MODELING GROUND SOURCE HEAT PUMP SYSTEMS IN A BUILDING ENERGY SIMULATION PROGRAM (ENERGYPLUS)

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ABSTRACT

Despite the low energy and lower maintenance benefits of ground-source heat pump systems, little work has been undertaken in detailed analysis and simulation of such systems. Long-term transient ground heat transfer significantly affects the performance of these systems. Annual and multi-year simulation consequently becomes an invaluable tool in the design of such systems – both in terms of calculating annual building loads and long-term ground thermal response.

Models of vertical ground loop heat exchanges and water-to-water heat pumps have been implemented in the EnergyPlus program. The models are described, and techniques for modeling zone-system-plant interactions in the EnergyPlus environment presented. A case study is presented to show how the long-term (multi-year) effect of unbalanced loads on the ground loop, and the propagation of these effects through the system and zone models, can be calculated.

INTRODUCTION

Using the ground as a heat source or sink in an air conditioning system is attractive from a thermodynamic point of view, as its temperature is generally much closer to room conditions than the ambient dry bulb or wet bulb temperatures over the whole year. This results in both improved performance and increased capacity during extremely hot or cold weather. For these reasons, ground coupled heat pump systems are potentially more efficient than conventional air-to-air systems.

Ground source heat pump systems commonly consist of either water-to-water or water-to-air heat pumps coupled to closed-loop vertical borehole heat exchangers. Closed loop ground heat exchangers of this type consist of a borehole (of diameter 75-150mm) into which is inserted one or more loops of High Density Polyethylene pipe with a 'U' bend at the bottom. The borehole is then either back-filled or, more commonly, grouted over its full depth. The depth of the borehole varies between typically 30m and 120m.

The performance of closed-loop ground heat exchangers is rather different than air coupled heat exchangers in that the primary heat transfer mechanism is conduction rather than convection. The most significant implication of this is that, depending on the balance between extraction and rejection of heat from and to the ground, the ground temperature in the neighborhood of the heat exchanger may rise or fall, not only in response to short-term fluctuations in plant operation, but also over the life of the system.

This is particularly important where the building loads are cooling or heating dominated. In such cases the ground temperature may potentially rise or fall over a number of years, resulting in a lowering of performance of the heat pump as the fluid temperature changes in the same direction over time (potentially exceeding its operating pressure limits). A design goal must therefore be to control the rise or drop in temperature within acceptable limits over the life of the system.

The net heating or cooling of the ground over each season clearly depends on the accumulated heat rejection and extraction, and therefore on the building loads throughout the whole year. It also depends on the depth, number and configuration of the boreholes. It is important that the design methodology account for thermal interactions between the boreholes and with the far field. Any design methodology has to be based then on the building loads calculated throughout the whole year, not just the peak heating and cooling loads. Annual and multi-year simulation consequently becomes an invaluable tool in the design and energy analysis of such systems – both in terms of calculating annual building loads, and long-term ground thermal response.

In the following sections of the paper we describe models of a ground loop heat exchanger and a water-to-water heat pump. These models have previously been implemented in HVACSIM+ (Clark, 1985) and TRNSYS (SEL, 1997) and used as a research tool in the evaluation of new system types and optimum control strategies (Yavuzturk and Spitler, 2000). It is recognized that uptake of this technology is partly dependant on this type of model being readily accessible to professional engineers. To this end, work has focused on implementing these models in an annual energy and plant simulation tool – namely Energy-Plus (Crawley *et. al.,* 1999).

THE GROUND-LOOP HEAT EXCHANGER MODEL

The theoretical basis for the single U-tube, multiple borehole ground-loop heat exchanger model comes from the work of Eskilson (1987). His approach to the problem of determining the temperature distribution around a borehole is a hybrid model combining analytical and numerical solution techniques. A two-dimensional numerical calculation is made using transient finite-difference equations on a radial-axial coordinate system for a single borehole in homogeneous ground with constant initial and boundary conditions. The thermal capacitance and thermal resistance of the individual borehole elements, including the pipe wall, grout and fluid flow are neglected in Eskilson's model but accounted for in Yavuzturk's short timestep model as described in the following paragraphs. The temperature fields from a single borehole are superimposed in space to obtain the response from the whole borehole field.

The temperature response of the borehole field is converted to a set of non-dimensional temperature response factors, called g-functions. The g-function allows the calculation of the temperature change at the borehole wall in response to a step heat input. Once the response of the borehole field to a single step heat pulse is represented with a g-function, the response to any arbitrary heat rejection/extraction function can be determined by devolving the heat rejection/extraction into a series of step functions, and superimposing the response to each step function. Eskilson has calculated g-functions (data sets) for a wide variety of borehole configurations.

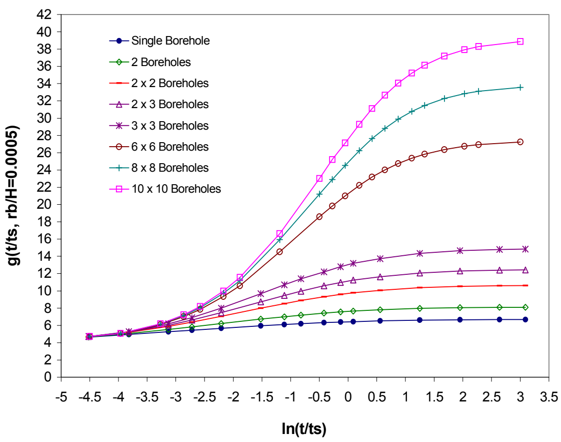

Figure 1: Temperature response factors (g-functions) for various multiple borehole configurations (Yavuzturk and Spitler, 2001).

Figure 1 shows the temperature response factor curves (g-functions) plotted versus non-dimensional time, where t is the time in seconds and ts is the time scale. The g-functions are plotted for a single rb/H ratio, where rb is the borehole radius and H the borehole depth, and a single L/H ratio of 0.1, where L is the borehole spacing. In Figure 1, various multiple borehole configurations are compared to the temperature response factor curve for a single borehole. The thermal interaction between the boreholes is stronger as the number of boreholes in the field is increased and as the time of operation increases. The detailed numerical model used in developing the long time-step g-functions approximates the borehole as a line source of finite length, so that the borehole end effects can be considered. The approximation has the resultant problem that it is only valid for times estimated by Eskilson to be greater than *($5*rb^2$/thermal diffusivity)*. For a typical borehole, that might imply times from 3 to 6 hours. However, for a model that is suitable for energy simulation, it is highly desirable that the solution be accurate down to an hour and below. Furthermore, much of the data developed by Eskilson does not cover periods of less than a month.

For short duration heat pulses, heat transfer within the borehole and heat transfer outside the borehole, in the radial direction, are much more important than heat transfer in the axial direction. It is consequently possible to use, a two-dimensional, radial-angular, finite volume model to calculate the average fluid temperature in the borehole in response to a heat pulse over sub-hourly time scales (Yavuzturk and Spitler, 2001). This temperature is then adjusted by the borehole resistance to determine the average temperature at the borehole wall and then non-dimensionalized to form a g-function. The borehole resistance includes the flow dependent convective heat transfer coefficient on the inside of the pipe, and is therefore adjusted at each timestep to properly reflect changes in the flow rate. The resulting short time-step g-function curve blends into the long time-step g-functions developed by Eskilson, as shown in Figure 2.

The operation of the ground-loop heat exchanger model has been verified by comparing calculated borehole temperatures for given periodic heat pulses, with a corresponding analytical solution (Murugappan, 2002). In one case a composite periodic heat pulse was formed from steady and sinusoidal components combined with a short pulse at the peak.

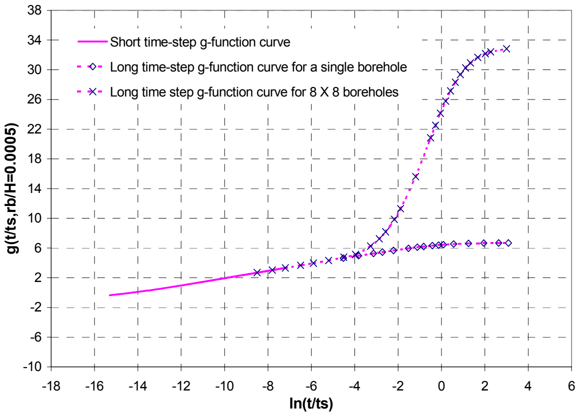

Figure 2: Short time-step g-function curve as an extension of the long time-step g-functions plotted for a single borehole and an 8 X 8 borehole field (Yavuzturk and Spitler, 2001).

The calculated and analytical borehole temperatures, and the differences in the results, are shown in Fig. 3 and 4. Predicted temperatures are within very reasonable limits, and are at a maximum of approx. 1.8 K only where a sharp step in load occurs. Field validation of the model has been reported elsewhere (Yavuzturk and Spitler, 2001).

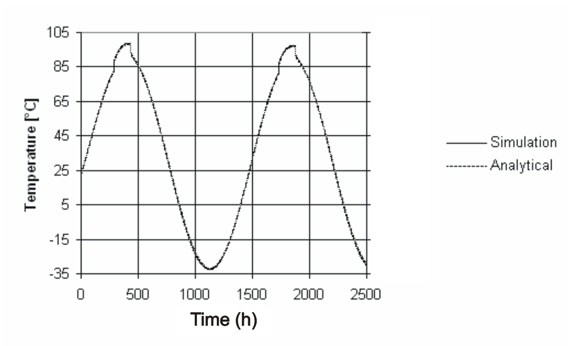

Figure 3: Comparison of borehole temperature predicted by analytical solution and model for a single borehole configuration with composite load.

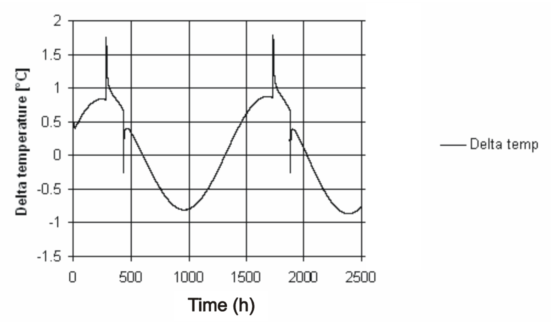

Figure 4: Difference between borehole temperatures predicted by the analytical and simulation models.

THE HEAT PUMP MODEL

Hamilton and Miller (1990) classified component models as equation fit models (also called "functional fit" or "curve fit" models) or deterministic models. Equation fit models treat the system as black box and fit one or more equations, using published data, to represent the system. Deterministic models represent the system as an assembly of components that are modeled by application of basic thermodynamic heat and mass balance relations.

This heat pump model may be denoted as a 'parameter estimation' model. The approach is to form a simplified model from basic thermodynamic heat and mass balance equations applied to each internal component, but to find the required parameter values using manufacturers published data. These parameters are estimated using a multi-variable optimization algorithm (Nelder and Mead, 1965). This approach has the advantage over deterministic models in that detailed data for individual components is not required, and over equation fit models in that the model behaves in a more physically reasonable way over a wider range of operating conditions. The latter benefit may be essential in a simulation where iterative calculation methods may require the model to predict performance with boundary conditions outside the normal range.

The water-to-water heat pump model presented here was developed by Jin and Spitler (2002). It incorporates a detailed compressor model along with simplified heat exchanger models (UA-effectiveness) and an ideal expansion device. The thermodynamic cycle for an actual single-stage heat pump system may depart significantly from the theoretical cycle. The principal departure occurs in the compressor. Hence, more attention has been paid to the thermodynamic processes occurring within the reciprocating compressor. Several modeling assumptions have been made:

- The compression and expansion in the compressor cycle are isentropic processes with equal and constant isentropic exponents
- The isentropic exponent is dependent on the refrigerant type; the values of the isentropic exponents are obtained from Bourdouxhe et al. (1994).
- The oil has negligible effects on refrigerant properties and compressor operation
- There are isenthalpic pressure drops at the suction and discharge valves.

The representation of the thermodynamic cycle is illustrated in Fig. 5. The compression process is assumed to be isentropic and when we allow for re-expansion in the clearance volume, the mass flow rate of the compressor refrigerant is a decreasing function of the pressure ratio so that,

$$\dot{m}_r = \frac{PD}{v_{suc}}\left[1 + C - C\left(\frac{P_{dis}}{P_{suc}}\right)^{1/\gamma}\right] \quad (1)$$

where $\dot{m}_r$ is the refrigerant mass flow rate, PD is the piston displacement, v_{suc} is the specific volume at suction state, C is the clearance factor, P_{dis} is the discharge pressure, P_{suc} is the suction pressure and γ is the isentropic exponent.

Making similar assumptions, the compressor work may be calculated from the thermodynamic work rate of an isentropic process:

$$\dot{W}_t = \frac{\gamma}{\gamma - 1}\dot{m}_r P_{suc} v_{suc}\left[\left(\frac{P_{dis}}{P_{suc}}\right)^{\frac{\gamma-1}{\gamma}} - 1\right] \quad (2)$$

The suction and discharge pressures play important roles in varying the magnitude of the theoretical mass flow rate. These two pressures are different from the evaporating and condensing pressures due to the pressure drop across suction and discharge valves. Hence, this effect was represented by including a constant pressure difference between the suction and discharge temperatures and the respective evaporating and condensing temperatures. This approach gave better agreement with manufactures data. A similar finding was made by Popovic and Shapiro (1995).

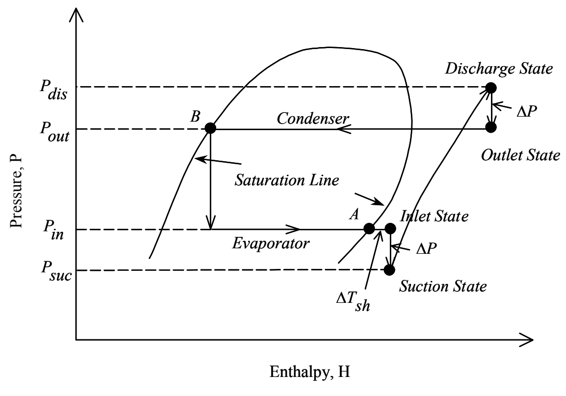

Figure 5: A Pressure-Enthalpy representation of the idealized Refrigeration Cycle (Jin and Spitler, 2002)

The electrical and mechanical losses are accounted for by assuming the actual work can be estimated from a constant component and some proportion of the theoretical work such that,

$$\dot{W} = \eta \cdot \dot{W}_t + \dot{W}_{loss} \quad (3)$$

where $\dot{W}$ is the compressor power input, $\dot{W}_t$ is the theoretical power, $\dot{W}_{loss}$ is the constant part of the electrical and mechanical losses, and η is a constant of proportionality.

The enthalpy of the refrigerant at the suction state is determined from the evaporator model by assuming a constant superheat. The condenser outlet conditions and entering evaporator conditions are calculated assuming zero sub-cooling and isenthalpic expansion. Heat transfer rates in the water/refrigerant heat exchangers, and the corresponding condensing and evaporating temperatures, can be calculated using a UA-effectiveness analysis of the heat exchangers and the respective water inlet conditions. Assuming isenthalpic expansion and constant superheat gives an idealized representation of a thermostatic expansion valve.

The required model parameters are given in Table 1. Values of each parameter are found using a multi-variable optimization procedure (Nelder and Mead, 1965) with heat pump power and heat transfer rates from catalogue data over a range of operating conditions. Objective function values were calculated using the differences between calculated and catalogue values of heat pump power, and between calculated and catalogue values of load-side heat transfer rates. Calculated values of heat pump power and load-side heat transfer rates are compared with the corresponding values of catalogue data, for a particular heat pump, in Figs. 6 and 7. Values of heat transfer rate and power are mostly with in a +/-10% error band.

Table 1
The Comparison of the Parameter Estimation Results for Heat Pump A in Cooling and Heating Modes

PARAMETER	COOLING	HEATING
Piston Displacement	0.012544 m^3/s	0.01299 m^3/s
Clearance Factor	0.05469	0.05347
Loss Factor	0.69	0.55
Constant Loss	2.80 kW	1.46 kW
Pressure Drop	92.20 kPa	93.66 kPa
Superheat	4.89 °C	4.8 °C
Source Heat Transfer Coefficient	4.0 kW/°C	4.20 kW/°C
Load Heat Transfer Coefficient	7.76 kW/°C	8.29 kW/°C

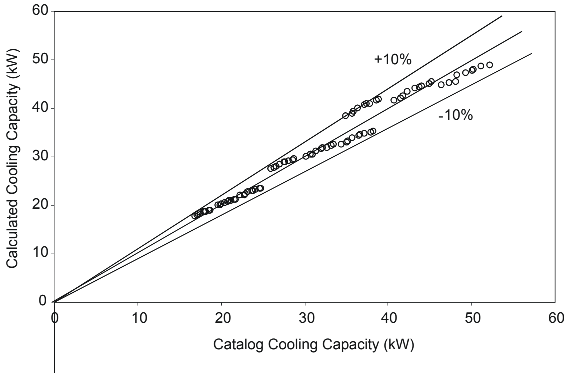

Figure 6: Calculated cooling capacity vs catalog cooling capacity (Jin and Spitler, 2002).

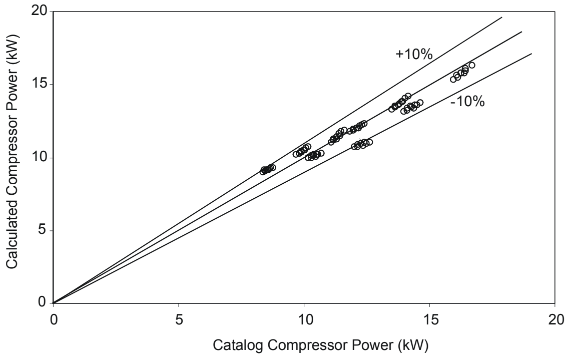

Figure 7: Calculated power vs catalog power (Jin and Spitler, 2002).

MODEL IMPLEMENTATION

Water-to-Water Heat Pump

The heat pump model has been implemented in EnergyPlus as a plant component model that, like other plant models (e.g. chillers), can be configured as a component of the plant and condenser loops, whose topology is defined by user input (Fisher, 1999). The heat pump is divided into heating and cooling components to allow connection of the 'load-side' to both heating and chilled water plant loops. The 'source-side' of each component is typically configured as part of a common condenser loop.

Within the heat pump model, heat transfer rates (and hence power) on the load and source sides are solved simultaneously using a successive substitution method. Once heat transfer rates have been found, the respective outlet fluid temperatures can be calculated. Implementation of the model requires that refrigerant properties (e.g. enthalpy at different temperatures and pressures) be available. This has been dealt with, in EnergyPlus, by providing utility functions that interpolate property values from tabulated data provided in user input. This approach allows any current and future refrigerants to be accommodated and eliminates reliance on encoded property equations for a small selection of refrigerants. Some representation of the on/off cyclic operation of unitary heat pumps has been implemented in an algorithm that forces the component on or off according to a user defined minimum cycle time.

Ground Loop Heat Exchanger

The vertical ground loop heat exchanger model, which simulates a configuration of single u-tubes, has been implemented in EnergyPlus as a plant component model that is intended to be configured as part of a condenser loop. To represent a typical ground-source heat pump system the ground-loop heat exchanger component is simply configured as the heat source/sink on the condenser loop along with the heat pump model. It should be noted that the flexible loop/component structure of EnergyPlus allows other heat rejecters (e.g. cooling towers or ponds) to be included in the condenser loop to allow representation of 'hybrid' ground-source heat pump systems.

The g-functions used by the model are configuration specific and account for both thermal interactions between the boreholes and with the far field. The g-functions are not calculated by the model, but are model input parameters. Within the ground loop heat exchanger model short and long time-step g-functions may be applied directly with hourly time-steps, and hence, hourly heat rejection/extraction pulses. However, this becomes computationally intensive when annual and multi-year simulations are performed. Since the importance of any given hour's response decreases as the hour gets further away in time, the loads are aggregated such that loads that occurred more than about two months previous to the current time are aggregated into 730 hour time blocks. Loads that occurred more recently are treated as hourly pulses. This approach gives significantly reduced computational time, while maintaining very good accuracy (Yavuzturk and Spitler, 1999). Further development of this approach has been required to enable the variable time steps used by EnergyPlus to be accommodated. Additional details of the load aggregation procedure implemented in the model are given by Murugappan (2002).

CASE STUDY

The modeling of ground-source heat pump systems within EnergyPlus is illustrated here by way of a case study. Architectural details of the building used in this study are taken from a small office building in Stillwater, Oklahoma. The building is shown in Fig.8. The building has a total area of approximately

1,300m^2. The following data and assumptions were made in the modeling of this building:

1. The building is modeled as eight thermal zones.
2. The air handling system is a draw-through system with reheat.
3. The office occupancy is assumed to be one person per 9.3 m^2 with a total heat gain of 132 Watts/Person of which 70% is radiant heat gain.
4. The lighting loads are assumed as 10.8 W/m^2.
5. The equipment plug load is 11.8 W/m^2.

The building was modeled as eight thermal zones with each zone serviced by a draw-through air handling system and reheat coils. There is a single chilled water loop that connects the cooling coil and the cooling mode heat pump. Similarly, the reheat coils are served by the heating mode heat pumps on a single hot water loop. The loop circulation pumps are modeled as separate components on the fluid loops in EnergyPlus. Since the purpose if this case study is to illustrate the performance of the ground source heat pump and ground loop heat exchanger models, an assessment of pumping requirements is not included in the discussion. The heat pump model parameter values are given in Table 1. The heat pumps operate on a condenser loop with ground loop heat exchangers. Though not necessarily practical design alternatives, ground loop heat exchanger configurations of 16, 32 and 120 boreholes of depth 73.2m were simulated in order to illustrate the effect of the ground loop on system performance. Each of the borehole fields is of a square configuration with borehole spacing of 3.66 m. Further details of the building can be found in Yavuzturk and Spitler (2000) and further details of the case study in Murugappan (2002)

The hourly building loads – calculated with weather data for Tulsa, Oklahoma – are shown in Fig.9. The load distribution shows cooling loads are more dominant than heating loads, with a peak-cooling load of 75 kW and peak-heating load of 65 kW. The calculated borehole wall temperatures for each configuration, over twenty years, are shown in Fig.10.

Figure 8: The case study building

(Stillwater, Oklahoma).

The borehole temperatures, for each configuration, can be seen to rise significantly during the first few years of operation. With a 16-borehole configuration, the borehole wall temperature at the end of the first year, and at the end of the 20th year, has risen by 3.1^{o}C and 8.7^{o}C above the initial ground temperature respectively. The reduction in heat pump performance over this period is shown in the energy consumption. In the first year the heat pump cooling energy consumption is 34.3 MWh but rises to 37.5 MWh in the 20th year. The rapid rise in temperatures, and the fact that peak temperatures start to approach the limits of heat pump operation (high pressure limits), mean that the system is probably undersized with only 16 boreholes.

With a borehole configuration of 120 boreholes the swing in temperature is much more modest. With this configuration, the borehole wall temperature at the end of the first year, and at the end of the 20th year, has risen by 2.1^{o}C and 4.1^{o}C above the initial ground temperature respectively. Though this ground loop heat exchanger is probably oversized, it does serve to illustrate the potential effect of ground loop sizing on heat pump performance. In the first year the heat pump cooling energy consumption is 29.2 MWh but only

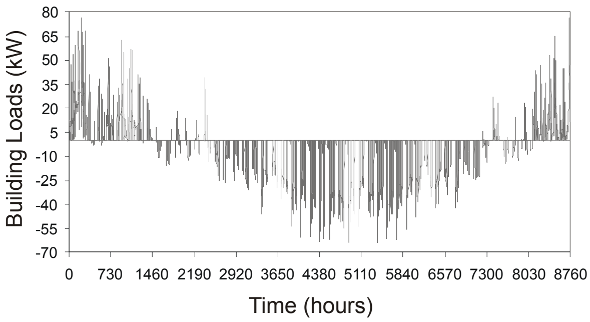

Figure 9: Annual hourly building loads for the example building (cooling loads are shown as negative values).

rises to 30.7 MWh in the 20th year. Heat pump performance is noticeably improved, and the degradation in heat pump performance over 20 years is much less. Thus, system optimization based on life cycle cost must include the interaction between the borehole model and the heat exchanger model as well as equipment and pumping costs.

CONCLUSIONS

Models of a water-to-water heat pump and ground loop heat exchanger have been implemented in the whole building annual energy simulation program EnergyPlus. The water-to-water heat pump model consists of simplified representations of heat exchanger and expansion device components with a more detailed compressor model. Multi-variable parameter estimation methods have been used to find model parameter values from readily available manufacturers' catalogue data. This model is able to reproduce this catalogue data within a +/-10% error bound. It has the advantage over deterministic models in that very detailed component data or measurements are not required, and over curve-fit models in that a wider range of operating conditions can be successfully dealt with.

The ground heat exchanger model can be used to predict the borehole and fluid temperature response over sub-hourly, annual and multi-year periods. The operation of this model has been verified by comparing results to analytical values. Using these models it is possible to represent ground-source heat pump systems in a flexible way, and examine their performance over the extended periods required for proper analysis. This has been demonstrated here by way of a case study. The functionality of EnergyPlus will allow these component models to be coupled with other heat rejection devices so that hybrid systems may be modeled and studied. Work is now underway to study such systems, further improve the computational efficiency of the model, simplify condenser loop configuration and represent part load operation.

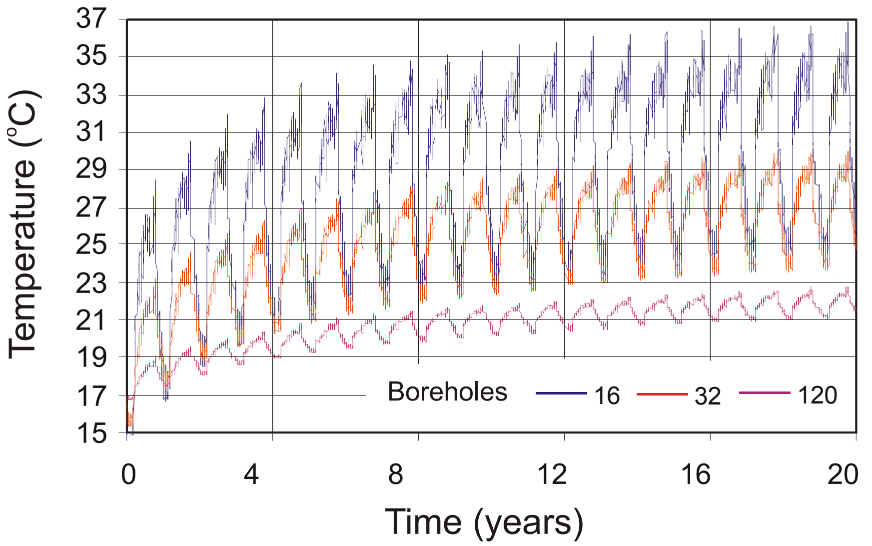

Figure 10: Comparison of the average borehole wall temperature for three ground-loop heat exchanger configurations: 16, 32 and 120 boreholes (Tulsa, Oklahoma weather data)

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