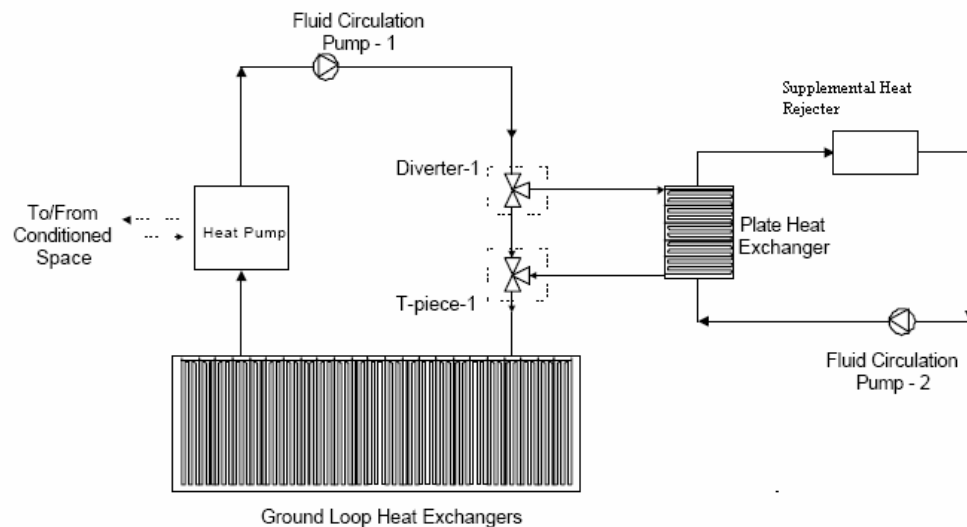


**Figure 1-1.** *Typical HGSHP Component Configuration (Yavuzturk, 2000)*

Although the HGSHP configuration reduces the first cost, the supplemental heat

rejecters typically require increased maintenance. The supplemental heat rejecters are

generally placed in a separate circulation loop as shown in figure 1-1. The reasons for

such a configuration are as follows:

- Placing the tower in a separate loop facilitates regular maintenance of the tower
  without interfering in the ground loop operation.

- Such a configuration will bypass the tower loop whenever required by the control system.

- Vertical heat exchangers with an open loop are not at all encouraged due to the high cost of pumping.

As a result, the supplemental heat rejecters require use of a separate circulating pump. A poorly designed system might consume more electricity in the secondary loop and require excessive maintenance. In such poorly designed systems, the savings in capital from reducing the size of the ground loop may be more than offset by increased maintenance and operating costs.

HGSHP system design and optimization can be viewed as a three-phase process as shown in figure 1-2. There are five critical considerations in a HGSHP design, which are enumerated below:

i.   The ground loop is the major component of a ground source system. The sizing of the ground loop affects the cost and the performance of the system. Design of a vertical includes selection of the grout, the borehole spacing, the borehole depth and borehole configuration. While determining the depth of the borehole is the important factor, the other parameters also affect the ground loop performance.

ii.  Once the ground loop is designed, the designer has to select and size the supplemental heat rejecter. The selection of the type of supplemental heat rejecter is based on the environmental conditions. For example, for systems operating near a water body, a pond loop will be a good choice. The sizing of the supplemental heat rejecter is very important to achieve desired system performance. The sizing is done based on the load not met by the ground loop.

iii. The system mass flow rate presents another important parameter in system design. Pressure drop calculations and consequently the circulating pump size is dependent on the mass flow rate.

iv. Control strategies need to be developed for system operation. It is important that the system perform efficiently under all circumstances. In order to do this the system must be controlled in such a way that achieves maximum heat transfer whenever possible.

v. There are different ways in which the HGSHP components can be configured. A designer has to decide the best configuration for the system under consideration. Each configuration behaves differently, and it is therefore important to give due consideration to the configuration.

A HGSHP system contains a ground loop heat exchanger whose characteristic time constant is very large. Thus, any heat rejection/extraction by the ground loop will have an effect on the system for a longer time. A system simulation can investigate the system performance over a long period and thus provide the designer with a better understanding of the system. Using the simulation the designer can predict quantities like the energy consumption, flow rates, pressure drops etc. Another advantage of the simulation is that the effect of any changes in the system can be studied without physically having to implement the change. Thus, simulation paves the way for a better design, which will lead us to an optimal system.
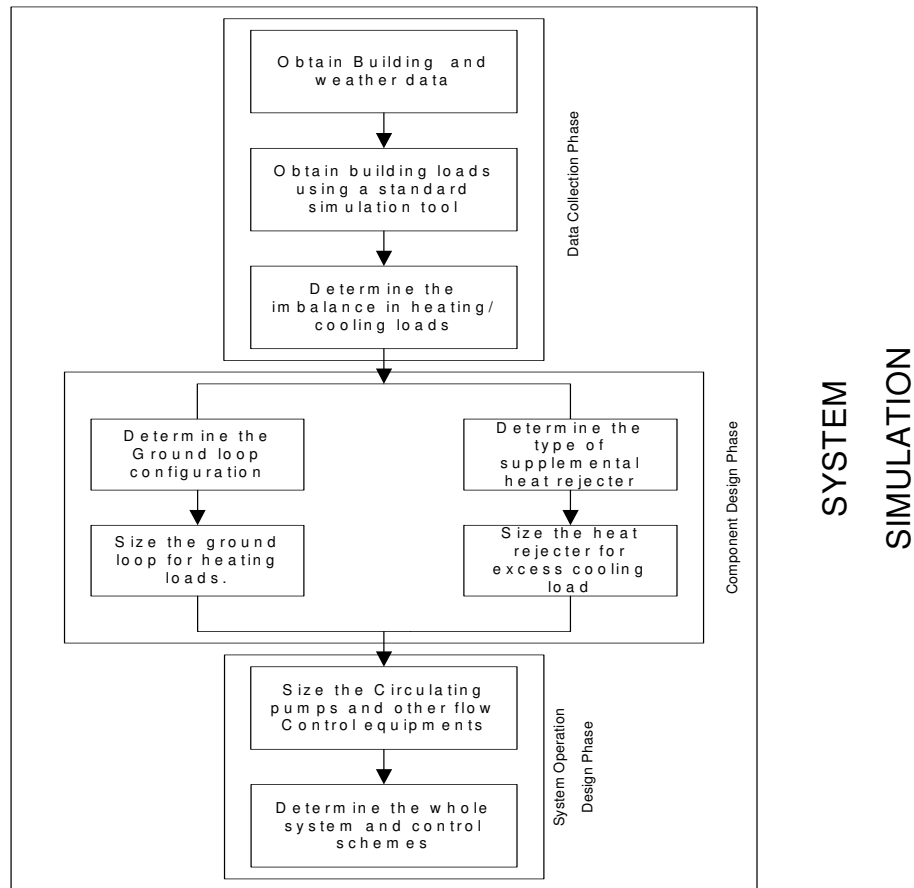
**Figure 1-2.** *Steps in an HGSHP Design and Analysis*

In order to optimize the design a designer can intelligently vary the parameters and analyze how the performance varies. A parametric study can be a good option if the range of the design is small or there are only a few design variables. In the case of HGSHP system, the number of design variables and the simulation time required makes the use of parametric study as an optimization exercise prohibitively expensive. It is thus necessary to use an optimization shell, which covers the whole range of design variables by a mathematical algorithm rather than a parametric study. The optimization procedure should answer the following questions:

- Is there the possibility of reduction in the ground loop length without generating unmet loads?

- Is the tower able to meet the excess cooling loads? For the cooling loads present, can we have a smaller tower?

- What are the optimal mass flow rates for the tower and ground loops?

- Does the increase in operating and maintenance costs of the system exceed the savings in the first cost?

- Is the control strategy used in the operation the best? What other alternative control strategies could be used?

## *1.2 Thesis Scope.*

The work presented in this thesis is aimed at simulating and optimizing HGSHP systems in the EnergyPlus simulation environment. To accomplish this, the program was modified as follows:

- Multi year simulation capabilities were developed and implemented. This included changes in weather processor to read a long weather file or else read a one-year weather file recursively.

- New supervisory control strategies, which can be used universally for, plant or condenser loops were developed. These strategies included Node set point based operation and Environmental temperature differential based operation.

- A plate heat exchanger model was implemented to allow configuration of heat rejecter loop.

The new models were verified by comparing simulation results with previous work by Yavuzturk (2000). Finally, GenOpt (Wetter, 1999) was used to optimize the system design parameters, component configuration and control strategies. Implementation of the optimization shell required the following additional changes to the EnergyPlus simulation:

- A daily time step model was developed and implemented to reduce computation time.

- Cooling tower and ground loop heat exchanger models were modified to operate on a daily timestep.

- The weather manager was modified to produce average daily values.

The optimization of the system was performed with the objective of reducing the life cycle cost of the system, energy consumption and balancing the loads. As a part of the optimization, the usefulness of each objective function was studied.

A review of previous work in HGSHP system design and simulation is presented in chapter 2. EnergyPlus models for different components are explained in detail in Chapter 3. In addition, control strategies present in EnergyPlus prior to this work are presented in detail. The chapter also presents the new component models, and control strategies required to enable the simulation of HGSHP systems in EnergyPlus. Verification of the simulation models is also presented in Chapter 4. As a case study, the new models are used to simulate the Oklahoma State University experimental system developed by Hern (2004). Simulation details and results are presented in chapter 4. Chapter 5 deals with the optimization problem related to HGSHP systems. The optimization is an initial attempt, which may lead to a broader optimization work in the

future. Chapter 6 presents the conclusions drawn from this work and recommendations

for possible future work.

# CHAPTER 2

## LITERATURE REVIEW

A review of the literature related to HGSHP system design and optimization is reported in the following paragraphs. Most of the work to date is related either to the design of HGSHP systems or to the performance of installed systems. Work done by previous researchers and published articles are summarized below.

The first design methodology for hybrid systems is discussed by ASHRAE (1995). The manual published by ASHRAE discusses the need for hybrid systems and their advantages. It takes into account the capital cost compared to ground source systems that utilize ground loop alone. The ASHRAE manual suggests a design procedure for the supplemental heat rejecter based on the difference between the average monthly heating and cooling loads, instead of the peak loads. The criterion for sizing the ground loops is identified as the annual heating loads. For a cooling dominated building, this results in excess cooling load, which is met by the supplemental heat rejecter. Guidelines are given regarding the method of connecting the supplemental heat rejecter to the central system. Use of a plate heat exchanger is recommended when an open cooling tower is used as the supplemental heat rejecter. This facilitates the bypass of either the cooling tower or the ground loop as required. The control strategy recommended for the supplemental heat

rejecter is set point control based on the heat pump entering water temperature. In cooling dominated buildings in southern climates, year round operation of the heat rejecter is suggested. For loops that are closely spaced the supplemental heat rejecter should be operated during night hours to restore heat to the ground.

Gilbreath (1996) conducted a performance study of a hybrid ground source heat pump system installed in a 75,000 ft$^2$ office center. The objective of the study was to suggest improved design methods for hybrid systems and to give recommendations for system monitoring. The work also investigated different flow control possibilities related to HGSHP systems. Flow control based on differential pressures across the pump and flow control based on ground loop temperatures was recommended by the author. The control variables (differential pressure or loop temperature) send signal to the circulating pump, which would be connected to a variable frequency drive. The HGSHP system was simulated for different cases with the heat pump part load ratio (PLR) serving as the control variable for the cooling tower. The tower was turned ON at heat pump PLRs of 30%, 50% and 75%. In all the three cases, the total energy used in cooling for a hybrid system was higher than a conventional system. The heating energy on the other hand increased for the case in which the tower was switched at 30% part load. For the 75% heat pump part load case, the heating energy decreased compared to the conventional system, due to the increased loop temperatures. A cost analysis, which deals the installation costs and operating cost, is given to compare the hybrid systems with conventional ground source systems. The benefits of the system were determined on the payback period of each system. The study showed that a conventional GSHP system for office center had a payback period of 14.2 years. An HGSHP system with a cooling

tower that switches ON at a heat pump PLR of 75% had a pay back period of 4.9 years, while systems with the cooling tower operating at 30 and 50% part load had a lower first cost due to shorter loop lengths and thus the payback period was immediate.

Kavanaugh and Rafferty (1997) addressed the cost related problems due to long ground loops. In places where the availability or cost of space is a constraint, the authors recommend use of hybrid systems with a fluid cooler or a cooling tower. Unlike the ASHRAE guideline, which sizes the heat rejecter based on the difference of monthly heating and cooling loads, Kavanaugh sizes the heat rejecter based on peak loads. The authors asserted that sizing based on the monthly heating loads would tend to oversize the ground loop. The authors also discussed the configuration of the supplemental heat rejecter, and supported the ASHRAE guideline of using a plate heat exchanger.

Kavanaugh (1998) recommends some improvement to the design procedures given by ASHRAE (1995) and Kavanaugh and Rafferty (1997). The improvements are suggested in order to address the heat build up in the ground, freezing in the loop, piping arrangements and controls. Though the ground loop is sized for the peak heating load in the Kavanaugh and Rafferty procedure it was not found to be a better design procedure. This is because the load profile in the building can result in a highly unbalanced load on the loop, thereby resulting in heat buildup in the ground. The new design method tries to balance the ground heat transfer on an annual basis. This helps to avoid the degraded heat pump performance due to heat build up. The balancing of the ground heat transfer is achieved by operating the supplemental heat rejecter on a schedule determined by a set point control strategy, based on the ground loop temperature. The number of operating hour for the supplemental heat rejecter was calculated so that the loads could be balanced

annually. The procedure is tested by analyzing a multi-storied building with a water-to-air heat pump system in three different climate regions. It was reported by the authors that tempering of the cold outside air by means of a heat recovery device was needed in cold climates to avoid freezing the ground loop. The airside heat recovery units reduced the building heating loads. For hybrid systems, the loop length was determined by the heating demand. Therefore, the minimum size of the ground loop increased from warmer to colder climate. A first cost savings of 66% was achieved for the warmest climate (Mobile, AL), while no savings were obtained for the coldest climate (Minneapolis, MN). For an intermediate climate (Louisville, KY), the first cost savings was calculated to be 19%. The operating costs were also calculated and the feasibility of installing an HGSHP system investigated. It was concluded that a HGSHP system is a good alternative for Mobile, AL, where the savings in first cost were far higher than the increased operating cost. For Louisville, KY, it was concluded that the increased operating cost of the HGSHP system would offset the savings in first cost.

Phetteplace and Sullivan (1998) conducted a case study of a hybrid ground source heat pump system installed at a 24,000 ft$^2$ (2,230 m$^2$) military base administration building in Fort Polk, LA. The system consisted of 70 vertical boreholes of 200 ft (61 m) depth placed at a spacing of 10 ft (3.3 m). The supplemental heat rejecter was a cooling tower with a capacity of 275 kW. The authors provided data collected from the building over a period of 22 months. Data analysis showed that the heat rejected to the ground was 43 times the heat extracted. The cooling tower was operated using a set point control strategy based on the heat pump exiting temperature. The cooling tower fan was activated when the heat pump exiting temperature reached 36$^0$C and was deactivated when the

temperature fell below $35^0$C. Hybrid system operation resulted in heat build up in the ground due to the high set point temperature for the cooling tower operation. Distribution of energy consumption was reported by the authors as 77% for the heat pump, 19% for circulating pumps, 3% for the cooling tower fan and 1% for the cooling tower pump.

A comparative study of different control strategies used in HGSHP systems was reported by Yavuzturk and Spitler (2000). The authors conducted a comparative study of three different control strategies based on a 14,000 ft$^2$ office building, located in Stillwater, OK. The building was simulated for two different climates, viz. Tulsa, OK (moderate climate) and Houston, TX (warm climate). A cooling tower was used as the supplemental heat rejecter. The basis of comparison was the life cycle cost of the system calculated over a 20-year period. The authors considered the following three control strategies:

- Set point control: Set point control strategy is the strategy recommended by ASHRAE (1995) and Kavanaugh et al. (1996). The strategy turns the supplemental heat rejecter on or off depending on the heat pump entering fluid temperature (EFT) or heat pump exiting fluid temperature (ExFT).

- Differential control: Differential control takes into account the difference between the heat pump EFT and the ambient wet bulb temperature. The cooling tower is turned on if the difference is above an upper set point and turned off when the difference is less than the lower set point.

- Schedule control: This strategy attempts to balance the ground heat transfer rate by operating the cooling tower on a fixed schedule each day. Generally, the cooling tower can be turned on to meet the load during the nighttime.

The authors simulated the building under consideration using the above strategies for both climates. The study concluded that the differential control strategy resulted in the most energy efficient hybrid system operation.

Ramamorthy et al. (2001) extended the work begun by Yavuzturk (2000) on the optimal design of HGSHP systems. The study used a shallow pond (Chiasson, 2000) as a supplemental heat rejecter. The authors used the differential control strategy and the building model developed by Yavuzturk et al. (2000). Initial savings in the ground loop cost for Houston (hot climate) was found to be as high as 50 – 65%. When the system was simulated with reduced length and a cooling pond as a supplemental heat rejecter, it was found that the heat pump energy consumption was reduced by around 37% from the base case with a GHE alone. For the Tulsa climate, the GHE length could not be reduced much due to the presence of heating loads. The heat pump energy savings for Tulsa was only 17% of the base case.

Work on the optimization of ground source heat pump systems was done by Khan (2004). Khan's work dealt with modeling antifreeze mixtures in GSHP systems. Khan (2004) studied of the effect of different anti-freeze mixtures on GSHP systems. He worked in the HVACSIM+ simulation environment. Optimization was carried out using GenOpt (Wetter, 2000) an optimization package developed at Lawrence Berkeley Lab. The author identified GLHE length, borehole radius, grout conductivity, antifreeze concentration and heat pump type as optimization variables. The optimization objective was selected as the life cycle cost. The major problem with the work was that the simulation was carried out for only one-year and the long-term ground heat build up (or ground cooling) was not taken into account. This could lead to erroneous results where

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the heating and cooling loads are highly unbalanced. Optimization of the GSHP systems yielded a 4% savings in life cycle cost. It was noted that the length of the borehole and anti freeze concentration were the significant parameters concerning the life cycle cost. Methanol was found out to be the best antifreeze compared to other antifreezes. Author also studied the performance of HGSHP systems with pavement heat rejecters as supplemental heat rejecter. The performance of pavement heat rejecter was compared to that of the cooling tower using the results obtained by Yavuzturk, 2000.

# CHAPTER 3

# MODELING OF HYBRID GROUND SOURCE HEAT PUMP

# SYSTEMS IN ENERGYPLUS

## *3.1 Overview of the Simulation Environment*

The work presented in this thesis is carried out in the EnergyPlus (Crawley et al.) simulation environment. EnergyPlus is an integrated whole building simulation tool developed by the US Department of Energy for analysis and design of buildings and related HVAC systems. By integrated, we mean that the building, system and plant simulations are solved simultaneously at each time step by a successive substitution scheme. The solution scheme successively solves for the state variables in each of the three fluid loops coupled as shown in figure 3-1.

The zone equipment/air loop consists of components directly connected to the zone (supply plenum, baseboard heaters, fan coils etc) and secondary HVAC equipment (cooling/heating coils, mixing boxes, humidifiers etc). This part of the simulation determines what type of energy each zone requires, the air mass flow required to maintain the zone set points and the equipment operating priority.

The plant loop consists of all pipes and primary equipments, circulating pumps etc. The loop is divided into two half loops viz. Plant demand side and Plant supply side to provide update points on the loop for the successive substitution solution scheme. The demand side is connected to the air loop at the coils, which are simulated in the zone/air loop. The demand side passes the coil loads to the plant equipment. All energy conversion equipments like chillers, boilers, heat pumps etc are simulated on the plant supply side. The demand side sets the flow rate for the loop depending on the coil loads while the final flow control is carried out by the circulating pump

The condenser loop in EnergyPlus is similar to the plant loop and is again divided into two half loops viz. Condenser demand and Condenser supply sides to accommodate the solution scheme. The condenser supply side consists of all the environmental heat exchangers including cooling towers, pond heat exchangers and ground heat exchangers. The condenser loop may be coupled to the plant loop at the chillers or water source heat pumps or coupled directly to the zone/air loop at the coils.

The work presented in this thesis deals with the plant and condenser loops. The existing component models that are used in HGSHP systems are briefly described. The existing controls are also described and the need for new control strategies detailed. New control strategies are added to the loops to better suit HGSHP applications. Required modifications to the simulation environment as well as several new models are also developed.
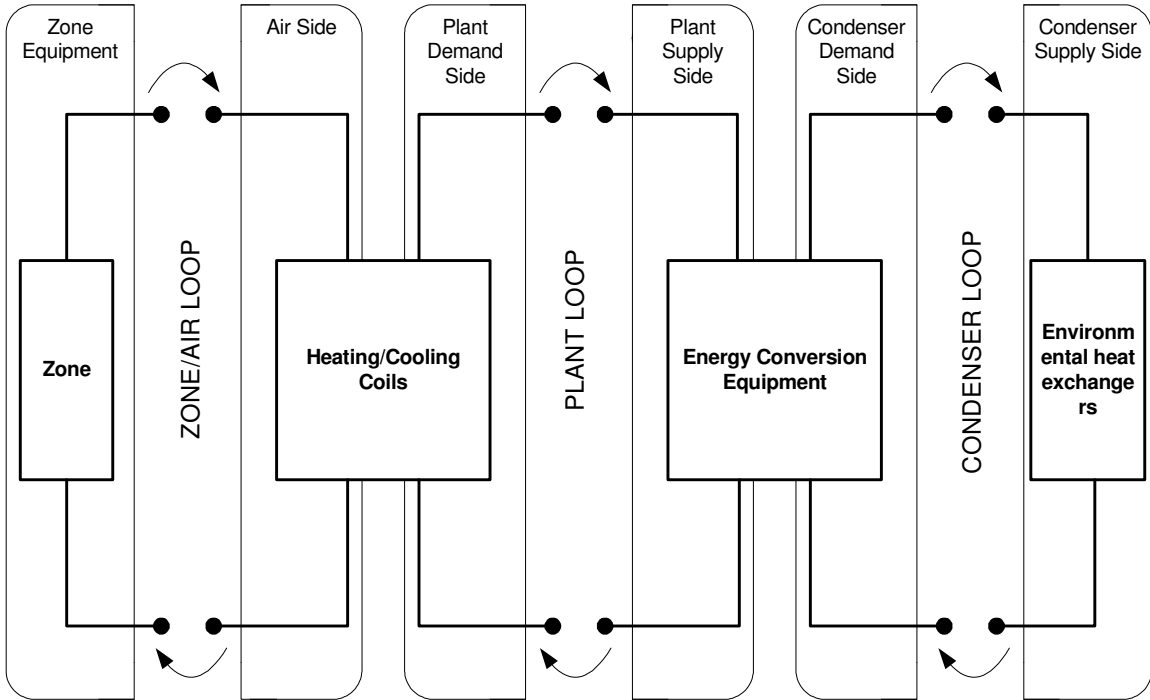
**Figure 3-1.** *Typical HVAC Loop Schematic in EnergyPlus*

## 3.2 EnergyPlus Component Models For HGSHP Simulation

This section briefly describes different EnergyPlus component models, which are central to HGSHP simulations. The component models described in this section will be located in the plant supply and condenser supply side loops, of the EnergyPlus structure. Since this study only considers the cooling tower as a supplemental heat rejecter, other supplemental heat rejecter models are not dealt with here.

### 3.2.1 Water Source Heat Pump Models

An integral component in any HGSHP application is the water source heat pump, which will act as the energy conversion equipment. Heat pumps with cooling towers, ground heat exchangers or pond heat exchangers are all examples of water source heat pumps. The load side can be either water (water-to-water heat pumps) or air (water-to-air

heat pumps). Water-to-water heat pumps are used for large commercial purposes, whereas water-to-air heat pumps are generally used for smaller applications.

EnergyPlus contains models for both water-to-air heat pumps and water-to-water heat pumps. Jin's (2002) water-to-air heat pump model and Shenoy's (2004) simplified water-to-air heat pump model were implemented in EnergyPlus. The water-to-water heat pump model available in EnergyPlus was also developed by Jin (2002). The model was implemented in EnergyPlus by Murugappan (2002).

Jin et al. (2002) presented the Parameter Estimation heat pump model that acts as a bridge between the less useful equation fit models and the highly complex deterministic models. Each heat pump component was modeled according to first principles as in deterministic models, but the problem of calculating the values for the parameters was tackled using a parameter estimation approach. The manufacturers catalog data was used to calculate the parameters so that the error is minimized using a global optimization algorithm. Jin developed the model for both water-to-air and water-to-water heat pumps. The water-to-air heat pump model takes into account the sensible – latent split, which is very important for airside heat transfer. The models were developed in such a way that they could be used with antifreeze solutions as the source side fluid too. The present work uses Jin's water-to-water heat pump model, which will be discussed later in this section.

Shenoy's (2004) water-to-air heat pump model is an extension of previous work done by Lash (1990). The model is an equation fit model and is computationally efficient compared to Jin's parameter estimation model. Equations for heat transfer and heat pump performance (EER/COP) are developed as functions of water inlet temperatures and water mass flow rates. The manufacturer's data is normally obtained with constant

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airflow, but this is not very helpful for all applications. In order to account for variable airflow rates correction factors were included to the manufacturer's data. The model consists of performance coefficients obtained from the manufacturers catalog data. The generalized least square method (GLSM) is used to obtain the performance coefficients. Shenoy (2004) also discusses the validation of the model with case studies involving two commercial water source heat pumps. The equations developed by Lash were modified so that the total heat transfer can be split into sensible and latent capacities.

As stated earlier the parameter estimation model is a combination of both deterministic and equation fit models. The method can be easily understood from the figure 3-2. The first step in developing the model is to obtain the heat pump's performance data and develop the governing equations. In a parameter estimation based water-to-water heat pump model, the heat pump components are modeled using basic thermodynamic relations. This helps to understand the processes better. However, the equations thus developed contain many parameters, which are not available in manufacturers catalog. In order to overcome this difficulty the parameter estimation procedure is used. The next step as shown in figure 3-2 is to use the catalog data to calculate the heat transfer and performance parameters for the basic model equations. The values thus obtained are compared with the manufacturer provided values and the error calculated. A multi variable optimization technique, generally a Nelder-Mead Simplex method is used to minimize the error. The parameters obtained for the minimum error models the heat pump in the most acceptable manner. These parameters are then used with the heat pump model in the system simulation.
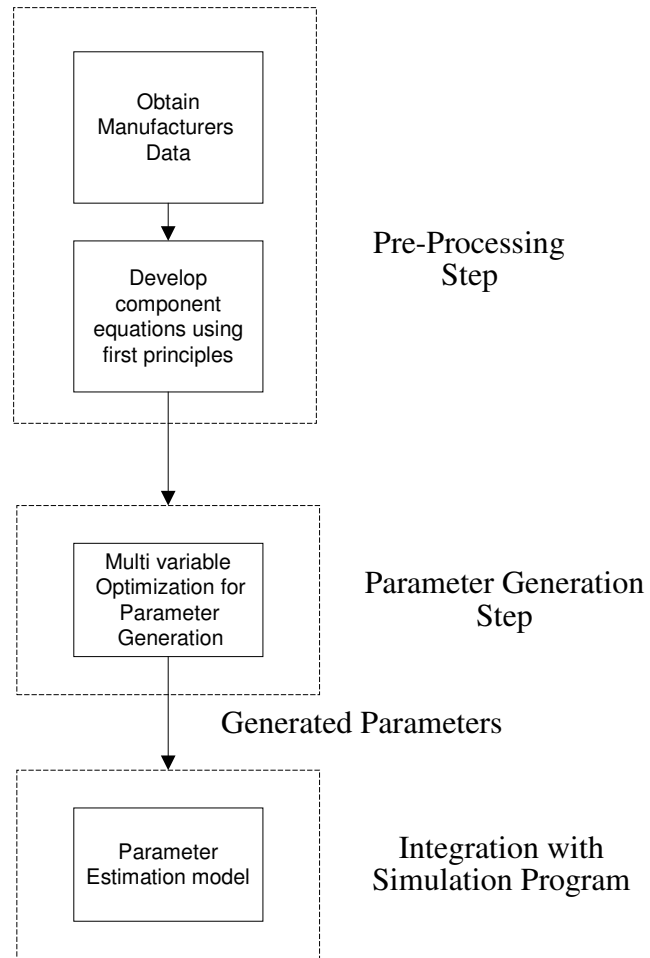
**Figure 3-2.** *Parameter Estimation Technique for Heat Pump*

Inputs, outputs and parameters for Jin's water-to-water heat pump model are shown in figure 3-3. The quantities labeled as parameters in the figure are the values, which are not available readily to a user. They are typically values, which are either physical measures or thermodynamic properties of the heat pump. Parameters like piston displacement (PD) and clearance factor (C) are physical measures of heat pump components while parameters like load/source side UA, maximum superheat etc are thermodynamic properties of the heat pump. While model inputs changes throughout the simulation due to interactions with other components, the parameters are component

specific and do not change during simulation or with system. As a result, a large library

can be developed with parameters for different heat pump models.



**Inputs**

$T_{wil}$  $m_{wil}$  $T_{wis}$  $m_{wis}$  $S$

**Heating Mode Parameters**

$(UA)_s$
$(UA)_L$
$PD$
$C$
$\Delta P$
$\Delta T_{sh}$
$W_{loss}$
$\eta$
$Phi$
$Plo$

Parameter Estimation Based Water to Water Heat Pump Model.

**Cooling Mode Parameters**

$(UA)_s$
$(UA)_L$
$PD$
$C$
$\Delta P$
$\Delta T_{sh}$
$W_{loss}$
$\eta$
$Phi$
$Plo$

$T_{wol}$  $m_{wol}$  $T_{wos}$  $m_{wos}$  $W$  $F$
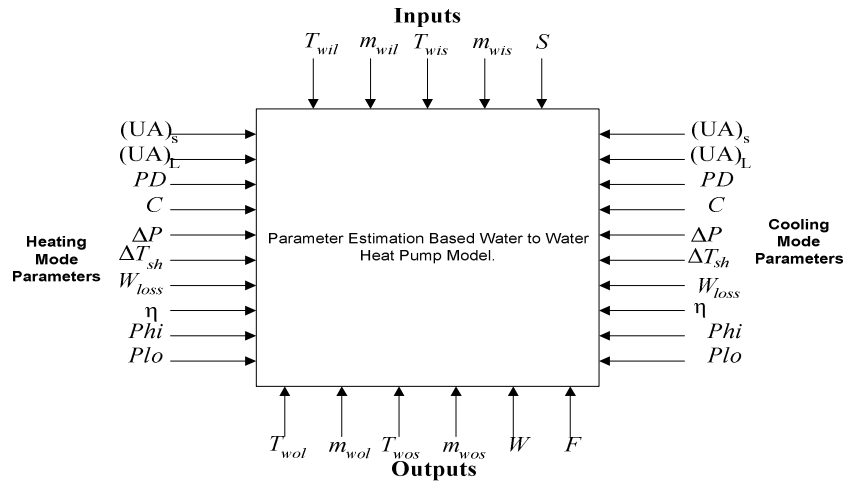
**Outputs**

**Figure 3-3.** *Block Diagram Showing Model Parameters, Input and Output (Jin, 2002)*

### 3.2.2 Vertical Ground Loop Heat Exchanger model

The ground source heat pump system as the name suggests uses a ground loop

heat exchanger (GLHE), which transfers heat to or from the ground. The EnergyPlus

GLHE model is a vertical borehole model based on the work done by Eskilson (1987).

Eskilson calculated the long-term g-functions for a number of borehole configurations.

This model's predictions are good if the analysis is done for a time step in the order of

months. Eskilson's long time step model is not helpful in its original form since

EnergyPlus and other system simulation programs can have time step, as short as one

minute. Short time step g-functions were developed to facilitate incorporation of ground

heat exchanger models in building simulation programs.

Yavuzturk and Spitler (1999) developed a new model for vertical ground loop

heat exchangers, which calculates short time step g-functions. Since EnergyPlus has a

variable time step, it requires a method to calculate g-functions at any time during

simulation. The model implemented in EnergyPlus is a load aggregation model, which takes into account the short time step g-functions. The flow chart for the GLHE model implemented in EnergyPlus is shown in figure 3-4. The important steps shown in the flow chart are explained below:

- Get the load on the ground loop for the simulation time step (hourly or sub hourly time steps). Also, get the GLHE parameters and g-functions obtained from a GLHE sizing program.

- Determine the present simulation time step. Interpolate between the g-function values to get the actual g-function for the present time step.

- Check if the current simulation time is less than the minimum sub-hourly time step. If YES, aggregate the load into the sub-hourly load array and update the outlet temperature and average fluid temperature using the sub-hourly update equations. If NO, then the simulation time has exceeded the sub-hourly stage and reached the hourly stage.

- Once the simulation time exceeds one hour, the sub-hourly loads are aggregated into hourly loads. The temperatures are updated using an hourly update equation.

- At each month (730 hrs), all hourly loads are aggregated into monthly loads. This aggregation improves the computational efficiency of the algorithm. New fluid temperatures are calculated using the monthly load aggregation equations.

- The process is repeated until the specified run period is complete. Since EnergyPlus uses variable time steps and the load for each time step varies, the method of load aggregation helps in reducing the simulation time considerably.
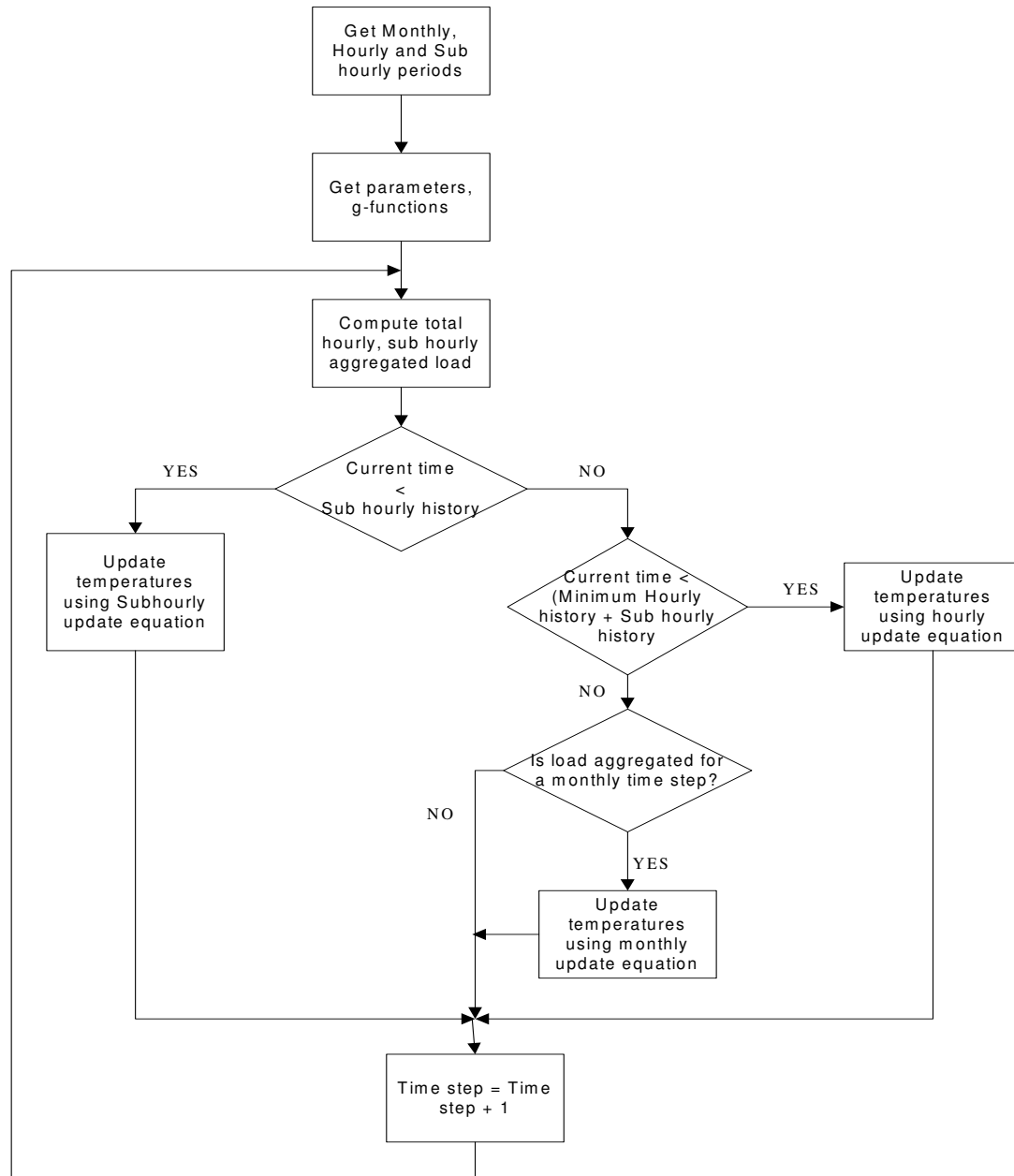
**Figure 3-4.** *Flow Chart for Short Time Step Ground Heat Exchanger Model*

### 3.2.3 Cooling Tower Model

EnergyPlus incorporates the cooling tower model developed by Bourdouxhe et al (1994),

which has its basis in the Merkel model (1925). Any mass transfer that may occur in the

cooling tower is ignored in this model. The cooling tower is modeled as a counter flow

heat exchanger using the Effectiveness – NTU approach. The model considers the total cooling tower area to be divided into a number of segments. The heat transfer in each segment is then integrated over the entire cooling tower area to give the total heat transfer. The EnergyPlus model assumes that the enthalpy change across the tower is proportional to the difference in wetbulb temperature. This model is iterative as shown in figure 3-5.

Models for single stage and double stage cooling towers are available in EnergyPlus. The whole cooling tower simulation is completed in two steps. Initially the tower is run to check whether the demand can be met by free convection. For this check, the fan is turned off, while the tower is operating. The tower is said to have met its demand if the outlet temperature is equal to or less than a required set point. If the demand is not met without the fan, the cooling tower fan is turned ON and the simulation is carried out again. For a double stage cooling tower, a similar approach is used to determine the fan speed. The temperatures are then updated to continue the simulation. Figure 3-5 shows the cooling tower algorithm. The model first predicts an outlet temperature for the cooling tower. This outlet temperature is used to calculate all other cooling tower thermal performance parameters including UA, effectiveness etc. Using these performance parameters, a new cooling tower outlet temperature is calculated. Convergence is achieved when the change in outlet temperature do not vary significantly. Once convergence is reached, the water outlet temperature and power consumption are updated.

```
                    ┌─────────────────┐
                    │   Get Model     │
                    │    Inputs       │
                    └────────┬────────┘
                             │
                             ▼
                    ┌─────────────────┐
                    │ Initial Guess for│
                    │   outlet air     │◄──────────────────┐
                    │  Temperature.    │                   │
                    └────────┬────────┘                   │
                             │                            │
                             ▼                            │
                    ┌─────────────────┐                   │
                    │ Calculate Specific│                  │
                    │   heat, UA and   │                   │
                    │ Effectiveness using│                 │
                    │  model equations │                   │
                    └────────┬────────┘                   │
                             │                            │
                             ▼                            │
                    ┌─────────────────┐                   │
                    │  Calculate new  │                    │
                    │    outlet       │                    │
                    │  temperatures.  │                    │
                    └────────┬────────┘                   │
                             │                            │
                             ▼                            │
                    ╱─────────────────╲                   │
                   ╱        Is          ╲                 │
        YES       ╱    Convergence       ╲   NO           │
     ◄───────────        reached          ───────────►    │
                  ╲                      ╱                 │
                   ╲                    ╱                  │
                    ╲─────────────────╱                   │
         │                                      │         │
         ▼                                      ▼         │
┌─────────────────┐                   ┌─────────────────┐ │
│ Calculate water outlet│              │ Update the Outlet│ │
│  temperature, fan │                  │ temperature and  │─┘
│  power and heat   │                  │    continue      │
│    transfer       │                  └─────────────────┘
└─────────────────┘
```
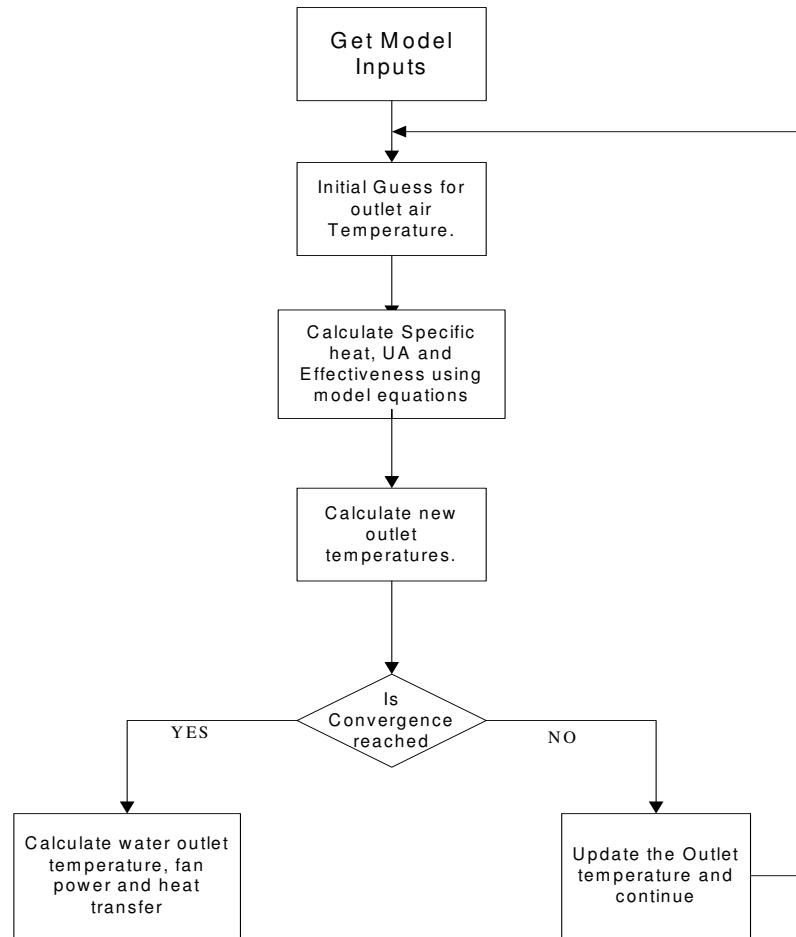
**Figure 3-5.** *Flow Chart Describing the Cooling Tower Model*

### 3.2.4 Circulating Pump Model

Aqueous fluids are used as the heat transfer medium in ground source heat pump systems, including hybrid systems. The flow in the system is maintained with the help of one or more circulating pumps. The type of pump generally used in ground source applications is a centrifugal pump. In EnergyPlus, the pump reacts to the flow rate requested by the demand side of the plant/condenser loop. The pump however is not placed on the demand side, but is the first component on the first branch of the plant/ condenser supply side. The actual loop flow rate is limited by the size of the circulating pump, and set as the larger of supply side and demand side flow requests.

The circulating pump implemented in EnergyPlus is an ideal pump model. The control algorithm for the circulating pump as implemented in EnergyPlus is shown in figure 3-6. The circulating pump can be controlled in four different ways:

- Constant flow: The pump has a constant flow through it irrespective of the demand/supply side request. The pump is in most cases sized for the maximum flow that could occur in the system. This pump is not a good choice where there are large flow demand fluctuations. The only controlling factor in this case is turning the pump ON/OFF.

- Variable flow: The pump adjusts the flow rate between a set maximum and minimum. The demand side and supply side components request a flow rate depending on the loop demand. The pump responds to the maximum of demand side and supply side flow requests. If the flow request is out of bounds, the flow rate is set to one of the limiting values.

Each of these pumps types can have two control schemes, thus giving four possible pump controls in EnergyPlus. The way the algorithm determines the pump control and set the flow rates are shown in figure 3-6 where:

- Continuous flow forces the pump to run even if the demand/supply side does not request any flow. A constant speed pump with this control will run throughout the simulation, even if there is no load on the loop. For a variable speed pump with continuous flow, the pump will always be ON, but the flow will change depending on load. If no load is present, the pump runs at minimum flow.

- Intermittent flow shuts down the pump if there is no load (demand/supply side does not request flow). A constant flow intermittent pump will have full flow if there is a demand and no flow in the absence of any load. For a variable speed pump running on this control, the flow again changes depending on demand/supply side request. For the no load condition, the pump is shut down.

Ground source heat pumps cycle ON/OFF under thermostat control. In practice, the circulating pump is cycled with the heat pump. In EnergyPlus constant flow and intermittent operation most closely approximates GSHP circulating pump operation. However, the variable length system timesteps in EnergyPlus does not give realistic simulation of cycling equipments based on timestep length cycle times.
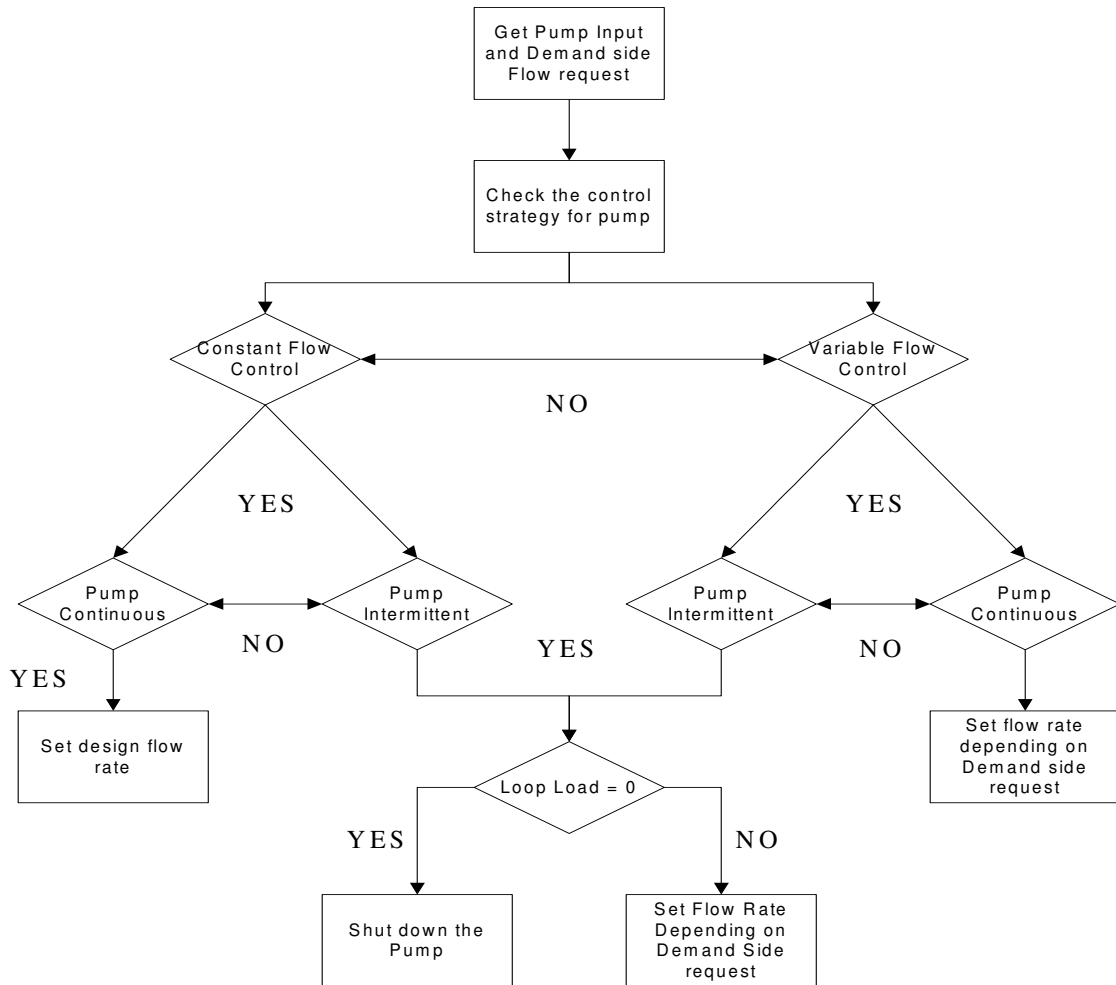
**Figure 3-6.** *Pump Control and Operation in EnergyPlus*

## 3.3 EnergyPlus Plant and Condenser Loops Controls

This section describes the supervisory control algorithm implemented in the EnergyPlus plant and condenser loops. These control algorithms are located in the "ManagePlantCondLoopOperation" subroutine, a high-level manager, which is called each time the plant or condenser loop is simulated. This subroutine determines the operating sequence of available components.

The supervisory control schemes available in the EnergyPlus plant and condenser loops are tabulated in table 3-1. Figure 3-7 shows a flow chart of how plant and

condenser loop components are controlled (turned ON/OFF) in EnergyPlus. The control algorithm is called from the plant/condenser supply side managers. The call is made just after the demand for the loop is calculated. The user input for plant/condenser control is in the form of "operation schemes", which are scheduled for any time of the year. Each operation scheme is defined elsewhere in the input according to its own input syntax. Generally, each operation scheme consists of an equipment list, which specifies the list of components controlled by the respective scheme. Once the loop demand and the current operation scheme are determined, the algorithm loops through all operation schemes to find an available equipment list. The list available at any time step is selected and the components are turned on. All other components are turned off. The loop demand is then distributed to the operating components. Detailed information on the operation schemes and load division are obtained in EnergyPlus Engineering Document and Input/Output Reference (http://www.eere.energy.gov/buildings/energyplus/documentation.html).
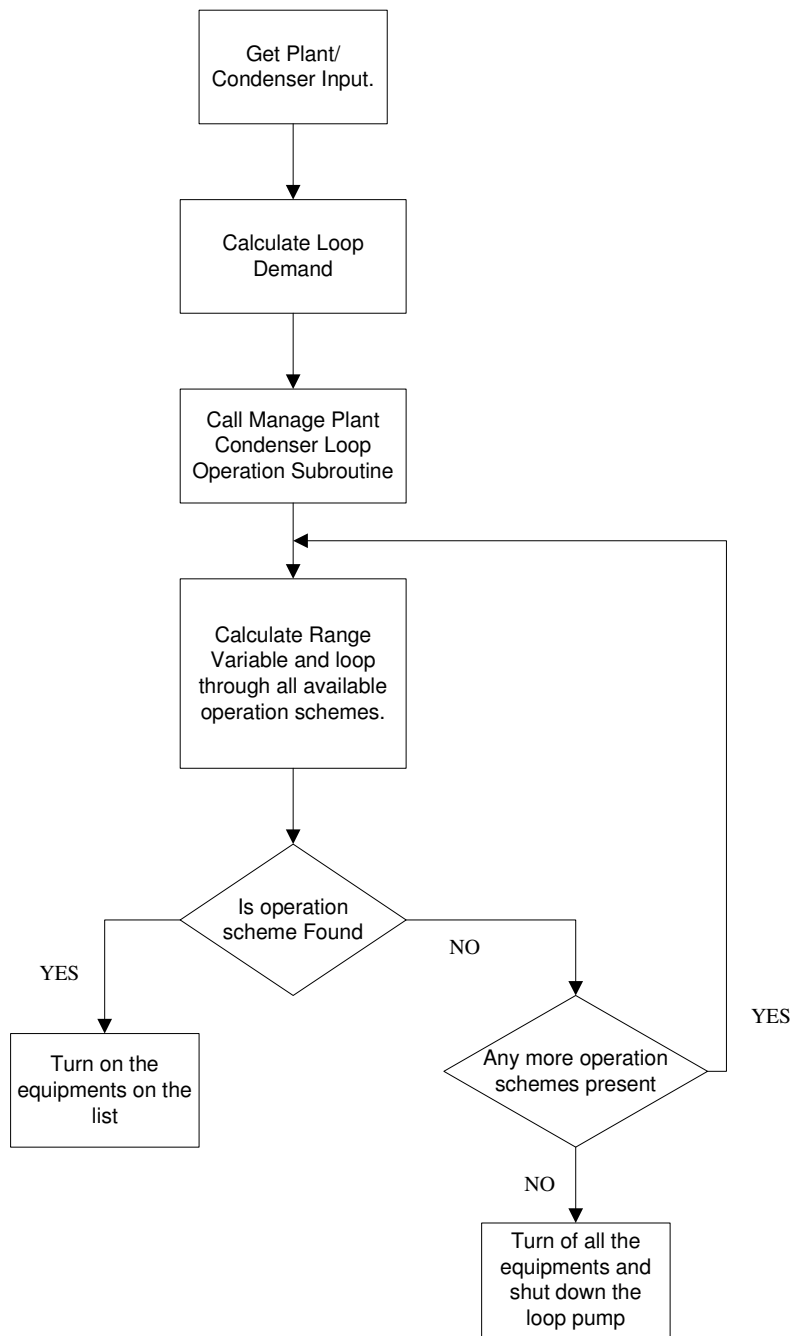
**Figure 3-7.** *Flow Chart Showing EnergyPlus Operation Schemes Logic*

**Table 3-1.** *Supervisory Control Strategies in EnergyPlus Plant and Condenser Loops*

| Control Scheme Type | Plant Loop | Condenser Loop |
|---|---|---|
| Uncontrolled Operation | YES | YES |
| Range based Operation | | |
| 1. Load range based | YES | YES |
| 2. Environmental conditions based | NO | YES |

The controls schemes available in EnergyPlus are described briefly in the following paragraphs. The need for new control schemes for HGSHP systems is also explained.

- Uncontrolled operation scheme: Each component in the equipment list associated with this operation scheme will be turned on when the loop pump is on. This type of control is typically used when there is only one component on the loop, for example a single ground loop heat exchanger.

- Load range based operation scheme: For this scheme, a list of components is associated with a particular load range. At each time step, the loop demand is calculated, and the components corresponding to that load range are turned on. All other equipment on the loop is turned off. Chillers and boilers are typically controlled using this scheme.

- Environmental Range Based Operation Scheme: A list of components is associated with a particular range of an environmental parameter (like wetbulb or drybulb temperature) in this scheme. This scheme works similarly to the

load range based operation scheme. In this case, however, once the loop
demand is calculated the current environmental conditions are used to
determine the list of components to be turned on. Components such as
cooling towers, evaporative coolers fluid coolers etc are controlled by this
scheme.

Though these operation schemes are very useful for general HVAC systems,
including conventional GSHP systems, better control alternatives are required for
HGSHP systems. Since a HGSHP system uses the supplemental heat rejecter only to
reject the excess cooling load, an uncontrolled operation scheme is not appropriate. The
load range based operation scheme could only work if the loop demand was already
known. Environmental range based operation considers only the current state of an
environmental parameter. If we are using a cooling tower as a supplemental heat rejecter,
the performance will be based on the "difference" of the water inlet temperature and the
outdoor wet bulb temperature. This suggests the basis for the new operation schemes
discussed in section 3.4.1.

## 3.4 Modifications to the Simulation Environment

The EnergyPlus simulation environment was significantly modified to
accommodate the analysis of hybrid ground source heat pump systems. These major
changes included:

- Addition of new supervisory control schemes for better control of
  HGSHP systems

- Addition of multi-year simulation capabilities to the program

- Development of a plate heat exchanger model to integrate ground loop with secondary tower loop

The following sections discuss each of these enhancements to the EnergyPlus program.

### 3.4.1 Control Strategies

The new operation schemes are based on control strategies for HGSHP systems used by Yavuzturk and Spitler (1999) as discussed in Chapter 2. This section explains the implementation of these control strategies in EnergyPlus. Two different control strategies viz. a node set point based operation scheme and a delta temperature based operation scheme were implemented.

### 3.4.1.1 Node Set Point Based Operation Scheme

This control scheme is based on turning ON or OFF a list of equipment by comparing the temperature of a reference node to a setpoint. This type of control allows additional equipment to be switched ON or OFF as the loop temperature changes. Although this type of control generally leads to non-optimal control, it can be used for HGSHP systems, when loop temperatures are very high. In such cases, the operation of the supplemental heat rejecter will be justified without any further control.

The control scheme input requires the name of a reference node, the lower and upper values of the control temperature and the corresponding equipment list. At any simulation time step, the reference node temperature is checked to see if it lies between the control set points. If the temperature lies between the setpoints the equipment is

turned ON otherwise the equipment is turned OFF. The idd object for this control scheme is shown in Appendix A.

### 3.4.1.2 Delta Temperature Based Operation Scheme

As the name suggests this operation scheme is based on the temperature difference between the outdoor conditions and a specified node condition. A HGSHP system with a cooling tower, for example performs better if the tower is turned on when the heat pump exiting temperature is high. However, if we look only at the heat pump exiting temperature, we may not control the system effectively, since the performance of the tower depends on outdoor wet bulb temperature. Tower operation can be optimized if the difference between the heat pump exiting temperature and the outdoor wet bulb temperature is considered.

This control scheme requires specification of a reference node, which the control module evaluates to determine whether the equipment should be turned ON/OFF. There are two types of delta temperature controls implemented in EnergyPlus:

- Outdoor Wetbulb based: For this type of control, the temperature of the reference node is compared with the outdoor wetbulb temperature. If the difference lies in the range specified, the corresponding equipment list is subjected to the action prescribed by the user input. The user can specify if the component has to be turned ON or OFF for any specified range. This type of control will be mostly used by cooling towers.

- Outdoor Drybulb based: Here the reference node temperature is compared to the outdoor dry bulb temperature. Fluid coolers and pavement heat rejecters are better controlled with this type of control scheme.

The idd specification for the outdoor wetbulb based controls scheme is shown in Appendix A.

### 3.4.2 Multi Year Simulation

Most conventional HVAC systems can be sized using two design days and analyzed using an annual simulation. HGSHP systems, on the other hand, must be simulated over the life of the system in order to evaluate the long-term change in ground temperatures and its effect on system performance. Such an analysis determines if a hybrid system is required and, if installed, how it would improve the system performance. Multi-year simulation also allows us to compare the ground temperatures with and without a supplemental heat rejecter.

EnergyPlus uses a weather file to obtain the weather data, which will be used for simulation. The weather file contains hourly values for outdoor drybulb temperature, dew point temperature, relative humidity, solar irradiation, wind speed etc. Wetbulb temperature and humidity ratio are calculated from psychometric relations. Some weather files may even contain ground temperatures. The ground temperatures are determined using a heuristic approach. The ground temperatures given in weather file are not suitable for ground source applications since the values do not correspond to deep ground. The way EnergyPlus interacts with the weather file is shown in figure 3-8. As seen in the figure the interaction between the simulation and the weather file occurs at each time step. The first step in the simulation is to call the weather file and get the information for the present time step. EnergyPlus is capable of sub hourly time steps and the variable time step feature can drive the HVAC simulation time step to as low as one minute. The weather file normally consists of hourly data. Thus, it is necessary to interpolate the

weather data in order to evaluate the weather variables at sub-hourly time steps. These calculations are done in the Weather Manager. Once the weather variables are calculated for any time step, these values will be fed back to the simulation. The individual loop simulations and heat balance calculations takes place after this.
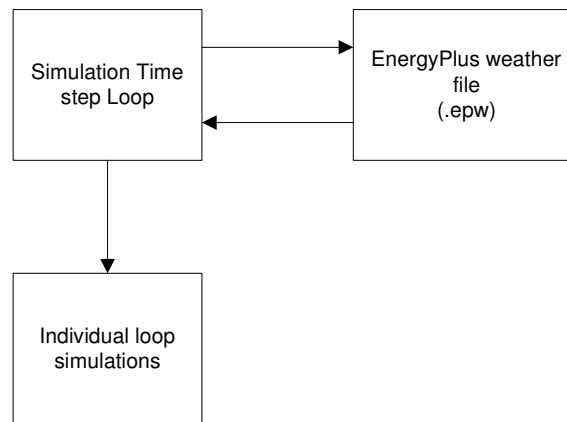


**Figure 3-8.** *Interaction Between EnergyPlus Loop Simulation and Weather File.*

A weather file typically contains data for a whole year, with a header on the first line of file that gives the details of the weather file. A multi-year simulation either reads a single weather file recursively or combines multiple years of data in a single file.

- Weather file with one-year of data: This type of weather file is the same as the weather files used in previous versions of EnergyPlus. It contains data for 8760 hours. To achieve a multi-year simulation using standard weather files the weather manager and the simulation manager were modified to read the file recursively.

- Weather file with multiple years of data: A Weather file of this type is produced by combining several years of data under a single header. To achieve multi-year simulation using long weather files, weather manager was modified such that the variables related to the day of simulation was

reset to the first day of simulation run period. This was necessary to

calculate all the solar calculations. The inbuilt link between the weather

file data and the day variables were modified.

The input data dictionary (idd) was modified to accommodate the multi-year

simulation capability. A numeric field was added to the "Runperiod" object of the

dictionary. This field specifies the number of years the simulation needs to be performed.

The idd object that contains the number of years of simulation is shown in Appendix A.

## 3.5 Plate Heat Exchanger Model

### 3.5.1 Necessity for Plate Heat Exchanger

The EnergyPlus loop architecture contains a splitter and a mixer between the inlet and

outlet branches as shown in figure 3-9. There can be any number of branches between the

splitter and mixer, which give users a lot of flexibility in configuring loop equipment.

Figure 3-9 shows a typical condenser loop with splitter, mixer and multiple branches.

Any branch between the splitter and mixer can contain more than one component. In a

series configuration, the components are connected to each other in a single branch,

whereas in a parallel configuration components are on different branches.
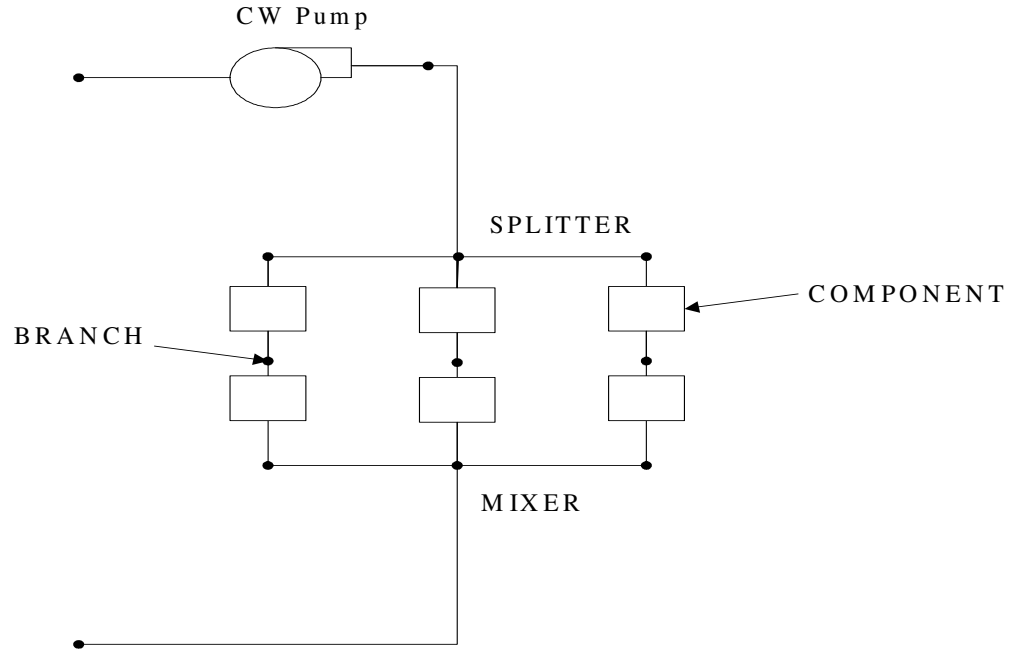
**Figure 3-9.** *Typical EnergyPlus Condenser Loop with Splitter and Mixer*

Figure 3-10 shows a typical series HGSHP configuration. Since the supplemental heat rejecter is used intermittently to reject excess heat series connection requires a plate heat exchanger and a secondary loop as shown in figure 3-10. The secondary loop is configured as a separate condenser loop, which is connected to the main ground heat exchanger loop by means of the plate heat exchanger.
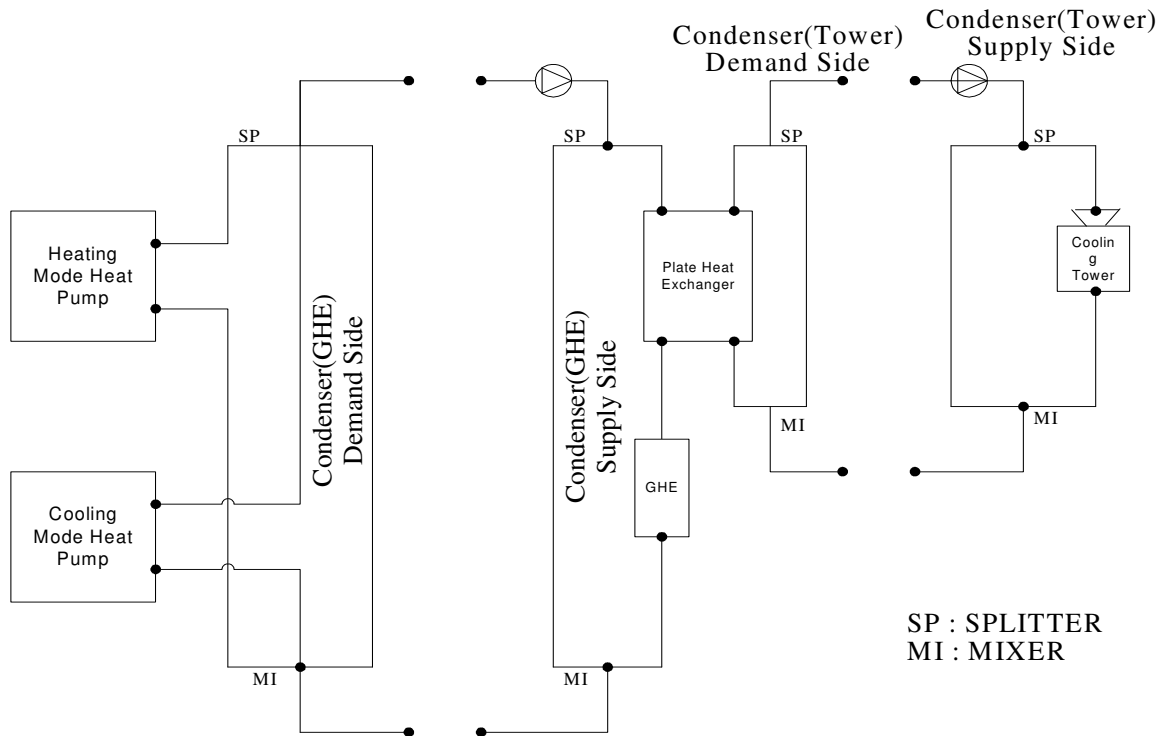
**Figure 3-10.** *EnergyPlus Loop Schematic for Plate Heat Exchanger HGSHP System*

## 3.5.2 Model Description

The plate heat exchanger is model developed for EnergyPlus as a simple cross flow heat exchanger. The two sides of the heat exchanger are the supply side branch of one condenser loop and the demand side branch of other condenser loop. The supply side branch is generally the ground heat exchanger loop whereas the other side is usually the demand side of the supplemental heat rejecter loop. The model does not take into account heat exchanger pressure drops. In addition, it is assumed that no phase change occurs during the heat transfer process. An Effectiveness- NTU (number of transfer units) approach is used to calculate the outlet temperatures and the heat transfer rates. The NTU value for the heat exchanger is calculated based on the fluid side that has the minimum capacity. The supply side and demand side capacity is given by:

$$C = \dot{m}c_p$$

$$C_{min} = Min(C_{Demand}, C_{Supply})$$

$$C_{max} = Max(C_{Demand}, C_{Supply})$$

Where $C$ = capacitance (J / s-K)

$\dot{m}$ = mass flow rate of fluid (Kg/s)

$c_p$ = specific heat of fluid (J/kg-s)

$C_{min}$ = minimum of demand side and supply side capacitance

$C_{max}$ = maximum of demand side and supply side capacitance

The Number of Transfer Units is then calculated as:

$$NTU = \frac{UA}{C_{min}}$$

Where *NTU = Number of Transfer Units.*

*UA = overall heat transfer coefficient (W/K)*

*$C_{min}$ = minimum capacitance rate* (J/s-K)

Once we calculate the NTU, we can easily calculate the effectiveness of the heat exchanger using the following equation (Incropera and Dewitt).

$$\varepsilon = 1.0 - e^{\left[ \frac{NTU^{0.22}}{\xi} \left( e^{(-\xi \cdot NTU^{0.78})} - 1 \right) \right]}$$

Where:    $\varepsilon$ = effectiveness of heat exchanger.

$\xi$ = ratio of maximum and minimum capacitances ($C_{min}/C_{max}$)

The overall heat transfer rate is then calculated as:

$$\dot{Q} = \xi \cdot C_{min} (T_{Sup,in} - T_{Dem,in})$$

Where:    $\xi$ = effectiveness of heat exchanger

$\dot{Q}$ = *overall heat transfer rate (W)*

$T_{sup,in}$ = *supply side inlet temperature ($^oC$)*

$T_{dem,in}$ = *demand side Inlet temperature ($^oC$)*

The inputs, outputs and parameters for the plate heat exchanger model are given in figure 3-11. The only parameter for the model is the total UA value for the heat exchanger. The UA, which is a parameter, is assumed to remain constant throughout the simulation.
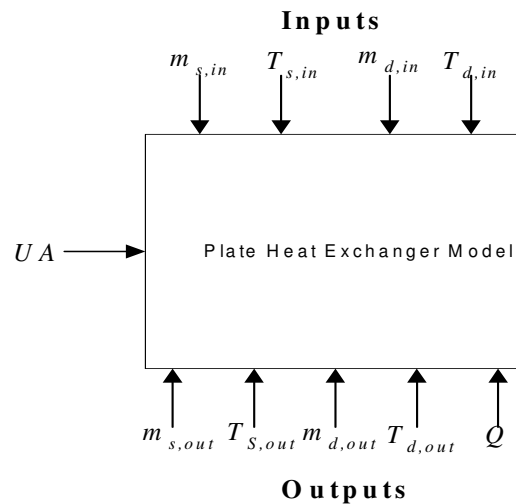


**Figure 3-11.** *Input and Outputs for Plate Heat Exchanger Model*

## 3.5.3 Model Implementation

The Plate heat exchanger model is implemented in EnergyPlus as a separate module on the condenser supply loop side. The plate heat exchanger is called from the condenser supply side manager. The module contains separate routines for input, initialization, main calculation, update and reporting. Figure 3-12 shows the flow of the module as implemented in EnergyPlus.

42
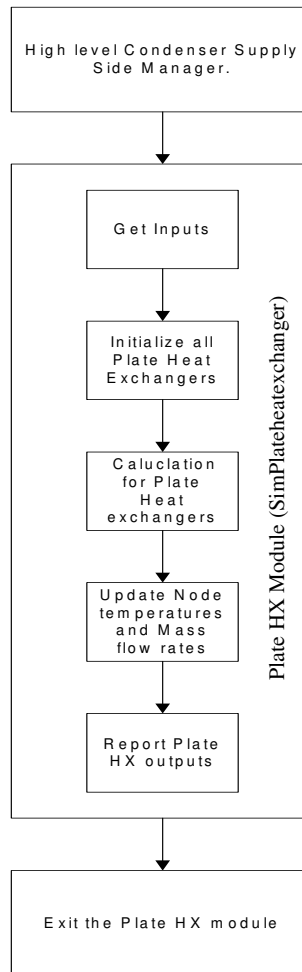
**Figure 3-12.** *Flow of Plate Heat Exchanger Module in EnergyPlus*

The addition of plate heat exchanger model in EnergyPlus required addition of a new key word, "HEAT EXCHANGER: PLATE: FREE COOLING" to input data dictionary (idd). The new object with all the related fields is explained in Appendix A.

# CHAPTER 4

## SIMULATION AND VERIFICATION OF HYBRID

## GROUND SOURCE HEAT PUMP SYSTEM SIMULATION

### *4.1 Introduction*

The method used to model the HGSHP system was explained in chapter 3 along with a description of the models involved in the simulation. Ideally, each new simulation model would be validated with experimental data. Since experimental data was not available from the OSU test facility at the time of this writing, the results were verified by comparison with the results published by Yavuzturk et al. (2000). The simulation input is specified to match the previous work as closely as possible. The verification methodology and comparison to the published results are explained in this chapter. Later, the different configurations used in HGSHP systems (viz. Series and Parallel) are shown and discussed. Finally, a simulation model of the test facility built at Oklahoma State University was developed in anticipation of experimental data available in the future.

## 4.2 Verification Methodology

### 4.2.1 Building Description

The building used for simulation purposes is an office building located west of Stillwater, OK. The building has a total area of 1320 m$^2$ (14,205 ft$^2$) conditioned by a conventional Ground source system. For simulation purposes, a HGSHP system input model is constructed for the building. Yavuzturk et al (2000) calculated the loads for this building for two different locations viz. Tulsa, OK and Houston, TX using BLAST (1986). The locations are selected so that the behavior of the HGSHP system in both temperate and hot humid climates can be studied. In order to calculate the building loads and subsequent simulations the following assumptions are made:

- The building is divided into eight thermal zones, and each zone is conditioned by "purchased air", an ideal EnergyPlus system.

- Each zone is controlled by its own thermostat. The daytime setting for the thermostat is 20.0$^0$C (68.0$^0$F) and the nighttime set point is 14.4$^0$C (58$^0$F).

- The building is served by two separate heat pumps one each for heating and cooling. The heat pump is served by a conventional ground source or Hybrid system.

- For all interior heat gains in the building 70% is considered radiant.

- Occupancy for the office is assumed to be one person per 9.3m$^2$ (100 ft$^2$). The heat gain from each person is 131.9 W (450 BTU/hr).

- The equipment gain is taken as 12.2 W/m$^2$ (1.1 W/ft$^2$), while lighting gains are set to 11.1 W/m$^2$ (1 W/ft$^2$).

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The above assumptions are used to calculate the loads for the building using EnergyPlus. In calculating the loads, the system is not considered. Instead, "purchased air" is used to meet loads. This eliminates the air system effects from load calculation. The loads obtained for both climatic conditions are shown in figure 4.1. It is seen from the plots that the loads for Houston, TX result in a cooling dominated building. For the Tulsa climate, the cooling load decreases while the heating load increases. The loads plotted in the figure are hourly loads with the cooling loads plotted in the negative direction. The load predicted by EnergyPlus had same trends as the loads obtained by Yavuzturk. However, the EnergyPlus slightly over predicted the loads in the peak summer. The peak cooling load predicted by EnergyPlus exceeded the loads used by Yavuzturk by 3%. For rest of the year the loads matched within 0.5%.
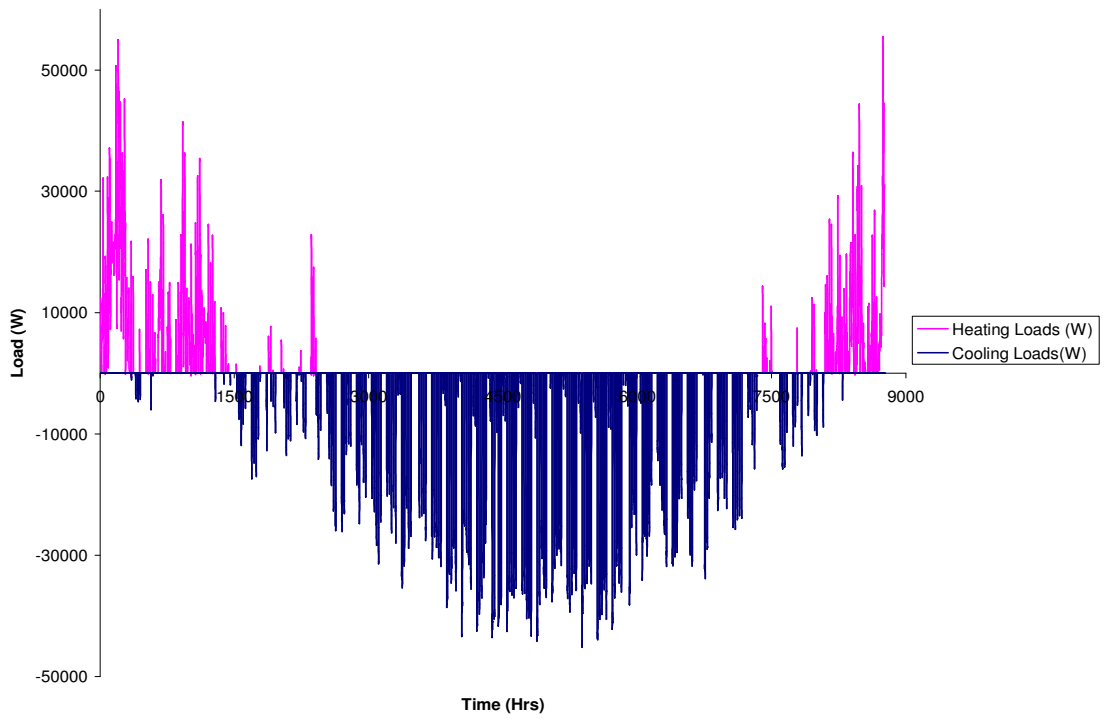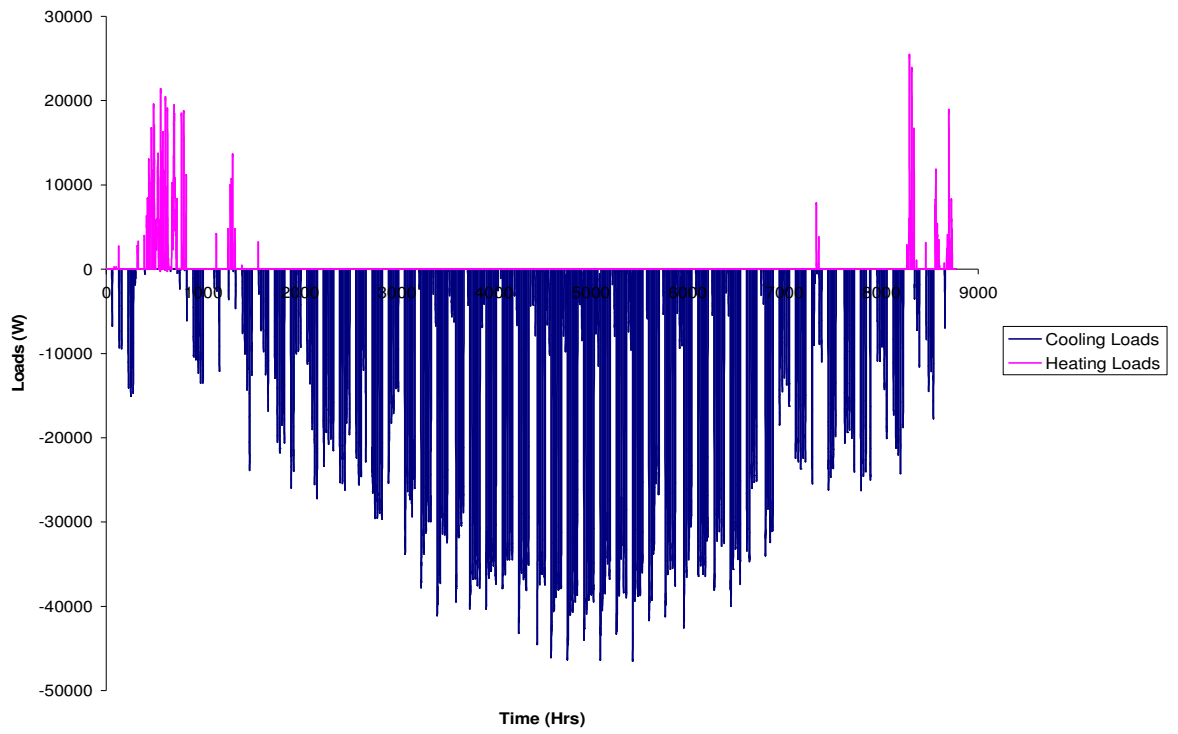
**Figure 4-1.** *Loads for Tulsa, OK (top) and Houston, TX (Bottom)*

## 4.2.2 Simulation Cases & Results

Three cases described by Yavuzturk (2000) are used in this verification. The test cases are selected such that different control strategies for HGSHP systems are investigated. Three cases were selected for verification. For each, the corresponding case from Yavuzturk's work is shown in parenthesis.

- Case 1 (Base case): The plant system consists of conventional ground source system alone.

- Case 2 (Set point control, Case 3b): In this case, the bore field was reduced in size and a cooling tower was used as a supplemental heat rejecter. The cooling tower loop is operated using set point control based on the heat pump entering fluid temperature (EFT). For this case, the set point is $36^0$C ($96.8^0$F).

- Case 3 (Differential control, Case 4a): The cooling tower loop, is controlled by a differential control strategy which looks at the difference between the heat pump EFT and the outdoor wetbulb temperature and compares the difference to a threshold value. For this case, a threshold value of $2^0$C ($3.6^0$F) is selected.

Case 2 and case 3 do not have a dead band for operation. This means that the tower is turned on when the control variable is above the set point and turned off when the value drops below the set point. Results for each case and comparison with results obtained by Yavuzturk are discussed in following sections.

### 4.2.2.1 Case 1 (Base Case) Results

The base case models the building a conventional ground source heat pump system. For simulation purposes, it is assumed that a single ground loop serves two separate heat pumps, one each for heating and cooling. The ground loop is sized and g-

functions are obtained using a modified version of the GLHE-Pro software. The borehole field for Houston consists of 36 boreholes (6x6, Rectangular configuration), with each borehole of depth 76.2 m (250 ft). For Tulsa, the borehole field is reduced to 16 boreholes (4x4, Rectangular configuration), keeping the borehole depth the same. The ground loop parameters used in the simulation are given in table 4.1 for both locations.

**Table 4-1** *Bore Field Parameters for Case 1*

| GLHE Parameters | Houston, TX | Tulsa,OK |
|---|---|---|
| Number of boreholes | 36 (6x6 rectangular) | 16 (4x4, rectangular) |
| Depth of borehole, m (ft) | 76.2 (250) | 76.2 (250) |
| Borehole radius, m (ft) | 0.089 (0.291) | 0.089 (0.291) |
| Shank spacing, m (ft) | 0.0032 (0.0104) | 0.0032 (0.0104) |
| Grout conductivity, W/m-K (Btu/hr-ft-F) | 1.47 (0.85) | 1.47 (0.85) |
| Borehole spacing, m (ft) | 3.8 (12.47) | 3.8 (12.47) |
| Total mass flow rate, Kg/s | 2.271 | 1.703 |
| Ground conductivity, W/m-K (Btu/hr-ft-F) | 2.08 (1.2) | 2.08 (1.2) |
| Far field temperature, $^0$C ($^0$F) | 22.8 (73.04) | 17.22 (63) |

The simulation results for the base case for both Houston and Tulsa climates are shown in figures 4.2 and 4.3. Since Houston has a cooling dominated climate, it is expected that the ground temperature will increase as the years pass by. This increase in ground temperatures is reflected in the increase in fluid temperatures. This is seen in the

figure 4.2, which shows the increase in loop outlet temperatures from the first year to the 20$^{th}$ year. The maximum EFT reached during first year is 29.5$^{0}$C, while for 20$^{th}$ year the maximum EFT is 36$^{0}$C.
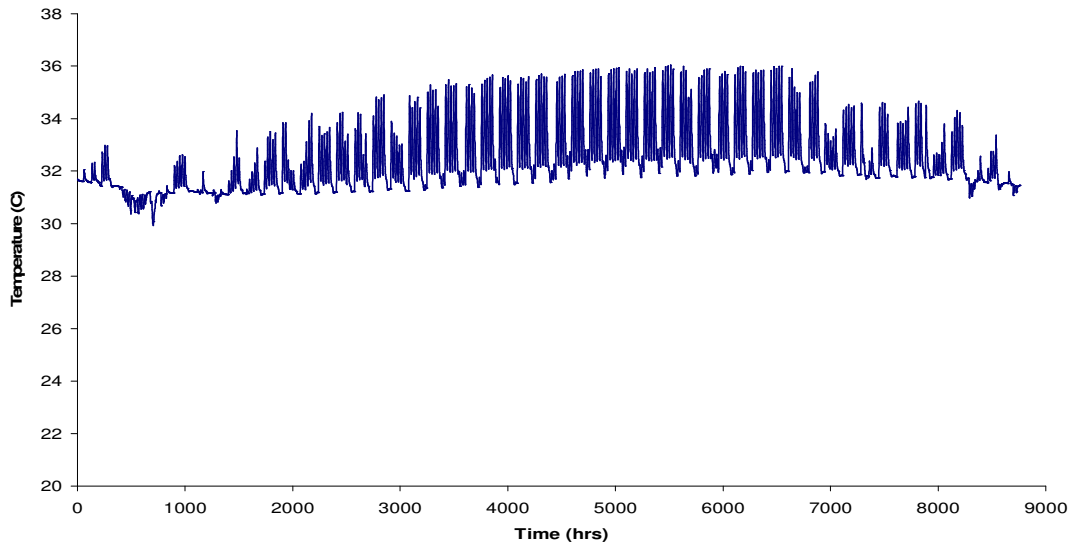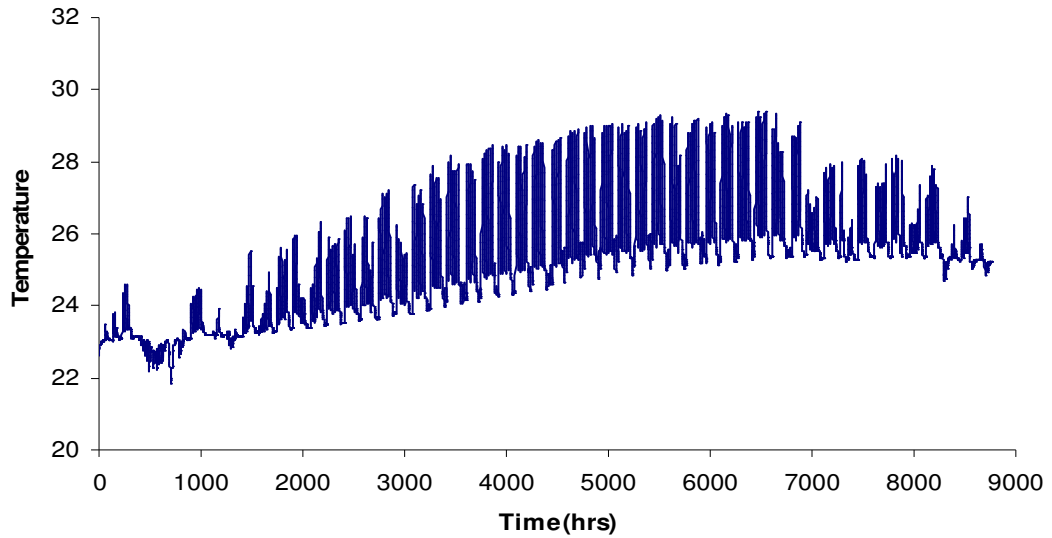


**Figure 4-2.** *Ground Loop Water Outlet Temperatures for Case 1(Houston, TX) First Year (top), 20th Year (bottom)*

For the Tulsa climate, the heating loads are considerable and therefore the increase in loop temperature is expected to be less than that in Houston case. The loop temperature for the first year is shown in figure 4-3. The heat pump EFT for the first year is $31^0$C, while the temperature in $20^{th}$ year is found to be about $36^0$C. In both cases the loop temperatures increases from first year to $20^{th}$ year. This justifies the implementation of HGSHP systems for both climates, though such an attempt is expected to be more beneficial for the Houston, TX climate where load imbalance is more pronounced and there is more demand for supplemental heat rejection. Comparison of the EnergyPlus simulation and Yavuzturk results is shown in tables 4.2 (Houston, TX) and 4.3 (Tulsa, OK).
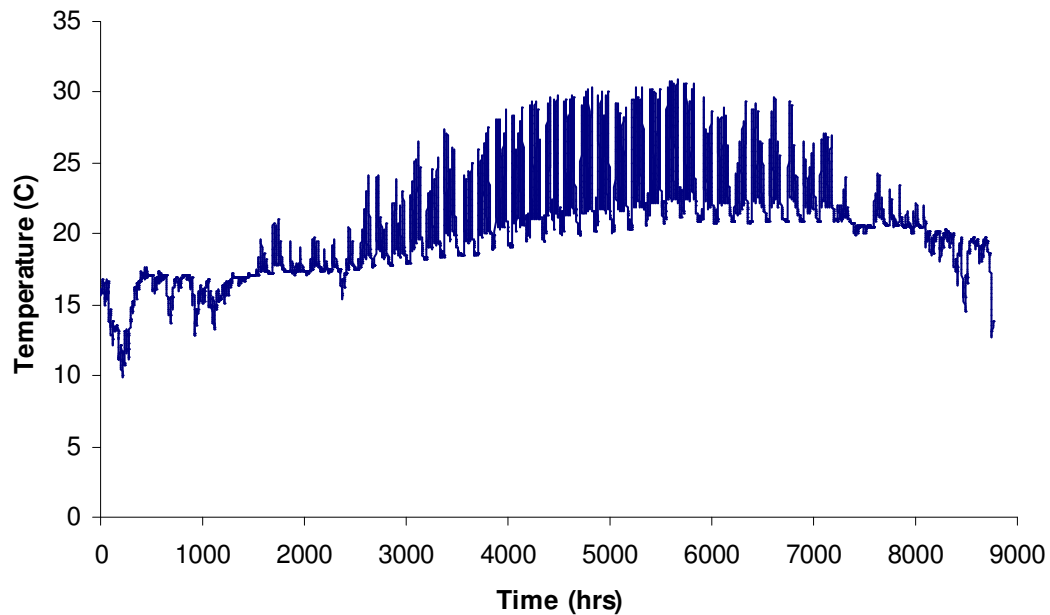


**Figure 4-3.** *First Year Ground Loop Water Outlet Temperatures for Case 1(Tulsa, OK)*

**Table 4-2** *Comparative Results for Case 1 (Base Case) for Houston, TX*

| | EnergyPlus simulation | | Yavuzturk results | |
|---|---|---|---|---|
| | First yr | 20th Yr | First Yr | 20th Yr |
| Heat pump EFT ($^0$C) | 29.5 | 36 | 30 | 35.9 |
| Heat pump energy consumption (kWh) | 19538 | 25591 | 20399 | 25904 |

**Table 4-3** *Comparative Results for Case 1 (Base Case) for Tulsa, OK*

| | EnergyPlus simulation | | Yavuzturk results | |
|---|---|---|---|---|
| | First yr | 20th Yr | First Yr | 20th Yr |
| Heat pump EFT ($^0$C) | 31 | 36 | 31.9 | 35.8 |
| Heat pump energy consumption (kWh) | 17283 | 20483 | 17931 | 20660 |

The tables show that the loop temperatures from both simulations are within $0.2^0$C for the 20th year, for both climates. The increase in Houston EFT after 20 years was found to be $6.5^0$C, whereas for Tulsa, the increase in temperatures was $5^0$C. This is a design parameter reflected in the sizing of ground loops. From the above tables, we can see that the heat pump power consumption also matches. The difference found in heat pump power consumption can be explained by the fact that Yavuzturk et al. use an equation fit model to predict the power, whereas the EnergyPlus model uses a parameter estimation model (Jin et al. 2001) for the heat pump. The difference in heat pump energy

consumption for both climates was found to be in the range of 3% to 4% for first year and within 1% for the 20th year.

### 4.2.2.2 Case 2 (Set point control, Case 3b) Results

This case compares the EnergyPlus simulation with case 3b described by Yavuzturk. The bore field in the base case was reduced and a supplemental heat rejecter was added to the second condenser loop. The cooling tower loop was connected to the ground loop by means of a plate heat exchanger. The reduced bore field consisted of 12 boreholes in a 3x4 rectangular configuration for Houston. In the case of Tulsa, the number of boreholes was reduced to 9 in a 3x3 rectangular configuration. For both cases, the depth of each borehole was kept at 76.2m (250 ft). In this case, a cooling tower was used to meet the cooling loads not met by the ground loop. The mass flow rate for the tower is calculated by the rule of thumb 3 gpm/ton ($5.382x10^{-8}$ m3/s per Watt) of capacity. For this case, the cooling tower was controlled using the set point control strategy described in chapter 3. The set point node was the condenser loop supply outlet node. This node can be used instead of the heat pump condenser side inlet node for practical simulation purposes. The EnergyPlus simulation restricts using the heat pump inlet node, since a single heat pump is modeled as two separate heat pumps one each for heating and cooling. The cooling tower is activated if the condenser loop outlet temperature is above $36^0$C ($96.8^0$F). There was no dead band for operation of the tower. In buildings, which require hybrid GSHP systems, the heat pump will operate in cooling mode most often and thus, the heat pump exiting temperature will usually be greater than the entering fluid temperature. This supports the selection of the heat pump EFT as the

set point. Figure 4.4 shows the ground loop exiting temperatures for Houston. Figure 4.5

shows the tower heat transfer for both locations.
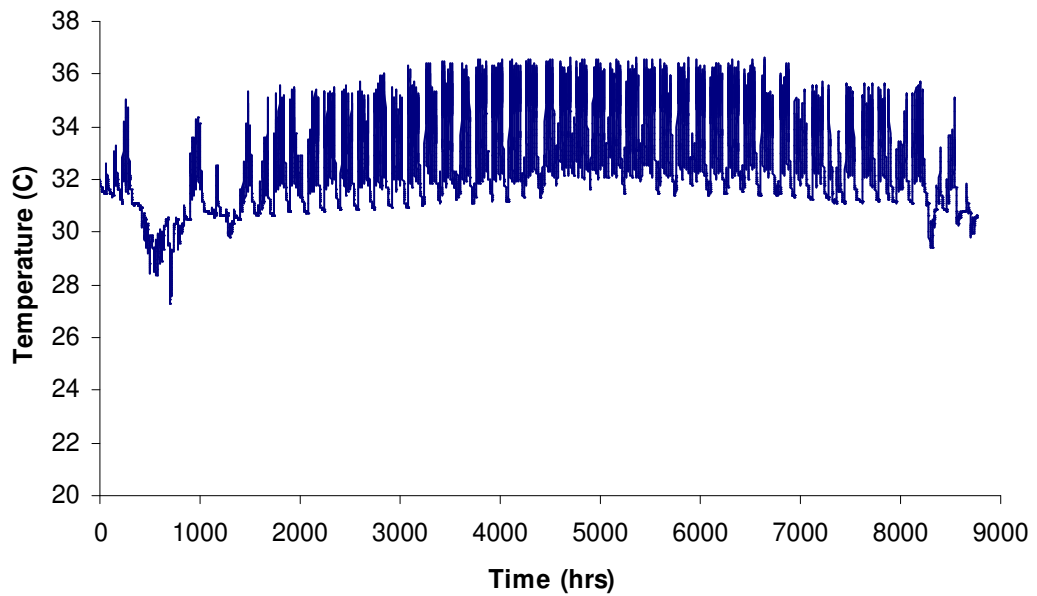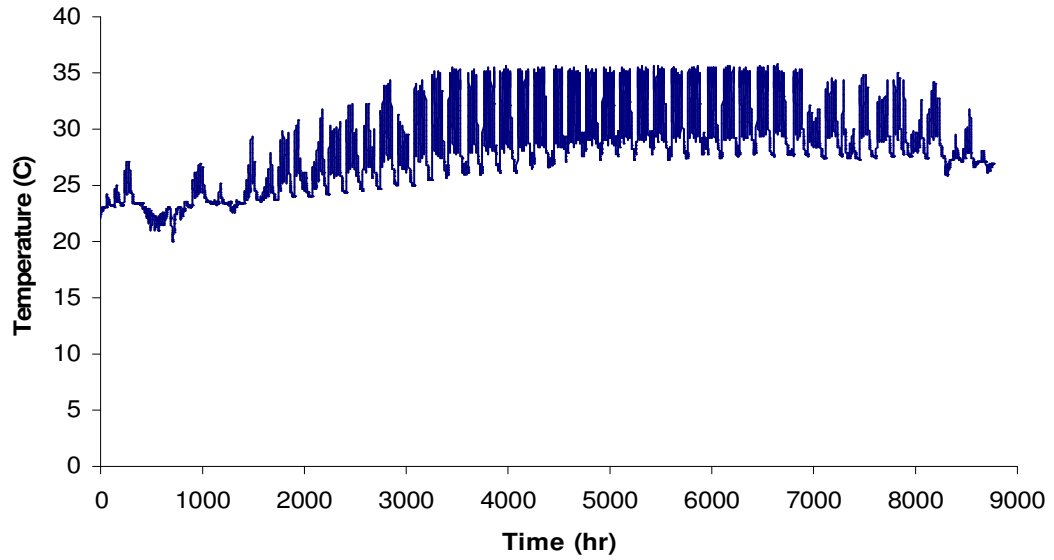


**Figure 4-4.** *Year 1(top) and Year 20 (bottom)Ground Loop Water Outlet Temperatures*

*for Case 2, Houston*

The figures show that the average outlet temperature over the year is higher than in the base case. This occurs due to the increase in loop temperatures during the hours when tower is not turned on. The tower remained OFF even during times when there was potential to reject heat. This combined with the smaller loop size resulted in an increase of the loop temperatures. However, during hours when the tower is turned on (during the summer when the node temperatures are above the set point) the loop temperatures are approximately the same as in base case. The HGSHP system when run using the set point control strategy did not balance the loads. Although the heat rejected in cooling tower increases from the first year to 20$^{th}$ year, the loop outlet temperature levels out by the 20$^{th}$ year. The results for case 2 for both climates are shown in tables 4-4 and 4-5.
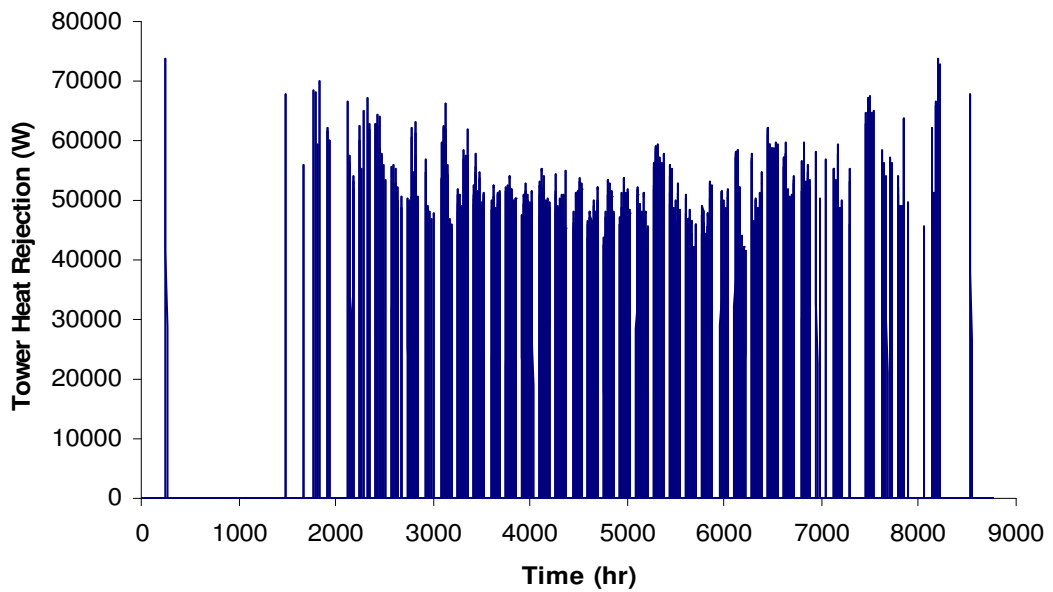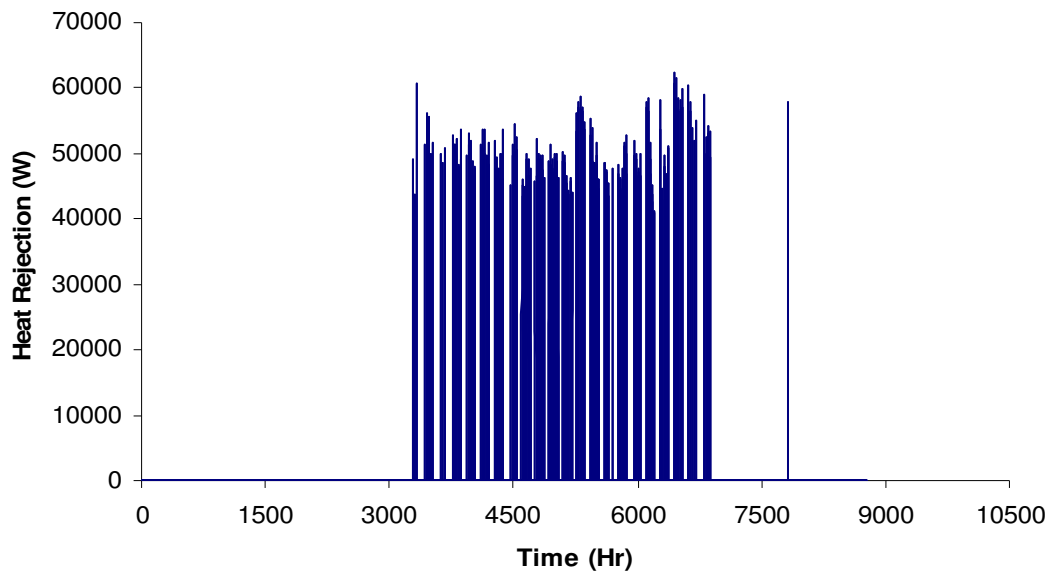
**Figure 4-5.** *Cooling Tower Heat Rejection for First Year (top), 20<sup>th</sup> Year (bottom) for*

*Case 2, Houston, TX*

**Table 4-4** *Comparative Results for Case 2 (Case 3b) for Houston, TX*

| | EnergyPlus simulation | | Yavuzturk results | |
|---|---|---|---|---|
| | First yr | 20$^{th}$ Yr | First Yr | 20$^{th}$ Yr |
| Heat pump energy consumption (kWh) | 24,611 | 25138 | 24,459 | 25653 |
| Cooling tower fan energy consumption (kWh) | 73.6 | 247.5 | 87 | 223 |
| Cooling tower circulating pump energy (kWh) | 24 | 63 | 14 | 36 |

**Table 4-5** *Comparative Results for Case 2 (Case 3b) for Tulsa, OK*

| | EnergyPlus simulation | | Yavuzturk results | |
|---|---|---|---|---|
| | First yr | 20$^{th}$ Yr | First Yr | 20$^{th}$ Yr |
| Heat pump energy consumption (kWh) | 19,750 | 20,973 | 20,742 | 21384 |
| Cooling tower fan energy consumption (kWh) | 29.4 | 92.7 | 31 | 100 |
| Cooling tower circulating pump energy (kWh) | 11 | 28 | 5 | 16 |

Case 2 shows increased loop temperatures compared to the base case for Houston. The ground loop temperatures for the first year for case 2 increased by 5$^0$C compared to the base case for Houston. However, the loop temperatures did not increase greatly in the 20$^{th}$ year due to the increased tower operation in subsequent years. It is seen from the results that case two is not an efficient system. There is 1.6$^0$C increase in loop temperatures for Houston and a 1$^0$C increase for Tulsa between first year and 20$^{th}$ year. This is because the tower is turned on only during the summer and thus, the tower is not utilized to its full benefit. The efficient operation of the system would result in increased tower use every year.

The heat pump power consumption for Houston increased by 26% from base case to case 2, whereas for Tulsa the increase was 14.2%. The EnergyPlus results and the Yavuzturk results for heat pump power consumption differ by approximately 5%. The total annual hours of tower operation increased from the first year to the 20$^{th}$ year. The tower fan power increased by approximately 70% for Houston, while for Tulsa the increase was 68%.

The difference between the EnergyPlus and Yavuzturk results can be explained from the different models used. An equation fit model was used in previous work, whereas the present work uses a Parameter Estimation based model for the heat pump. It is to be noted that the fan power consumption for Yavuzturk's case is more than the EnergyPlus case. This difference in power consumption was due to the dead band control scheme used by the Yavuzturk model. In Houston case, it was seen that the cooling tower did not help the system in the first year. However, increased temperatures in the 20$^{th}$ year resulted in an increased operation of the tower. The dead band had relatively little effect in the 20$^{th}$ year. It was seen from the simulation that the power consumed by heat pump in heating mode did not change much over 20-year period, while the cooling mode power consumption was high due to increased over time due to the rising loop temperature.

### 4.2.2.3 Case 3 (Differential control, Case 4a) Results

For this case, the control strategy is changed to the differential control type described in chapter 3. This case corresponds to case 4a described by Yavuzturk. The cooling tower is controlled based on the difference between the condenser loop outlet temperature and outdoor wetbulb temperature. For this study, the cooling tower is turned on when the temperature difference is greater than a set point of 2$^0$C. No dead band is in

58

effect for the cooling tower operation. The bore field for Houston consists of 12 boreholes (3x4, rectangular) while for Tulsa it is nine boreholes in a 3x3 rectangular configuration. All the bore field parameters were kept the same as in case 2. This control strategy allows the tower to be turned on to cool the ground even in the winter. The heat Pump EFT for Houston is plotted for the first year and $20^{th}$ year in figure 4.6. Figure 4.7 shows the cooling tower heat transfer rate for Houston for first and $20^{th}$ year. Table 4.6 and 4.7 compares the results for Houston and Tulsa.

The figures show that the ground loop temperature has decreased considerably for this case compared to the base case. The hybrid operation in this case was beneficial and helped to cool the ground over the years. This can be seen from the $20^{th}$ year loop temperature, which is slightly less than the loop temperature in the first year. The year round operation of the tower helped the ground to recharge during the winter, thereby decreasing the load on the GLHE during the summer. The tower heat transfer rate is also less in the $20^{th}$ year compared to first year. This is also due to the lower loop temperatures, which resulted in the tower staying OFF for longer periods. This situation is entirely opposite to the case 2 situation, where tower operated much more in the $20^{th}$ year than in the first year.
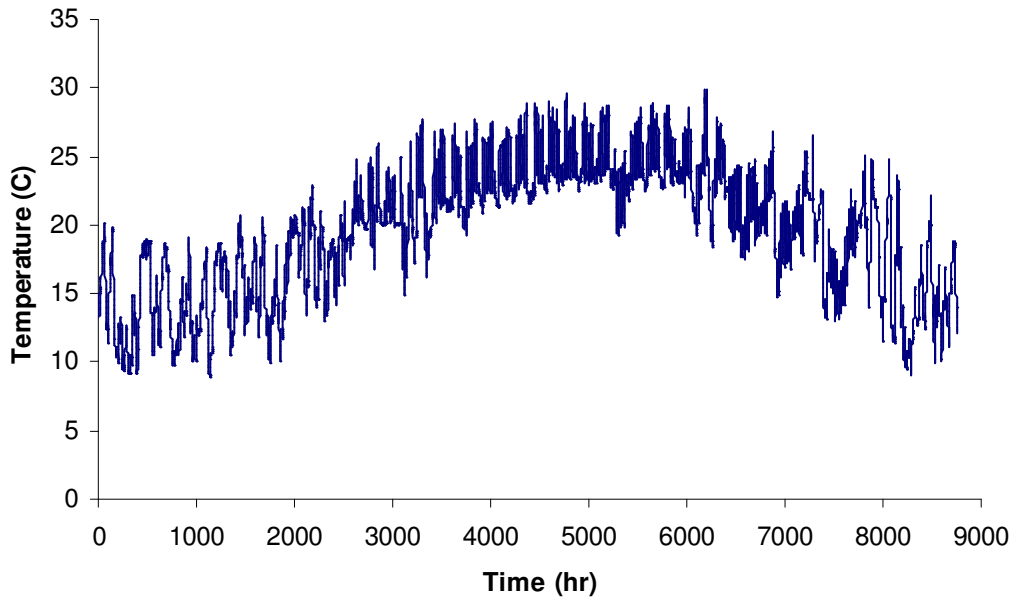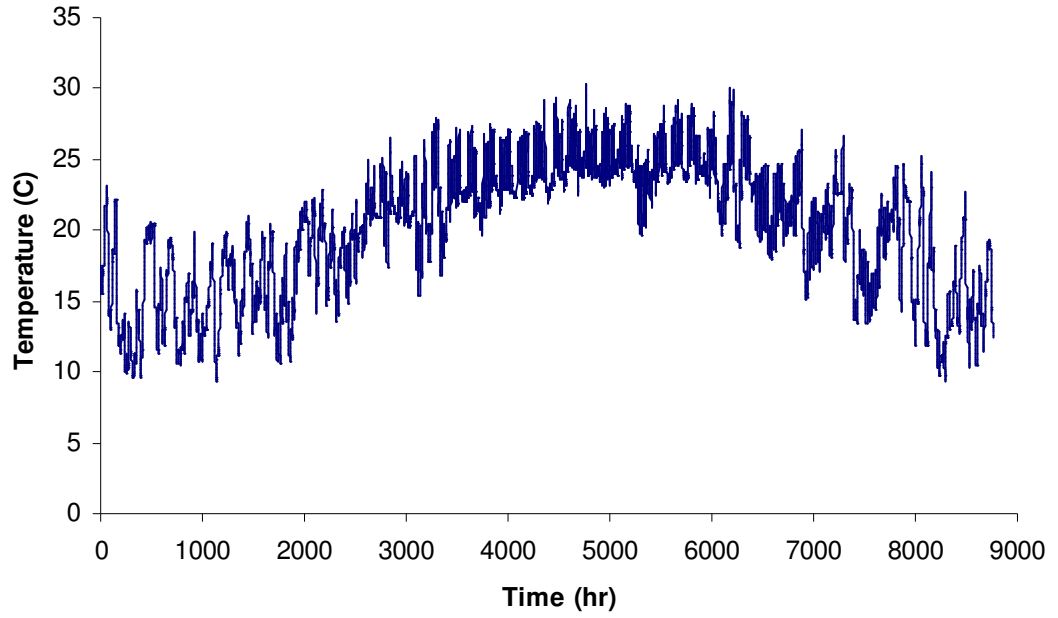
**Figure 4-6.** *Ground Loop Water Outlet Temperatures for first year (top) and 20th year (bottom) for Houston, TX, Case 3*
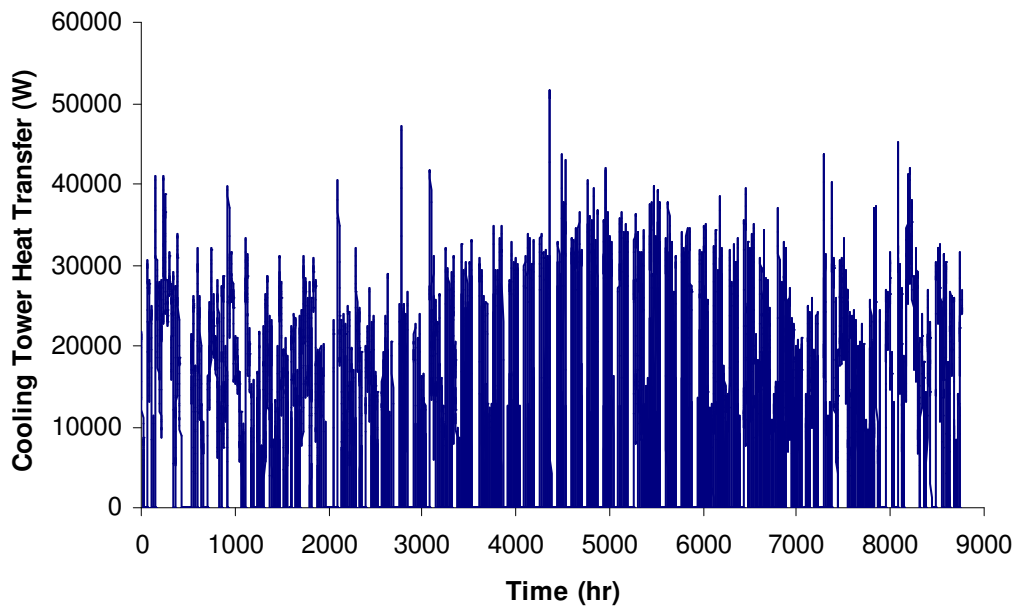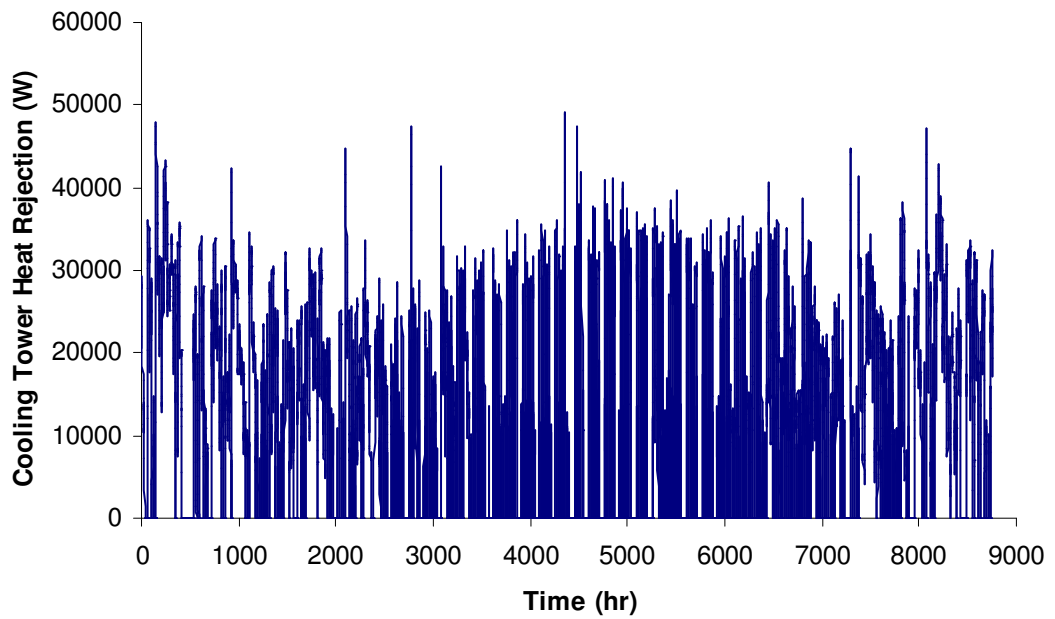
**Figure 4-7.** *Cooling Tower heat Transfer, Case 3, Houston, TX for First year (top) and*

*20ᵗʰ year (bottom)*

**Table 4-6** *Comparative Results for Case 3 (Case 4a) for Houston, TX*

| | EnergyPlus Simulation | | Yavuzturk Results | |
|---|---|---|---|---|
| | First yr | 20th Yr | First Yr | 20th Yr |
| Heat Pump Energy Consumption (kWh) | 18,983 | 18,847 | 19,199 | 19,016 |
| Cooling tower Fan Energy Consumption (kWh) | 1817 | 1619 | 1901 | 1653 |
| Cooling tower circulating pump energy (kWh) | 387 | 325 | 311 | 270 |

**Table 4-7** *Comparative Results for Case 3 (Case 4a) for Tulsa, OK*

| | EnergyPlus Simulation | | Yavuzturk Results | |
|---|---|---|---|---|
| | First yr | 20th Yr | First Yr | 20th Yr |
| Heat Pump Energy Consumption (kWh) | 17,487 | 16,943 | 17,664 | 17,542 |
| Cooling tower Fan Energy Consumption (kWh) | 1873 | 1697 | 1908 | 1719 |
| Cooling tower circulating pump energy (kWh) | 384 | 323 | 312 | 281 |

Tables 4-6 and 4-7 show that the results obtained from EnergyPlus are in good agreement with the Yavuzturk results. The heat pump power consumption for both climates for the first and the 20th year matched the published results within 1%. Heat pump power consumption dropped from first year to 20th year by 1% for Houston and by 3% for Tulsa. Thus, for the building under consideration the differential control strategy is found to be more beneficial than the set point control.

The above tables show that the cooling tower was on for a longer time compared to case 2. It was also seen that the system performance improved from first year to 20th year. This can be attributed to the fact that the operation of the cooling tower in winter

allowed the ground to cool and thereby reducing the average loop fluid temperature. In this case, the ground is cooled every year due to efficient operation of the hybrid system. The tower fan power consumption dropped considerably from the first year to the 20th year. For Houston the fan power dropped by approximately 11%, while for Tulsa the reduction in fan power was 9%. The EnergyPlus results for power also matched with the Yavuzturk results within an error of 4% for Houston and 2% for Tulsa. As previously noted the Yavuzturk model had a dead band in the control of the cooling tower, while the EnergyPlus model did not. This caused the tower to run for longer periods in Yavuzturk's case. However, due to the reduction in loop temperatures over the years, the effect of dead band was minimized. As a result, the difference between the models reduced. Thus, for the 20th year the difference between the power predicted by EnergyPlus and Yavuzturk was only 2%.

## 4.3 Modeling the OSU Experimental HGSHP Facility

An experimental HGSHP facility was recently constructed at Oklahoma State University. Details of the experimental facility are given by Hern (2004). The facility consists of a ground loop heat exchanger, a cooling tower and a pond loop on the condenser side. The facility was designed in such a way that the components can be configured to operate alone or with other components in series or parallel. The simulation case study was developed in anticipation of experimental validation data and covers GLHE and cooling tower configurations. Figure 4.8 and figure 4.9 shows two different configurations considered in the study.

In the parallel configuration, the primary condenser loop (GLHE loop) circulation pump is sized for the combined flow of the GLHE and the plate heat exchanger. For the

series configuration, the pump is sized for the GLHE design flow rate. The loads for the building are obtained using EnergyPlus, with purchased air meeting the building loads. A 3-ton nominal capacity Florida Heat Pump W036 is selected for both heating and cooling mode operation. Heat pump parameters are estimated for use in the simulation. The estimated parameters for both modes are tabulated in table 4.8 and table 4.9. Building loads obtained from EnergyPlus are shown in figure 4.10 (heating loads are plotted on the positive Y-axis).
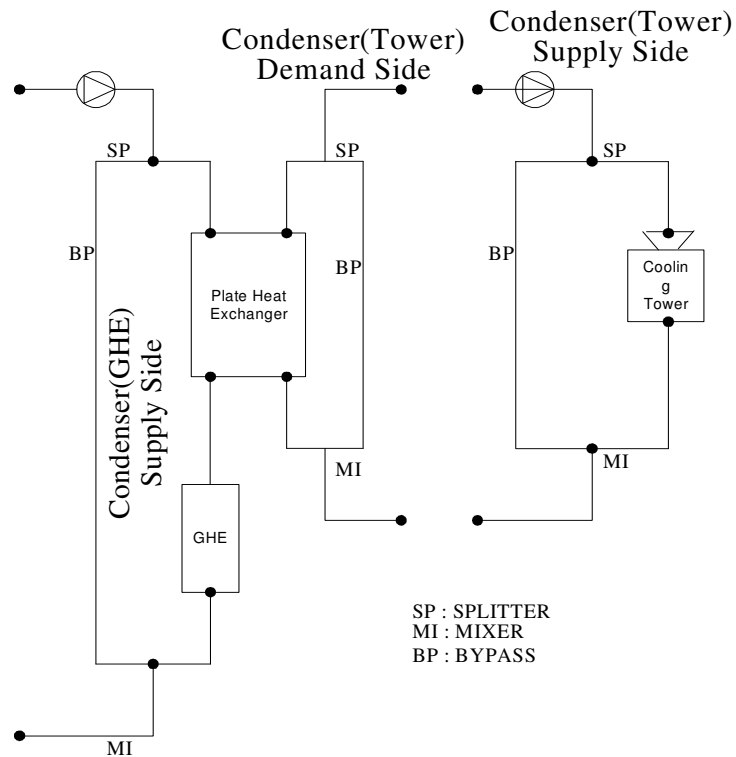


**Figure 4-8.** *GLHE and Cooling Tower in Series Configuration*
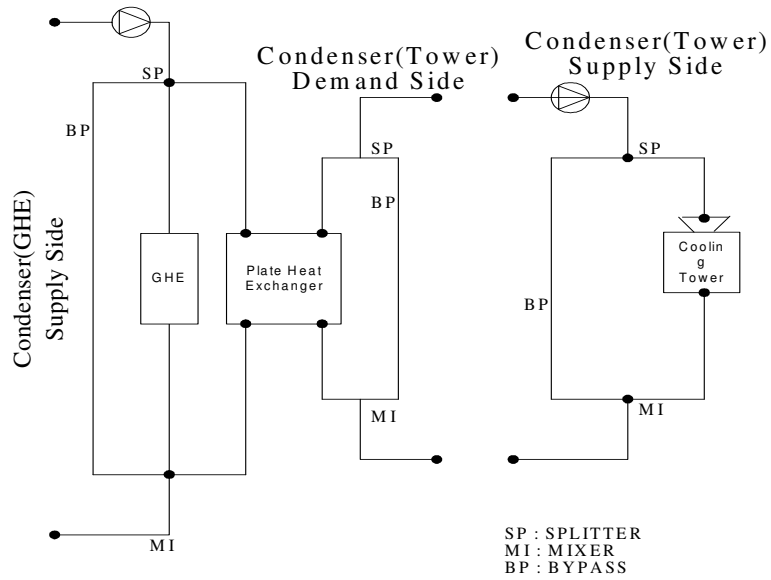
**Figure 4-9.** *GLHE and Cooling Tower in Parallel Configuration*

**Table 4-8** *Cooling and Heating mode Parameters*

|  | Cooling mode | Heating mode |
|---|---|---|
| Load side heat transfer coefficient {W/K} | 1510 | 1217 |
| Source side heat transfer coefficient {W/K} | 2160 | 2129 |
| Piston displacement {m3/s} | 0.00336 | 0 .00254 |
| Compressor clearance factor % | 010758 | 0 .04431 |
| Compressor suction and discharge pressure drop {Pa} | 96426.5 | 103791.7 |
| Superheating {C} | 6.42 | 10.11 |
| Constant part of electro mechanical power losses {W} | 522.4 | 802.1 |
| Loss factor | .8276 | .885 |
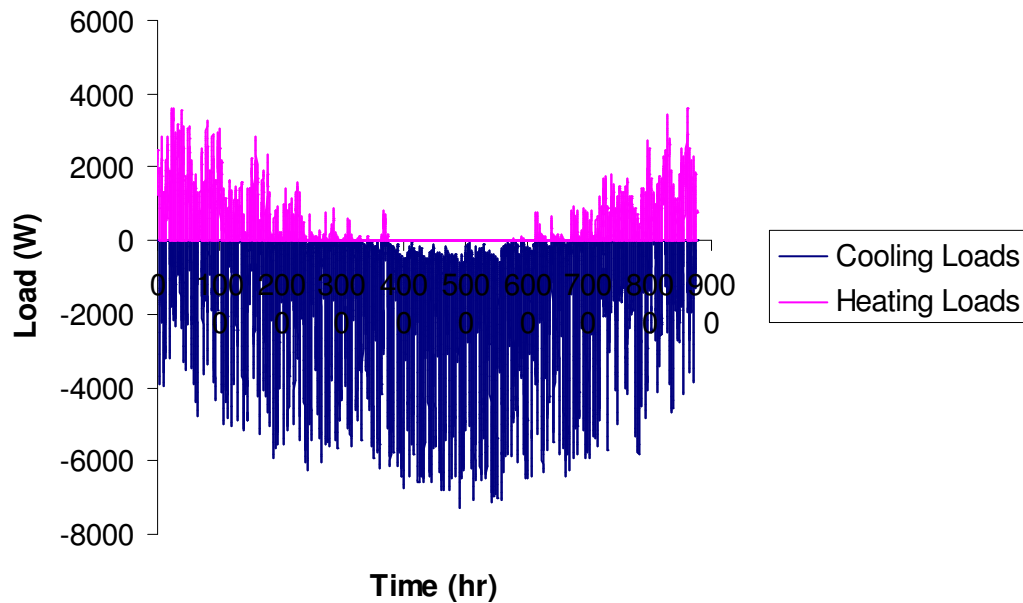
**Figure 4-10.** *Building Loads for OSU System Case Study*

The results obtained from the simulation case study are shown in figure 4-11 and figure 4-12 (cooling tower heat transfer). It can be seen from the results that the peak EFT for the heat pump in a series configuration is $25.44^0$C, while for a parallel configuration the peak EFT is $28.6^0$C. There is approximately a $3^0$C rise in the fluid temperature associated with the parallel configuration. The outlet temperatures tell us that in the series configuration, the fluid is cooled by cooling tower before it goes to the ground loop, where it is cooled further. In the parallel configuration with a constant-speed circulating pump, any flow greater than GLHE design flow rate is bypassed if the plate heat exchanger is off. Since in that case there is considerable flow at higher temperatures, the heat pump EFT also increases resulting in higher temperatures for the parallel configuration. The increased temperatures for the parallel configuration result in higher heat rejection potential for the cooling tower. The heat rejection plots show that

more heat is rejected in the parallel configuration than in the series configuration. There is no need for the cooling tower to operate during peak summer, since the ground temperature is well within the acceptable range. This was due to the pre cooling of GLHE fluid throughout the year by the cooling tower.

We can see in figure 4-11, the cooling tower heat transfer and GHE outlet temperatures from August 11 to August 21. The maximum cooling tower heat transfer rate occurred at hour 6131. It can be observed that due to the operation of the cooling tower the GLHE fluid outlet temperature decreases. There is a marginal rise in the fluid temperature as the cooling tower heat transfer rate drops. When the tower turns OFF after hour 6300, we can see the GLHE outlet temperature rising considerably.
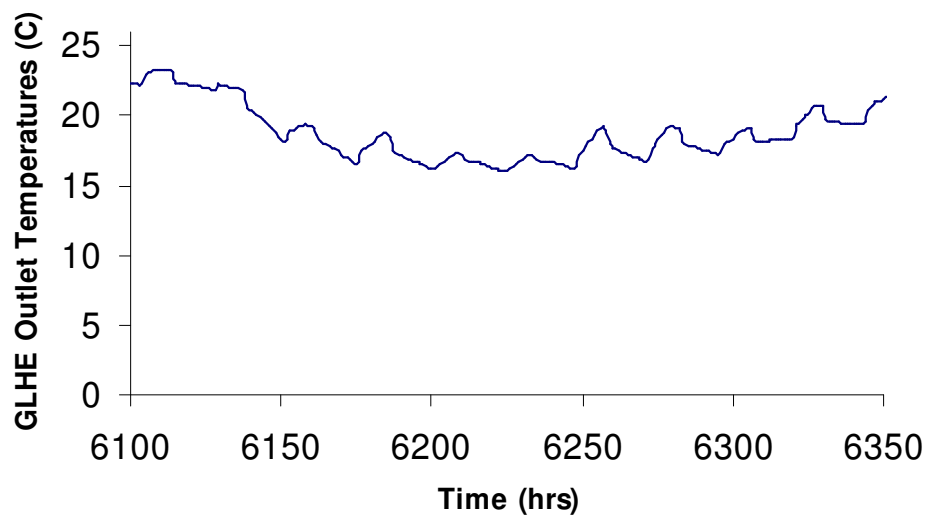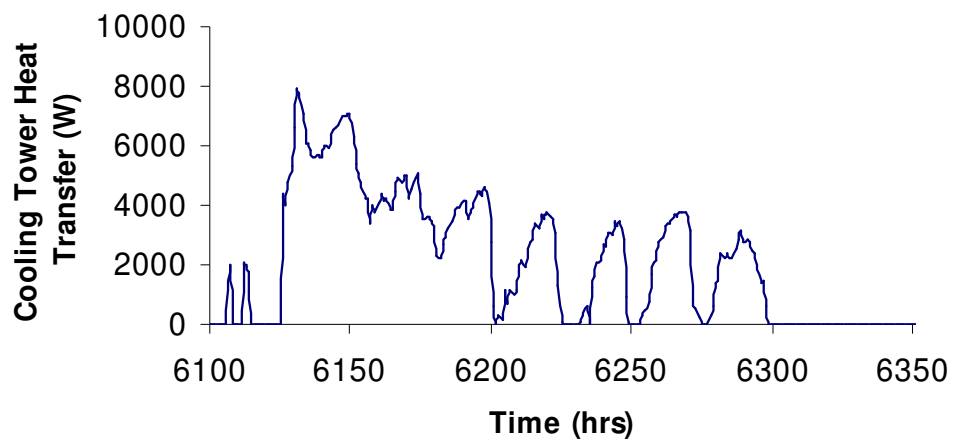
**Figure 4-11.** *Cooling Tower Heat Transfer (top) &GLHE Outlet Temperatures (bottom)*

*from August 11 to August 21*
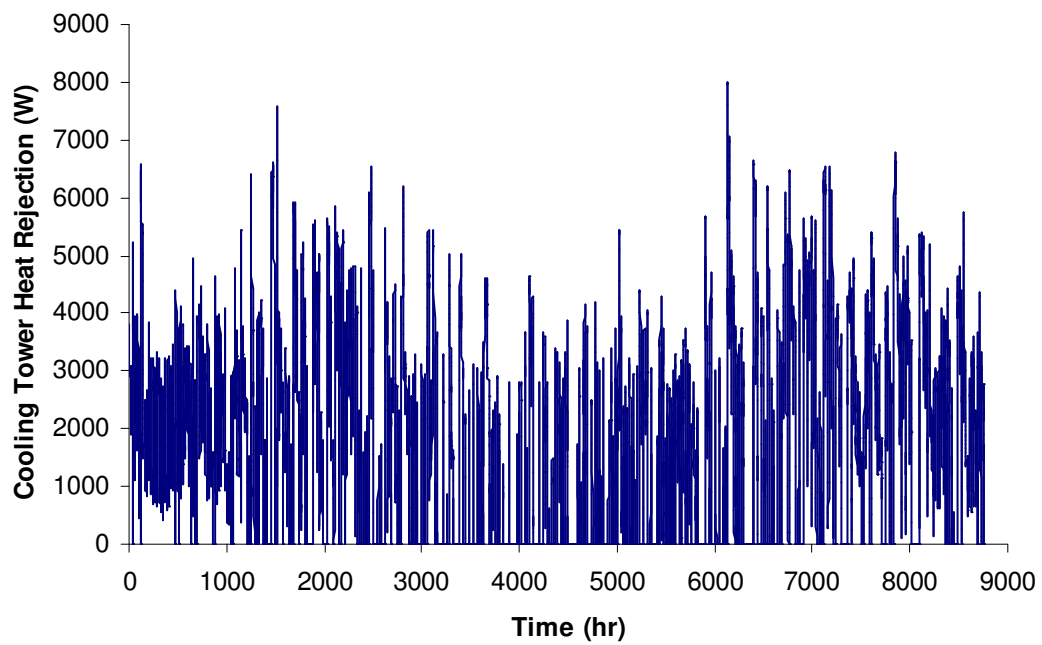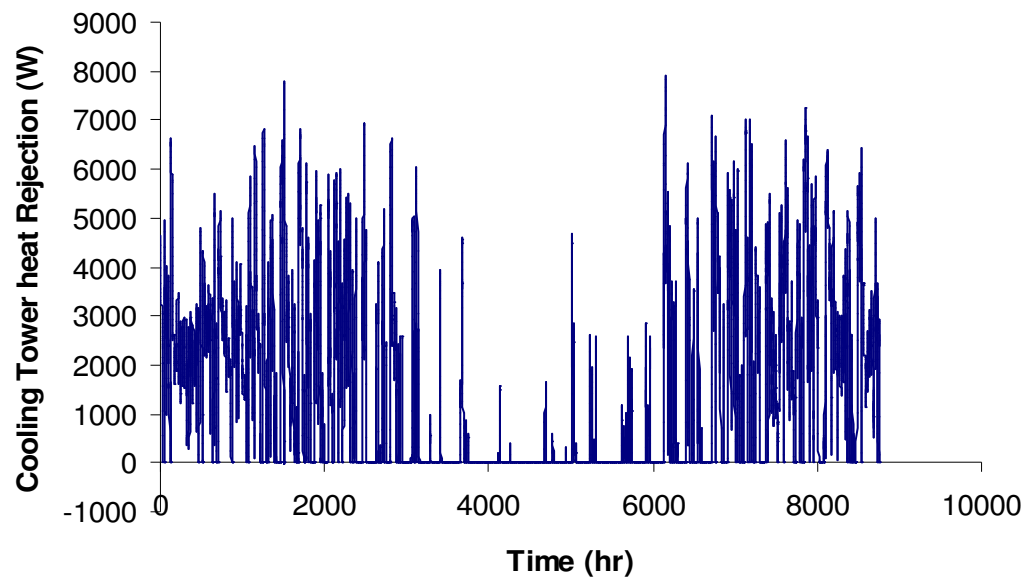
**Figure 4-12.** *Cooling Tower Heat Transfer for Series (top) and Parallel (bottom)*

# CHAPTER 5

## OPTIMIZATION OF HYBRID GROUND SOURCE HEAT PUMP SYSTEMS

*All that is superfluous displeases God and Nature.*
*All that displeases God and Nature is evil.*
*- Dante*

## 5.1 Introduction

Optimization algorithms can be classified as deterministic or stochastic methods (Jain, 1999). Deterministic methods adopt a directional approach to search for the optimal point. The new point is based on the previous test point. Deterministic methods are again divided into gradient based or non-gradient based (direct) methods (Reklaitis, 1993). Gradient-based methods use the derivative information of the function over the optimization domain to determine the direction in which it has to move. A non-gradient method or direct method calculates the objective function value at different points and compares them to determine the optimal point. Some gradient based optimization algorithms are steepest descent, Lagrange multiplier, conjugate gradient etc. Some direct search methods are Nelder mead simplex, Hooke Jeeve's, Box's Complex etc. Stochastic methods, on the other hand use a random distribution of points over the optimization

domain. Rather than having a single initial point to start the optimization, stochastic methods have a family of initial points to start with. At any step of the optimization, the algorithm handles more than one test point. Available stochastic methods include Genetic algorithm (Holland 1975), simulated annealing (Kirpatrick et al 1983), Tabu search (Glover and Laguna 1997) etc.

For thermal system problems, gradient-based methods are not useful since for most cases the behavior of the system is unknown. This uncertainty in behavior makes it difficult to obtain a general mathematical formulation for the whole domain. Direct optimization methods and stochastic methods are thus more appropriate for thermal system problems. The work described in this chapter deals with the optimization of hybrid ground source heat pump systems using the GenOpt program (Wetter, 1999), coupled with EnergyPlus.

## *5.2 GenOpt Interface and Algorithm*

GenOpt (Wetter, M, 2004) is a generic optimization program developed at Lawrence Berkeley Laboratory. The program was developed mainly to be used with building simulation programs that use text based input and output. GenOpt by itself is not an optimization algorithm, but is a library of different algorithms. The user can select from the library, any one algorithm depending on the problem being solved. Some of the algorithms implemented in GenOpt are:

  i.    Nelder Mead simplex method

 ii.    Hooke Jeeves method

iii.    Generalized particle search

 iv.    Particle swarm optimization

v.    Fibonacci search

vi.    Golden section search

The library contains both single variable (v and vi) and multi-variable optimization (i to iv) algorithms. GenOpt and the simulation program are coupled as shown in figure 5-1. GenOpt writes the text input file, which is used by the simulation engine. The simulation writes the processed or raw output to a text file. This output text file is then read by GenOpt to calculate the objective function value. GenOpt can perform basic post processing. However, complex objective functions requiring more intense post processing of outputs call for an intermediate program between the simulation engine and GenOpt. At times GenOpt cannot directly write input files for the main simulation engine. In such cases, the intermediate program must also write the input file, call the main simulation engine and post-process the output files.
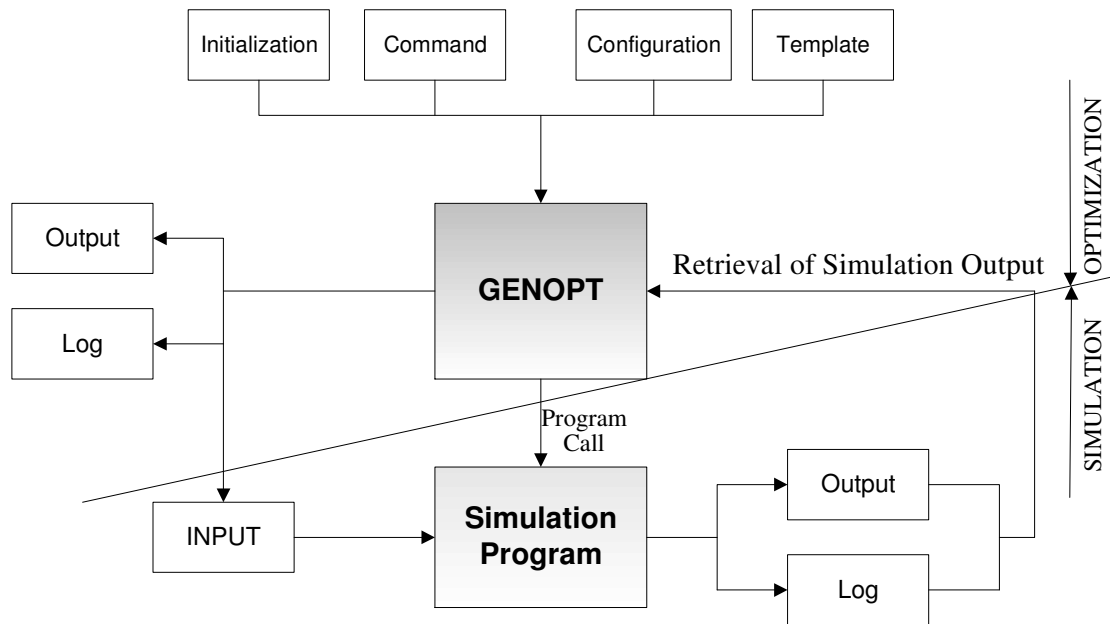


**Figure 5.1.** *GenOpt Interaction with Simulation Engine (GenOpt Documentation)*

As seen from the figure 5-1, GenOpt requires different files in order to perform the optimization. Details of the files used by GenOpt can be obtained from the program documentation. A brief summary of the files is given below.

- Initialization file: This file contains the path to all the files used by GenOpt (command file, input files, log files) and the path to the main simulation program. GenOpt recognizes the output variables from the output file with the help of delimiters, which are specified in the initialization file.

- Command File: This file defines all the optimization variables, their ranges and their initial values. The file also contains the name of the algorithm to be used in the optimization.

- Configuration file: This file is unique to the operating system and the simulation engine and does not need to be changed for different optimization problems. This file contains the path and command line invocation required to start the simulation. It also contains any error messages that would be obtained in the error file. This allows GenOpt to stop, if the simulation crashes.

## 5.3 Development of the Optimization Driver

In a typical optimization problem employing GenOpt, an input template file is a required file. This input template file is read by GenOpt and an input file is written with the changed values of optimization variables. GenOpt then calls the main simulation engine, which uses the new input file. The output of the simulation engine is also processed by GenOpt for simple cases. For the present work, GenOpt could not be coupled to EnergyPlus directly for the following reasons:

- When the length and mass flow rates of the GHE are changed, the g-functions also change. Therefore, for every simulation run, the g-functions must be recalculated with new parameters given by GenOpt. This is achieved by using the g-function calculation dll of GLHEPro, a professional GLHE sizing tool.

- The EnergyPlus input files differ significantly for different control and configuration strategies. Since the optimization involves control and configuration strategies, switching of input template files is necessary depending on the strategies requested by GenOpt.

- The objective function identified in the present work cannot be obtained directly from the EnergyPlus output files. The data obtained from the output files needs to be summed up for a whole year (in the case of energy usage and heat transfer) or has to be used in independent equations (in the case of the life cycle cost). Though GenOpt has some post processing capability, it was not capable of handling the life cycle cost function.

For these reasons, GenOpt was not directly coupled to EnergyPlus. Instead, GenOpt was configured to call an optimization driver developed for the present work. Fig 5-2 illustrates the main functions of the optimization driver. As shown in the figure, the driver reads the input file created by GenOpt in the first step and gets the values for the GLHE parameters. Once the GLHE parameters are obtained the g-func calculator program (Khan, 2004), which calculates the g-functions is called. The g-func calculator writes the new g-functions along with the other parameters in the EnergyPlus GLHE object format. From the input file created by GenOpt, new control and configuration strategies are obtained and the corresponding input template file is selected. The next task

of the optimization driver is to rewrite the input file with all new parameters. In order to rewrite the input file completely, all circulating pumps need to be resized for new mass flow rates. Once the input file is written, the EnergyPlus executable is called to perform the simulation. After the simulation runs, the optimization driver post-processes the eso file (output file for EnergyPlus), as required by the optimization.

The optimization driver interacts with a number of different files in order to carry out the functions explained earlier. Fig 5-3 shows how different files are used by the optimization driver and their interaction with the driver. As evident from the figure, the optimization driver has only one input file, to interact with GenOpt. Similarly, only one output file created which will be read by GenOpt. All other files are intermediate only and help the driver for its internal function calls and subroutine calls.
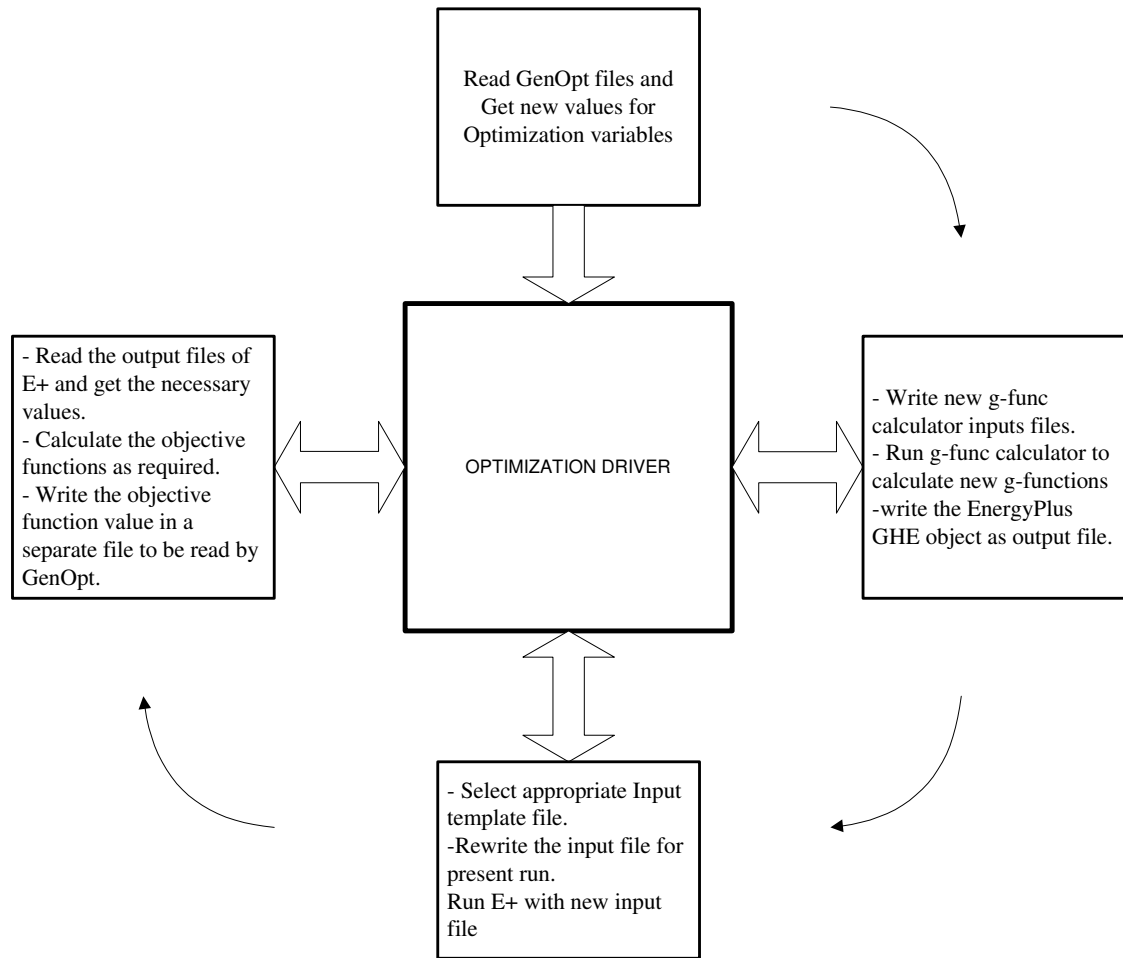
**Figure 5.2.** *Functions of Optimization driver*
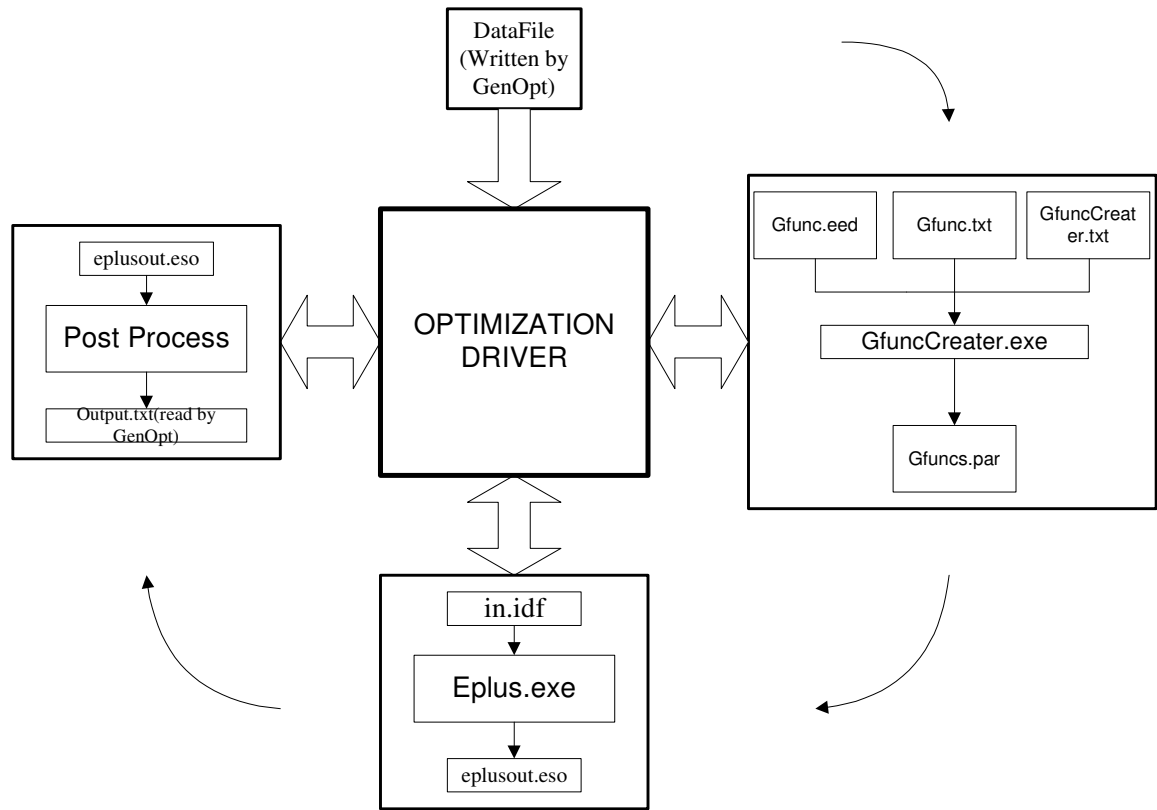
**Figure 5.3.** *Interaction between Optimization driver and files*

Since the EnergyPlus input file varies considerably for each configuration and control strategy, a template file was written for each combination of control and configuration strategy. Table 5-1 shows all the different possibilities and their corresponding input template files. Four different combinations of control and configuration strategies are identified.

**Table 5-1.** *Possible input Template files for Optimization driver*

| File Type | Configuration Type | Control Strategy |
|---|---|---|
| File Type 1 | GHE and Tower in Series | Outdoor Wet bulb temperature difference control |
| File Type 2 | GHE and Tower in Series | Reference Node set point control |
| File Type 3 | GHE and Tower in Parallel | Outdoor Wet bulb temperature difference control |
| File Type 4 | GHE and Tower in Parallel | Reference Node set point control |

A non-dimensional model to size the circulating pump is developed and incorporated in the optimization driver. Every time the flow is changed, the optimization driver resizes the pump and calculates the new power usage and pump head. These values are used to write the new pump object in EnergyPlus input file. Table 5-2 gives the number of pumps in the condenser loop and their flow rates during the optimization for both configurations.

**Table 5-2.** *Flow rate and number of pumps for different HGSHP configurations*

| HGSHP Configuration | Number of pumps (1 pump per Condenser Loop) | Design Pump flow rate |
|---|---|---|
| GHE and Tower in Series | 2 | $\dot{V}_{GHE}$ or $\dot{V}_{Tower}$ |
| GHE and Tower in Parallel | 1 | $\dot{V}_{GHE} + \dot{V}_{Tower}$ |

The head against which the pump has to operate is calculated using the system piping and fitting data for the system. A detailed explanation of the HGSHP piping and pressure drop calculations is given by Hern (2004). A Bell & Gosset series 80 pump is selected for both the primary GHE loop and the tower loop in the system. The mass flow rate for the system is set between 0.5 kg/s and 1 kg/s. A non-dimensional model is used to describe the selected pump based on the manufacturer's performance curve. Non-dimensional mass flow rate, non-dimensional pressure and efficiency are calculated using the following equations:

$$\varphi = \frac{\dot{m}}{\rho N d^3} \qquad (5\text{-}1)$$

$$\psi = f(\varphi)$$
$$\eta = g(\varphi) \qquad (5\text{-}2)$$

Where, $\varphi$ = Non-dimensional mass flow rate (-)

$\dot{m}$ = Mass flow rate (kg/s)

$\psi$ = Non-dimensional head (-)

$\eta$ = Efficiency (-)

$\rho$ = Density of fluid (kg/m$^3$)

$N$ = Speed pf the pump motor (revs/s)

$d$ = Diameter of pump impeller (m)

$f,g$ = functions defined between $\varphi,\psi,\eta$

Once we calculate the non-dimensional mass flow rate, non-dimensional pressure head and efficiency, then the actual pump operating pressure and power consumption can be calculated as follows:

$$\Delta P = \psi \rho N^2 d^2 \qquad (5\text{-}3)$$

$$P = \frac{\dot{m}\Delta P}{\rho.\eta} \qquad\qquad (5\text{-}4)$$

Where,                    $\Delta P$ = Pressure head against which pump operates.

$P$ = Power consumption

Figures 5-4 shows the curve used to fit the non-dimensional mass flow rate and efficiency data. The curve shown in figure 5-5 fits the non-dimensional mass flow rate and non-dimensional head. The theoretical and actual power usage for different mass flow rates is plotted in figure 5-6.

**Dimensionless Mass flow rate Vs Efficiency**

$$y = -1724.4x^3 - 303.6x^2 + 24.604x + 0.0553$$
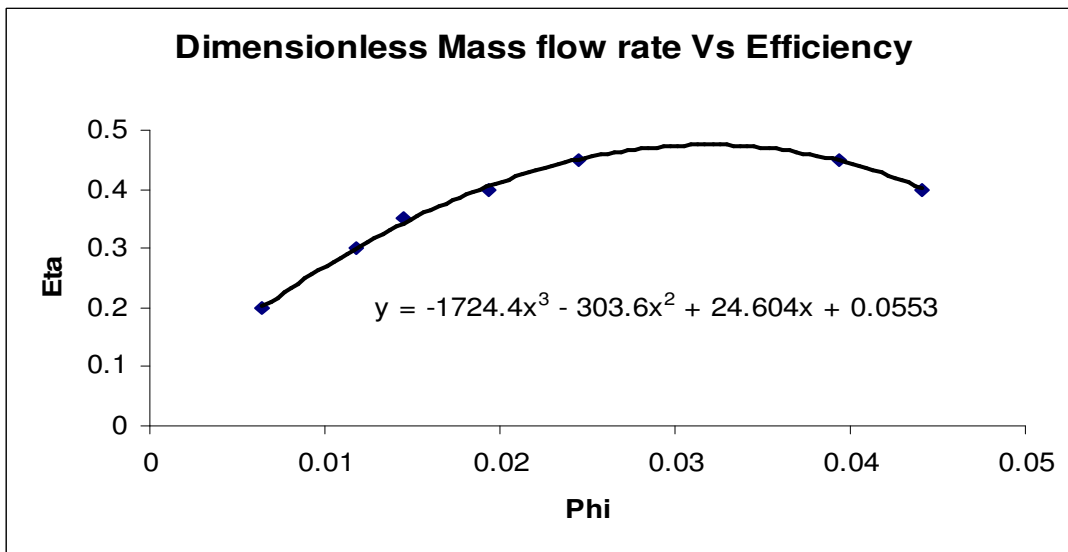
**Figure 5.4.** *Phi Vs Eta for Bell and Gosset Series 80 pump*
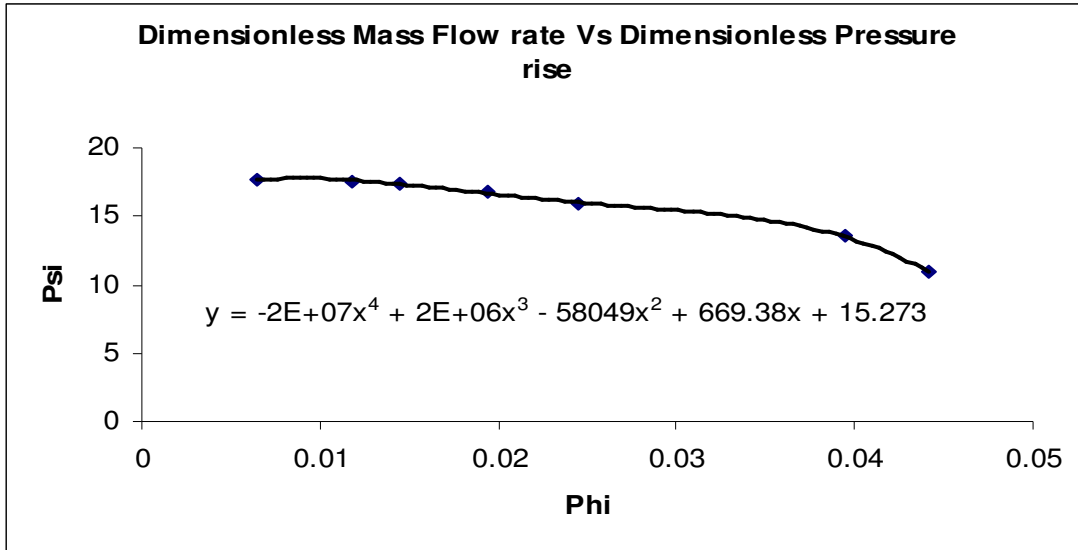
**Figure 5.5.** *Phi Vs Psi for Bell and Gosset Series 80 pump*
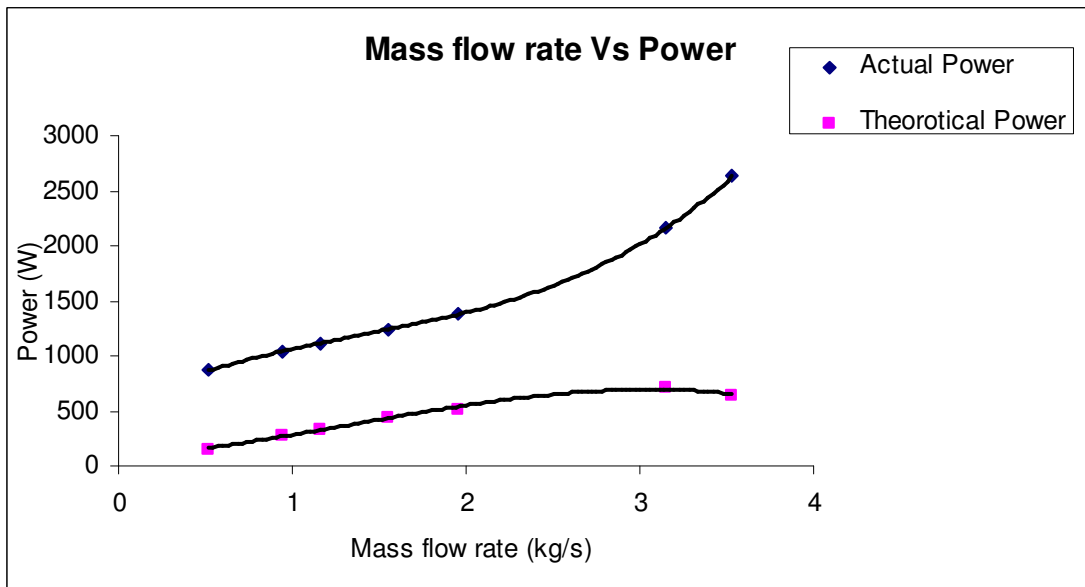


**Figure 5.6.** *Actual and Theoretical power for circulating pump*

## 5.4 Optimization of HGSHP systems

The primary requirement, which forces the designer to use a hybrid ground source heat pump system, is the imbalance in the heat extracted and heat rejected by the ground loop. The initial cost of the system can be greatly reduced by the decreased loop length for the HGSHP system. On the other hand, a supplemental heat rejecter such as a cooling

tower may result in higher operating costs. In configurations where the ground heat exchanger and the supplemental heat rejecter are in separate loops, there is a need for two circulating pumps, which also add to the operating cost. The increase in operating cost can easily offset the initial cost savings in the long term if the system is not properly designed. This reinforces the need for optimization of such systems.

Three important objectives in optimizing the design of HGSHP system are:

- Load Balancing: A HGSHP system is used mainly because of the imbalance in heating and cooling loads on the ground loop. Thus, it is required that an attempt be made to balance the load in any system design optimization.

- Energy Consumption: Energy standards or other factors may require energy consumption be minimized.

- Life Cycle Cost: As explained earlier, the operating cost of a poorly designed HGSHP system may offset the savings in initial cost. Thus, one of the major objectives in an optimal design is to minimize the life cycle cost of the system.

Once the objectives for optimization are identified, the optimization variables need to be chosen. A HGSHP system has a number of design variables that affect the performance of the system. The following design variables, which are assumed to be discrete, are used as parameters:

- Mass flow rates: The GHE mass flow rate and tower mass flow rate are the two flow rate variables that affect the system performance. The GHE flow rate is set as a variable for optimization. The tower mass flow rate is set to autosize in EnergyPlus. This allows the program to calculate the flow rate depending

on the capacity. A rule of thumb of 3 gpm/ton is used to calculate the tower flow rate in the autosize algorithm.

- GHE Loop Length: A hybrid system is implemented in the place of a conventional GSHP system in order to reduce the ground loop length and thereby reduce the first cost. Thus, the GHE loop length is a very important variable in any optimization related to HGSHP systems.

- Tower Capacity: The initial cost of the tower depends on the tower capacity. The capacity should be carefully selected such that there is no excess capacity.

- HGSHP configuration: As explained in chapter 4, a HGSHP system can be configured in two different ways: GHE and tower in parallel, GHE and tower in series. Both configurations might have different performance, which needs to be analyzed.

- HGSHP Control Strategies: Two supervisory strategies are evaluated: node setpoint control and delta temperature based control.

  ➢ Node set point control turns the cooling tower ON/OFF based on the heat pump EFT. If heat pump EFT is higher than 24$^{o}$C, cooling tower is turned ON, otherwise the cooling tower is shut down

  ➢ Delta temperature based control turns the cooling tower ON when the difference between the heat pump EFT and outdoor wetbulb temperature is greater than 2$^{o}$C. If the difference is less than 2$^{o}$C, the cooling tower is shut down.

The objectives and design variables used in HGSHP optimization are summarized in table 5-3.

**Table 5-3.** *Examples for Objectives and design variables for Optimization problem*

| OBJECTIVE FUNCTIONS | • Total energy consumption<br>• Load balancing on GHE<br>• Life cycle cost |
|---|---|
| DESIGN PARAMETERS | • GHE loop length<br>• GHE mass flow rate<br>• Tower capacity<br>• System configuration<br>• System control strategy |

## *5.5 Optimization Methodology*

The optimization of HGSHP systems can get quite complex, since there are a large number of independent parameters involved in the operation of the system. The variables can range from mass flow rates to pipe specifications, from ground properties to supplemental heat rejecter specifications. Even though it is very difficult to know the exact value of each parameter used in the simulation, we can obtain knowledge of the acceptable range for these parameters. For a HGSHP system, we cannot select a single objective that will satisfy all requirements. Although GenOpt contains a library of optimization algorithms, none of them is well suited to handle multi-objective optimization. In most of the algorithms, an equivalent objective function should be developed which is a linear combination of the separate objectives. For an Optimization problem with *n* separate objective functions, the single equivalent objective function is given in formula 5-5:

$$O_{eq} = \lambda_1 O_1 + \lambda_2 O_2 + \dots\dots + \lambda_n O_n \qquad (5\text{-}5)$$

Where $O_{eq}$ = Equivalent Objective function value

$O_1, O_2, .., O_n$ = Individual objective function values

$\lambda_1, \lambda_2, .., \lambda_n$ = Scalar multiplier values.

This linear combination approach is useful when the individual objective functions behave in a similar way throughout the domain. For problems where the behavior of the system is predictable, this linear combination approach works fine. As discussed earlier, thermal systems problems are quite complex and their behavior is difficult to predict with reasonable accuracy. This uncertainty forces the designer to optimize the system separately for each objective function and then use heuristic arguments to select the final set of system parameters. The overall optimization of HGSHP systems explained in this chapter is based on separate evaluation of the three objective functions shown in table 5-3.

In obtaining a more meaningful optimization, penalty functions have to be implemented, which restrict impractical results. In a HGSHP system, in order to reduce the system life lycle lost, the optimization algorithm would tend to reduce the GLHE length and Tower capacity. However, as we go on decreasing the GLHE length and Tower capacity, the potential of the system to meet loads decreases. At some point, the system will fail to meet the loads. In order to avoid this problem, an unmet load penalty was implemented in the program. This penalty function affects the objective function if the system is not able to meet loads. The function was selected such that its implementation would not send the optimization algorithm in a reverse direction. Care was taken to assure that the penalty function provides only a slight slope to the domain and not a barrier. The objective function was calculated as follows:

$$f_{obj,f} = f_{obj,i} + f_{Pen} \qquad (5\text{-}6)$$

$$f_{Pen} = 20 * \Delta Q \qquad (5\text{-}7)$$

Where,

$f_{obj,f}$ = Objective function after penalty function implementation

$f_{obj,I}$ = Objective function before penalty function implementation

$f_{Pen}$ = Penalty function

$\Delta Q$ = Unmet loads (W)

The optimized system for both the load balancing and the energy consumption objective functions was one of the limiting cases. When load balancing was used as the objective function, the optimization resulted in the shortest possible ground length and the largest possible supplemental heat rejecter capacity. Similarly, when energy consumption was used as the objective function, the optimization resulted in a smaller tower and longer loops lengths. Longer loop lengths kept the loop temperatures low thereby resulting in less heat pump power consumption. The smaller supplemental heat rejecter capacity resulted in a smaller tower water flow rate, thereby reducing the size of the tower circulating pump. Both of the above objective functions drove the optimization to the limiting values of the design variables with the only constraint being the need to meet the load.

Thus, in order to obtain a meaningful result, the life cycle cost of the system is selected as optimization objective. The following section describes how the objective function value was calculated.

### 5.5.1 Objective Function

The life cycle cost of the system deals with the economic aspects of the design, while the other two objectives deal with the performance of the system. The cost of the ground loop is taken as $20 per meter of loop. The tower installation cost is calculated as

$99.5/KW of tower capacity. The operating cost is calculated by using the energy rate of $0.075/KWh. Initial cost, operating cost and life cycle cost for the system is calculated as:

$$C_{init} = L_{GHE}C_{GHE} + Q_{Tow}C_{Tow} \qquad (5\text{-}8)$$

Where:        $C_{init}$ = Initial cost of the system ($)

$L_{GHE}$ = Length of GHE Loop (m)

$C_{GHE}$ = Cost of GHE loop ($/m)

$Q_{Tow}$ = Tower Capacity (KW)

$C_{Tow}$ = Cost of tower ($/KW)

The annual operating cost for the system is obtained from the total energy consumption of the system as:

$$C_{Oper} = E_{tot}.C_{Energy} \qquad (5\text{-}9)$$

Where:        $C_{Oper}$ = Annual operating cost of the system ($)

$E_{tot}$ = Total annual energy consumption (KWh)

$C_{Energy}$ = Cost of energy ($/KWh)

Once we obtain the initial cost and operating cost, the life cycle cost for the system is calculated. In order to calculate the life cycle cost, the present worth factor ($\phi_{PWF}$) is used in conjunction with the operating costs (Stoecker, 1989). As will be discussed later, the annual operating cost considered is the 20th year operating costs. For a hybrid system, the 20th year cost is either the best or worst during the life of the system. Now the assumption that the operating cost for each year is same as the 20th year operating cost, we can project each years cost to the present value. An interest rate of 6% is used in this work. With all the above information, the life cycle cost is calculated as:

$$C_{LC} = C_{init} + \phi_{PWF} C_{Oper} \qquad\qquad (5\text{-}10)$$

$$\phi_{PWF} = \frac{(1+\alpha)^n - 1}{\alpha(1+\alpha)^n} \qquad\qquad (5\text{-}11)$$

Where $\qquad\qquad C_{LC}$ = Life cycle cost ($)

$C_{init}$ = Initial cost of the system ($)

$C_{oper}$ = 20$^{th}$ year operating cost ($)

$\phi_{PWF}$ = Present worth factor

$\alpha$ = Interest rate

$n$ = Number of years for pay back

For a HGSHP system, if the loads are highly unbalanced, the operating costs will not remain same during every year of operation. In a system of this type, heat pump COP for cooling will decrease as year each passes. The operating cost of the system on the 20$^{th}$ year will be more than that for first year. Similarly, for some other systems depending on ground loop size and tower capacity, the operating costs could come down over 20 years compared to first year. Thus in order to obtain an accurate optimization, we need a method to calculate the operating cost for every year of 20 year simulation. This was not possible within the scope of this study due to the level of effort required to modify the EnergyPlus reporting routines. However, a method was developed to calculate the operating cost of the 20th year accurately as explained in the following section. By comparing the life cycle cost based on 1st year operating costs with the life cycle cost based on 20th year operating costs, the maximum error associated with a one-year optimization could be evaluated.

## 5.5.2 Long Time Step Simulation

As previously explained, the HGSHP optimization is meaningful only if the simulation is run for 20 years (typical lifetime of a HGSHP system). Though computationally expensive, it is practical to run a single 20-year simulation for analysis purposes. However, in the case of optimization, where the simulation program (EnergyPlus in this case) has to be run a number of times, (typically on the order of 600 to 1000 depending on the system complexity) it is prohibitively expensive to run a 20-year simulation. A typical yearly simulation with an HGSHP system takes around 7 minutes on a Pentium 2.6 GHz machine. A twenty-year simulation would then take approximately 140 minutes and a complete optimization run would take around 50 days to complete (assuming the optimization requires 500 simulations).

In order to make the optimization practical, the simulation time for 20 years has to be reduced considerably. In a system simulation with a GLHE, the most time consuming part of the simulation is the GLHE model. Any attempt to decrease the simulation time requires reducing the GLHE simulation time. Khan (2004) recommended aggregating the loads over 19 years and then using them for simulation in the 20th year. The present work is built upon this recommendation. In order to reduce simulation time, it was necessary that the simulation time step be increased from the present value of one hour to at least a day. However, in order to perform energy analyses on the system or a life cycle cost study, it was necessary that an hourly simulation be performed. Reduction of simulation time was achieved by running a daily simulation until the start of the 20th year and then an hourly simulation for the 20th year. This approach required a significant reengineering

of the EnergyPlus code to handle daily time steps. The following steps were taken to equip EnergyPlus to handle daily time steps:

- Total building loads were averaged for each day, and it was assumed this average load was in effect for each hour of the day.

- The weather manager was modified to calculate the average wet bulb temperature for the day (since this is used in the tower calculations).

- The GLHE model was changed so that no sub hourly calculations were done until the end of the 19th year. From the 20th year, the simulation is set in order to do the regular calculations at hourly/sub hourly time steps. The loads are aggregated after each day (24 hrs). No changes to monthly aggregation were made.

- The cooling tower was run for every hour with the daily average inlet temperature and the corresponding hourly wetbulb temperature. The output from the tower for each hour was then averaged over a day to obtain the average daily output temperature.

The power consumption and heat transfer rate is calculated as the average of the corresponding values over the hours the tower was operating each day. This was equivalent to defining a runtime fraction for the cooling tower. This approach let us predict whether the tower would run more in the 20th year or not and approximately predict the change in tower performance. The plot for the average inlet temperature for tower and daily wetbulb temperature is given in figure 5-7. The figure shows when the tower would be considered ON/OFF, using this approximation.
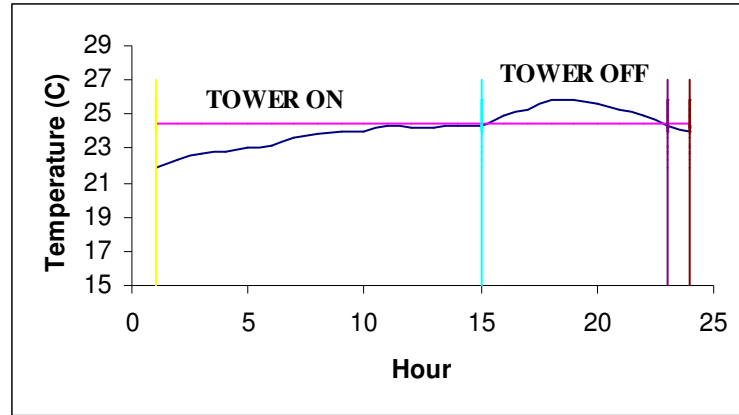
**Figure 5.7.** *Typical Tower Daily Average Inlet and Wetbulb Temperatures*

The above method reduced the computation time for a 20-year simulation by 95%. With the modifications, in effect a single year simulation took just as much time as a 20-year simulation.

The daily time step version was compared with the hourly time step version of the program to check the error resulting from the simplifications. The OSU system was simulated for 20 year using the detailed algorithm and using the simplified procedure. The plots for GLHE outlet temperature for the 20$^{th}$ year in both cases are shown in figure 5-8. The break down of system energy consumption in the 20$^{th}$ year of operation is shown in table 5-4.

In order to investigate the source of error in the tower fan energy consumption, a system with a GLHE alone was simulated using the two versions. From the runs, it was seen that the detailed version resulted in a slightly higher loop temperature (around 0.2$^0$C). This temperature rise did not affect the heat pump performance, which was almost same for both cases. However, the increase in loop temperature resulted in an increased operation of tower (due to the differential temperature control). Because of this tower fan power consumption was 6.7% higher for detailed model.

Instead of doing sub hourly calculations for the whole simulation period and accumulating the loads as monthly loads, the modified approach aggregates the loads daily and then at the end of each month. Thus, at the beginning of the 20$^{th}$ year all previous loads are saved as monthly loads. Moreover, the effect of loads older than a few months is small. The modification results in a slight under prediction of the cooling tower energy consumption. This is due to the assumption of an average daily inlet temperature. The error however is reduced considerably by introducing a runtime fraction for the tower. This is calculated by simulating the tower for all 24 hours separately using hourly wetbulb temperatures. Thus, the tower is not run for the whole day, which would result in gross over prediction of the energy consumption.
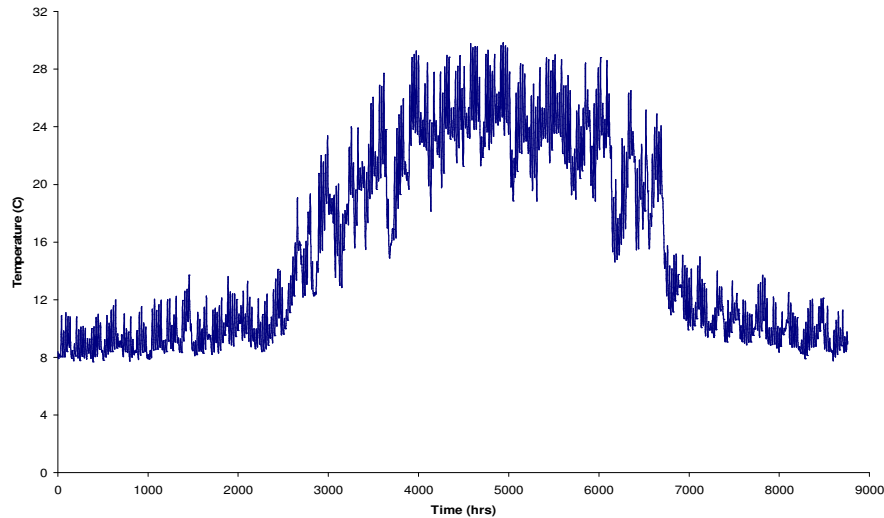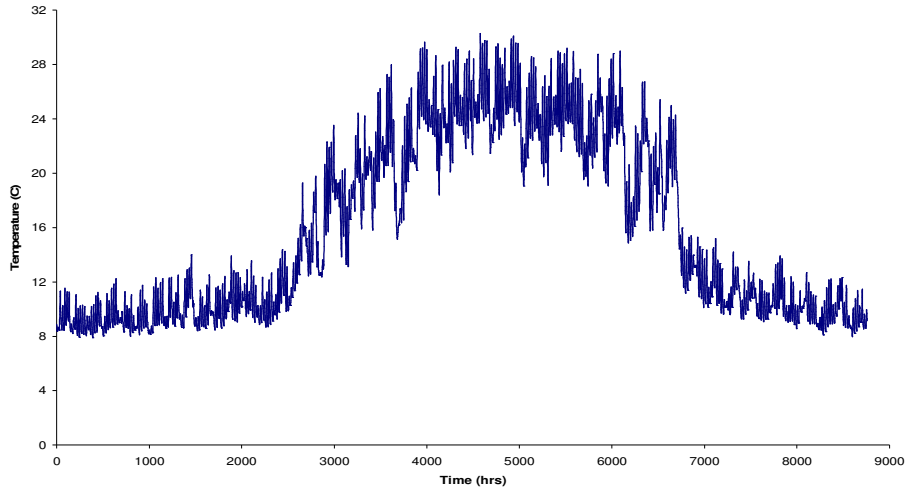
**Figure 5.8.** *GHE outlet temperature for Modified (above) and Detailed(below) Case*

**Table 5-4.** *20$^{th}$ year Energy Consumption for modified and detailed methods*

|  | **Heat Pump (Cooling), Kwh** | **Heat Pump (Heating), Kwh** | **Cooling tower Fan, Kwh** |
|---|---|---|---|
| Modified Method | 3609 | 1122 | 821 |
| Detailed Method | 3611 | 1119 | 880 |

## *5.6 Results and Discussions*

In order to simulate the building with a highly cooling dominated load profile a space heater was run in the room from April to September. The cooling loads were boosted so that the GLHE was unable to meet the loads without a significant increase in the loop temperature. The load profile used for optimization is shown in figure 5-9 (cooling loads in the negative y-axis).
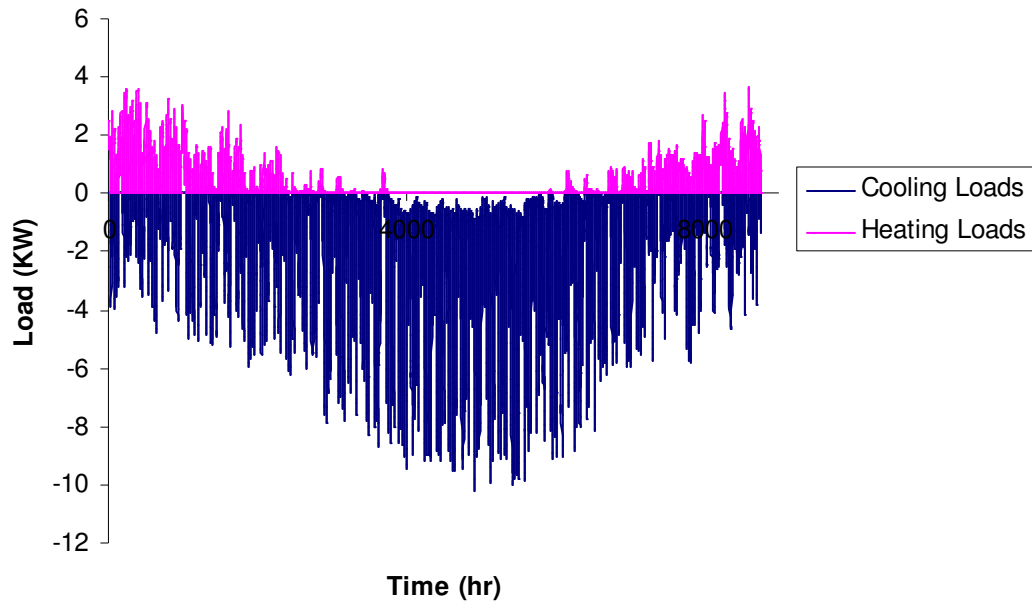
**Figure 5.9.** *Building Loads for Optimization Runs*

A Particle Swarm Optimization with 'VonNeumann' neighborhood topology was selected for the simulation. This is because 'VonNeumann' neighborhood topology performs better for discrete variables. The optimization settings used during the work are tabulated in table 5-5.

All the optimization variables are assumed discrete with the lower bound, upper bound and step for each variable set in the command file. The lower bound for the GLHE is selected such that the heating loads are met by the ground loop. Table 5-6 shows the lower bound, upper bound and step for each optimization variable.

**Table 5-5 *GenOpt Optimization Settings***

| Algorithm | Particle Swarm Optimization |
|---|---|
| Neighborhood topology | VonNeumann |
| Neighborhood size | 15 |
| Number of particles | 12 |
| Number of generations | 500 |
| Max velocity discrete | 4 |
| Acceleration — Social | 1.2 |
| Acceleration — Cognitive | 2.8 |
| Inertia weight — Initial | 1.2 |
| Inertia weight — Final | 0.1 |

**Table 5-6 *Lower bound, Upper bound and step for design variables***

| Design Variable | Lower Bound | Upper Bound | Step |
|---|---|---|---|
| GHE length (m) | 40 | 120 | 2 |
| Tower Capacity(W) | 4000 | 10,000 | 250 |
| GHE Flow Rate (Kg/s) | 0.5 | 1 | 0. 1 |

The optimization run took 74 hours on a Pentium 2.6 GHz machine running on Microsoft Windows XP®. Table 5-7 shows the comparison between the base case and the results obtained from the optimization run. Since the GLHE length was a major contributor to the initial cost, it was logical that the optimization would tend to move towards the lower GLHE length. However, the GLHE cannot reach the lower bound

since at some reduced GLHE length, the system would fail to meet the loads. This situation would result in implementation of penalty function on life cycle cost. Thus, the optimal system will have the minimum possible ground loop that would prevent the system from failing to meet the loads. Comparison of different cost components involved in the life cycle cost calculation is shown in Figure 5-10. The figure shows the comparison between the base case and the optimal case obtained using $20^{th}$ year operating costs. Figure 5-11 shows how total energy is split between different components for the optimal case. It can be seen that the Heat Pump is the major consumer of energy in the optimal case. The optimal configuration was found to be a series configuration. This can be explained by the fact that for the parallel configuration, constant speed pump power consumption was very high because the pump was sized for the combined tower and GLHE flow. For the series configuration, the flow is split between primary and secondary loop and as a result, the tower loop pump can be shut down while the tower is not working.

In order to meet the loads on the building the tower can be operated on either a set point control based on the heat pump EFT or the differential control based on the difference between heat pump EFT and outdoor wetbulb temperature. Use of set point control strategy would require a larger tower with less operating hours, while the differential control requires a smaller tower running for more hours. The factor that determines the efficient control is the set point used in the set point control strategy. In the present work, a set point of $24^{o}C$ is forced on heat pump EFT. The low set point did operate the tower for more number of hours and thus meet the loads. Since the circulating pump power consumption is the major factor towards the operating costs, a differential

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control would result in high operating costs. It has to be noted that the decision of temperature set point affects the decision of efficient control strategy. A high set point would results in less number of hours of cooling tower operation when controlled based on heat pump EFT, thus leading to high fluid temperatures. In such a case, a differential control would be a better choice. It is therefore recommended that the set point temperatures be included as optimization variables in the GenOpt/EnergyPlus environment.

**Table 5-7** *Comparison of Base Case and Optimal Case*

|  | **BASE CASE** | **OPTIMAL CASE (1 yr)** | **OPTIMAL CASE (20 Yrs)** |
|---|---|---|---|
| GHE loop length (m) | 128 (64x2) | 108 (54x2) | 108 (54x2) |
| Tower capacity (W) | 5000 | 6500 | 6000 |
| GHE flow rate (Kg/s) | 0.5 | 0.5 | 0.5 |
| Configuration strategy | Series configuration | Series configuration | Series configuration |
| Control strategy | Outdoor wetbulb delta temperature based | Node set point based | Node set point based |
| GLHE cost ($) | 2560 | 2160 | 2160 |
| Tower cost ($) | 498 | 647 | 597 |
| Heat pump energy Cost ($) | 369 | 379 | 376 |
| Tower circulating pump energy cost ($) | 256 | 227 | 209 |
| Tower fan energy consumption ($) | 78 | 65 | 55 |
| Energy cost ($) | 703 | 671 | 640 |
| Net present value of energy ($) | 8157 | 7797 | 7437 |
| Total life cycle cost ($) | 11215 | 10480 | 10194 |
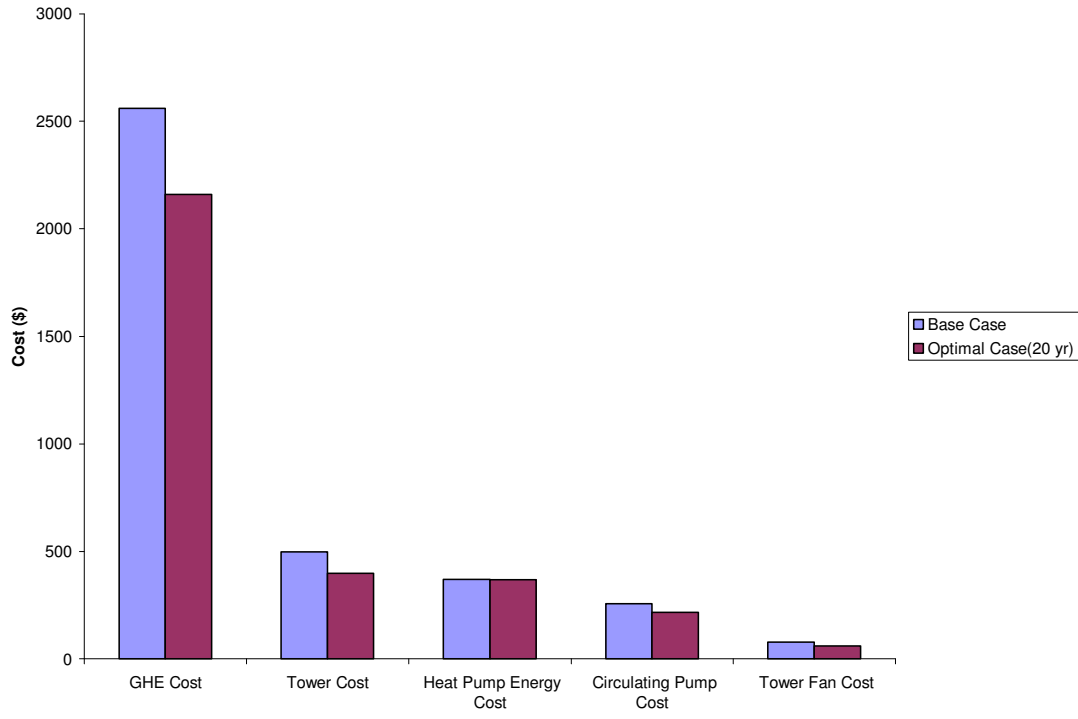
**Figure 5.10.** *Comparison of Cost Components for Base and Optimal Case*
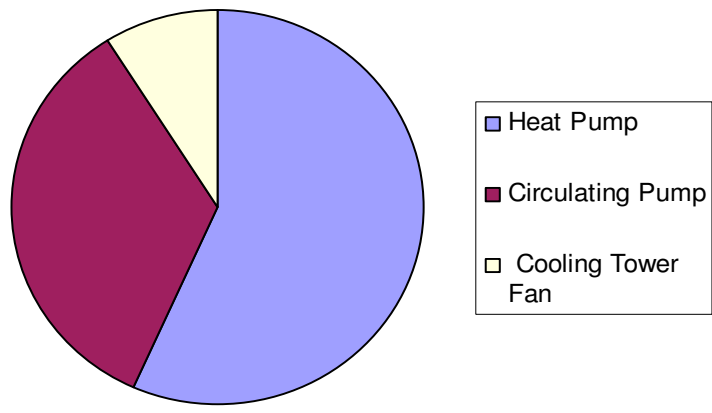


**Figure 5.11.** *Pie Chart showing different Energy Consumption for Optimal Case*

For the system under consideration, the energy consumption during the 20$^{th}$ year is less compared to the energy consumed in the first year. This is because of the positive

effect of the supplemental heat rejecter on the ground loop. The optimization with the first year costs used for the annual operating cost results in a bigger tower and a higher life cycle cost. Optimizing the system on the basis of $20^{th}$ year operating cost shows more savings. As previously noted that the optimization is system specific and shows additional savings over a single-year simulation. For system where the operation in the $20^{th}$ year is worse than the first year the situation would be reversed, yielding in an incorrect optimal point with a one-year run simulation rather than a 20-year simulation. In this case, the optimal system would be undersized compared to base case.

It is seen from the above table and plot that there is a 16% savings in GLHE first cost. The circulating pump energy consumption was also 8% less for the optimal case. This is due to the Node Set Point control strategy, which allows the tower to run only during midyear. Since the GLHE is the major contributor towards the cost, the optimization always tries to reach the minimum GLHE length. However, the heating loads on the system would limit the reduction of the GLHE length. Once the minimum possible GLHE length is obtained, the tower capacity is adjusted so that the cooling loads are also met. This can be seen from the results in the table, which shows a decreased GLHE length and increased tower capacity for the optimal case. For the twenty-year optimization, we obtain some reduction in the tower capacity. A savings of 6.5% is obtained in life cycle cost using the operating cost for the first year. For the particular system under consideration, using $20^{th}$ operating cost year resulted in a savings of 9.1% in life cycle cost compared to the base case. Since the operating costs are expected to increase linearly over the life of the project, the actual savings due to a 20-year simulation will be slightly lower. Although the difference in life cycle cost is relatively

small, it is large enough to suggest that additional investigation of the procedure is warranted.

# CHAPTER 6

## CONCLUSIONS AND RECOMMENDATIONS

The work presented in this thesis describes the modeling, verification and optimization of hybrid ground source heat pump systems in EnergyPlus. This chapter summarizes the present work, gives some conclusions drawn from the investigation and provides recommendations for future work.

## *6.1 Summary and Conclusions*

The work can be divided into three groups viz. modeling, verification and optimization. The following paragraphs summarize the results and discuss conclusions drawn from each group.

### *6.1.1 Modeling*

EnergyPlus was modified in order to simulate hybrid ground source heat pump systems. Three significant enhancements were developed and incorporated in EnergyPlus as follows:

- Multi-year simulation capabilities were added to EnergyPlus. Two types of weather files were implemented: a single year of data read recursively and a long weather file with multiple years of data. Multi-year simulations modeled the ground temperature increase for HGSHP systems accurately.

- Two new supervisory control strategies were added to EnergyPlus viz. node set point control and delta temperature based control. The new control schemes allowed efficient control of HGSHP systems.

- A plate heat exchanger model based on a cross flow heat exchanger was developed for use in EnergyPlus. The model with its associated control was implemented in EnergyPlus as a separate component on the condenser supply side. The connection and associated controls allowed better configuration of HGSHP systems with multiple condenser loops.

## 6.1.2 Verification

EnergyPlus HGSHP simulations were verified against published results from Yavuzturk's (2000) work. Three cases out of five cases considered by Yavuzturk were selected for verification. Simulation was performed for two different climate regions. The following results were obtained from the verification:

- Loop temperatures predicted by EnergyPlus matched the Yavuzturk predictions within $0.5^0$C for all cases.

- Heat pump power consumption agreed within an error of 5% for node set point control and within 2% for differential control. The difference in heat pump power consumption was due to differences in the models used by EnergyPlus and Yavuzturk.

- The cooling tower fan power consumption differed by 7% for node set point control and by 4% for differential control. Lack of dead band for control ranges in EnergyPlus under predicted the tower fan power consumption.

- The loop temperature predicted by both the models agreed within $0.5^0$C.

### *6.1.3 Optimization*

An optimization study aimed at obtaining optimal design parameters, control strategies and configurations for HGSHP systems using GenOpt was undertaken. A particle swarm optimization algorithm in a linked GenOpt/EnergyPlus program was used to carry out the optimization with life cycle cost as the objective function.

An optimization driver was developed to preprocess the input files, call EnergyPlus to perform the simulation and finally post-process the EnergyPlus output files to give the objective function value was developed. A non-dimensional model for the circulating pump was used in the driver program to resize the circulating pump during the optimization.

The GLHE model and the program structure of EnergyPlus were modified to handle daily time steps. The simplification is summarized by following points:

- The EnergyPlus structure was modified to run with daily time steps for all but the final year of simulation. For final year, the program reverts to hourly and sub-hourly time steps.

- The cooling tower model was modified to calculate a runtime fraction for the tower while running on daily time steps.

The modifications reduced the computational time for 20-year simulation by 95%. The heat pump power consumption by the detailed and the simplified versions were same. The cooling tower fan power predicted by both versions differed by approximately 7%. This was due to the slight difference in loop temperatures predicted by both versions.

The optimization was performed with a one-year simulation and with 20-year simulations. The following conclusions were drawn from the optimization:

- The optimization proceeded towards the lowest possible GLHE length since it was the major contributor towards cost.

- The one-year simulation resulted in a bigger tower, while the 20-year simulation resulted in a tower with 7% less capacity.

- The heat pump energy consumption was only reduced by 0.3% between the base case and the optimal case, whereas in tower fan power was reduced by 17%.

- The savings in life cycle cost using a one-year simulation was 6.5%, while using a 20-year simulation with $20^{th}$ year operating costs for the life of project resulted in a 9.1% savings on life cycle cost compared to base case.

## *6.2 Recommendations for Future Work*

Following recommendations are given for future work in HGSHP modeling and optimization:

- Validate the individual component models using experimental data.

- Extend control strategies to account for dead band in the operation.

- Develop new control strategies, which are more useful for HGSHP systems. One specific recommendation would be supervisory control schemes, which can change the strategies depending on the loop temperatures.

- Replace the counter flow heat exchanger model by an established plate heat exchanger model in literature.

- Improve the scope of optimization with the inclusion of more design variables such as grout conductivity, borehole spacing etc.

- Investigate the possibility of a universally applicable optimization algorithm, which is not system specific.

- Modify EnergyPlus to report the energy consumption during the mid years. This would provide an accurate estimate of annual operating costs for the life cycle cost calculation.

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# APPENDIX A

The idd objects added or modified during the present work are explained in this section.

```
RUNPERIOD,
  \min-fields 11
 N1, \field Begin Month
    \required-field
    \minimum 1
    \maximum 12
    \type integer
 N2, \field Begin Day of Month
    \required-field
    \minimum 1
    \maximum 31
    \type integer
 N3, \field End Month
    \required-field
    \minimum 1
    \maximum 12
    \type integer
 N4, \field End Day of Month
    \required-field
    \minimum 1
    \maximum 31
    \type integer
 A1,  \field Day Of Week For Start Day
    \note =<blank - use
  \Weatherfile>|Sunday|Monday|Tuesday|Wednesday|Thursday|Friday|Saturday];
    \default UseWeatherFile
    \type choice
    \key Sunday
    \key Monday
    \key Tuesday
    \key Wednesday
    \key Thursday
    \keyFriday
```

```
    \key Saturday
    \key UseWeatherFile
…. (Reduced for Brevity)
  N5; \field Number of simulation years
    \type integer
    \default 1
```

**Figure A.1.** *idd Object for RUNPERIOD*

*/field Number of Simulation Years*:

This numeric field was added to the RUNPERIOD object to accommodate the multi year

simulation. It is an integer field and is set optional. If user does not enter a value, a

default of one is taken and a single year simulation is carried out. The number of years

can also be more than one even if the Start and end dates in Runperiod object does not

make up a whole year. Such a situation would result in a design week simulation

```
NODE TEMPERATURE SET POINT RANGE BASED OPERATION,
  A1, \field Name
     \required-field
     \reference ControlSchemeList
  A2, \field Reference Temperature Node Name
    \required-field
    \type alpha
  N1, \field Node Setpoint Range Lower Limit 1
     \required-field
    \type real
    \units c
    \minimum -100.0
    \maximum 100.0
  N2, \field Node Setpoint Range Upper Limit 1
     \required-field
    \type real
    \units c
    \minimum -100.0
    \maximum 100.0
  A3, \field Priority Control Equip List Name 1
     \required-field
     \type object-list
```

```
    \object-list CondenserEquipmentLists
--reduced for brevity--
  A12; \field Priority Control Equip List Name 10
    \type object-list
    \object-list CondenserEquipmentLists
```

**Figure A.2.** *idd specification for Node Set Point Control Scheme*

```
OUTDOOR WETBULB TEMPERATURE DIFFERENCE BASED OPERATION,
  A1, \field Name
    \required-field
    \reference ControlSchemeList
  A2, \field Reference Temperature Node Name
    \required-field
    \type alpha
    \units C
  N1,\field Wetbulb temperature difference Range Lower Limit 1
    \type real
    \units C
    \minimum -50.0
    \maximum 100.0
  N2, \field Wetbulb temperature difference Range Upper Limit 1
    \type real
    \units C
    \minimum -50.0
    \maximum 100.0
  A3, \field Switch Type
    \type choice
    \key ON
    \key OFF
    \default ON
  A4, \field Priority Control Equip List Name 1
    \required-field
    \type object-list
    \object-list CondenserEquipmentLists
--reduced for brevity--
  N19, \field Wetbulb temperature difference Range Lower Limit 10
    \type real
    \units C
  N20, \field Wetbulb temperature difference Range Upper Limit 10
    \type real
    \units C
  A21, \field Switch Type
    \type choice
    \key ON
```

```
    \key OFF
    \default ON
A22; \field Priority Control Equip List Name 10
    \type object-list
    \object-list CondenserEquipmentLists
```

**Figure A.3.** *idd object for Outdoor Delta Temp based Control*

```
HEAT EXCHANGER: PLATE: FREE COOLING,
    \memo A fluid/fluid heat exchanger designed to couple one condenser loop to
another condenser loop.
    \memo May be set as an ideal heat exchanger to emulate complete coupling of the
loops.
    \memo May also be used as a 'real' heat exchanger using an UA-Effectiveness
model.
    \memo For controlled free cooling a component is series (e.g. chiller) can be
switched on/off.
  A1, \field Name of free cooling heat exchanger
    \type alpha
    \required-field
  A2, \field Component name
    \type alpha
    \required-field
  A3, \field Component type
    \type alpha
    \required-field
  A4, \field Demand side fluid inlet node
    \type alpha
    \required-field
  A5, \field Demand side fluid outlet node
    \type alpha
    \required-field
  A6, \field Supply side fluid inlet node
    \type alpha
    \required-field
  A7, \field Supply side fluid outlet node
    \type alpha
    \required-field
  A8,  \field Demand side loop fluid name (water, ethylene glycol, etc.)
    \type object-list
    \object-list GlycolConcentrations
    \required-field
    \default water
  A9, \field supply side loop fluid name (water, ethylene glycol, etc.)
```

```
        \type object-list
        \object-list GlycolConcentrations
        \required-field
        \default water
    A10, \field Heat exchange mode
        \type choice
        \required-field
        \key IDEAL
        \key UA-EFFECTIVENESS
        \default IDEAL
    N1, \field UA
        \type real
        \minimum> 0.0
    N2, \field Demand side flow rate
        \type real
        \minimum> 0.0
        \units m3/s
        \ip-units gal/min
    N3; \field Supply side flow rate
        \type real
        \minimum> 0.0
        \units m3/s
        \ip-units gal/min
```

**Figure A.4.** *idd specification for Plate Heat Exchanger Object*

*Field: Name of Plate Cooling Heat Exchanger*

This alpha field contains the unique identifying name of this component.

*Field: Component name*

This alpha field is the name of the component, which allows the plate heat exchanger to

decide whether the loops have to be coupled or not. If this component is ON, the loops

are coupled else, the loops are uncoupled.

*Field: Component type*

This alpha field tells the type of the component with the name given in the previous field.

*Field: Demand Side Fluid Inlet Node*

This alpha field contains the name of the inlet node on the condenser demand side connected to the heat exchanger.

***Field: Demand Side Fluid Outlet Node***

This alpha field contains the name of the outlet node on the condenser demand side connected to the heat exchanger.

***Field: Supply Side Fluid Inlet Node***

This alpha field contains the name of the inlet node on the Condenser supply side connected to the heat exchanger.

***Field: Supply Side Fluid Outlet Node***

This alpha field contains the name of the outlet node on the Condenser supply side connected to the heat exchanger.

***Field: Demand and Supply Side Fluid Name (Water, Ethylene Glycol, Etc.)***

This alpha field contains the name of the fluid used in the Demand/Supply side of heat exchanger. The default is WATER. This fluid name must be the same as defined in the loop specification and its data must be available from the input file.

***Field: Heat Exchange Mode***

The heat exchanger can operate as either an ideal heat exchanger (the max possible energy is transferred across the loops) or as specified by the UA and effectiveness. The options are correspondingly 'IDEAL' and 'UA-EFFECTIVENESS'. Effectiveness is calculated assuming counter-flow. The default is 'IDEAL'.

***Field: UA***

This numerical field is used to specify the overall UA for use in the calculation of actual heat transfer when in 'UA-EFFECTIVENESS' mode.

*Field: Demand/Supply Side Flow Rate*

This numerical field is used to specify the design flow rate of this component on the

Demand/Supply side. This would normally be the same as the loop/cooling tower flow

rate.

# VITA

Sankaranarayanan Padhmanabhan

Candidate for the Degree of

Master of Science

Thesis: MODELING, VERIFICATION AND OPTIMIZATION OF HYBRID GROUND SOURCE HEAT PUMP SYSTEMS IN ENERGYPLUS

Major Field: Mechanical Engineering

Biographical:

Education: Received Bachelor in Engineering from Regional Engineering College, Durgapur, India in May 2001. Completed the requirements for the Master of Science degree with a major in Mechanical Engineering at Oklahoma State University in May 2005.

Experience: Employed as Research Associate by Department of Mechanical and Aerospace Engineering at Oklahoma State University, Oklahoma. March 2005 to present. Employed as Graduate Research Assistant by Mechanical and Aerospace Engineering Department at Oklahoma State University. Jan 2003 to Dec 2004. Employed as Junior Engineer at Fluid Control Research Institute, Palakkad, India. June 2001 to July 2002.

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# ABSTRACT

Name: Sankaranarayanan         Date of Degree : May, 2005
       Padhmanabhan
Institution: Oklahoma State University    Location: Stillwater, Oklahoma

Title of Study:    MODELING,VERIFICATION AND OPTIMIZATION OF HYBRID GROUND SOURCE HEAT PUMP SYSTEMS IN ENERGYPLUS

Pages in Study: 116          Candidate for the Degree of Master of Science

Major: Mechanical Engineering

Scope and Method of Study: Hybrid Ground Source Heat Pump system simulation capability was added to EnergyPlus. Program was modified to handle multi-year simulation capability. New supervisory controls were added and a plate heat exchanger model developed to configure realistic systems. The models were verified by comparing to the published results in literature by Yavuzturk et al. (2000). GenOpt was used to investigate the optimal design, control strategies and configuration. Life cycle cost of the system was used as objective function. Effects of using first year operating cost and $20^{th}$ year operating cost on optimization results were studied.

Findings and Conclusions: The modifications and addition of new models successfully modeled HGSHP systems in EnergyPlus. The results obtained from EnergyPlus model agreed well within the modeling differences with Yavuzturk results. Optimization proceeded to decrease the GLHE length to the minimum possible, since it was the highest contributor towards cost. Optimization of HGSHP systems was found to be system specific and highly affected by the load imbalance. Use of $20^{th}$ year operating cost instead of first year operating cost to calculate the life cycle cost affected the results of optimization.

ADVISER'S APPROVAL: _____Dr. Daniel Fisher_____