Recent Developments in Ground Source Heat Pump System Design, Modeling and Applications

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Summary

This paper gives an overview of recent research and developments in ground source heat pump system design, modeling, and applications for commercial and institutional buildings. Design methodology suitable for determining ground loop heat exchanger size and modeling suitable for predicting hourly or minutely response of the ground loop heat exchanger are presented. In addition, *in situ* measurement techniques for ground thermal properties, hybrid ground source heat pump system control, and a UK example are discussed.

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. For this reason, ground coupled heat pump systems are potentially more efficient than conventional air-to-air systems. In practice, ground-source heat pump systems have also proved to have lower maintenance costs due to the absence of equipment exposed to the atmosphere (1).

to reduce peak demand. Application of this technology has also been made in the small commercial and institutional sectors. Ground source heat pump systems have proved popular with some school authorities that are particularly attracted by the lower maintenance costs (2,3,4).

Details of a number of case studies involving application of ground-source heat pump technology in the United States have been given by the GHPC (5). Ground-source heat pump technology has been applied to only a few projects in the U.K. (6). This may partly be due to the lack of demand for residential air-conditioning in the U.K. The applications that have been made are mostly to small commercial buildings. Details of a number of U.K. case studies are available from a UK company (7).

In the following sections of the paper we set out firstly, a design methodology that has been developed for the sizing of vertical closed-loop ground heat exchangers. This is partly based on earlier work by Eskilson (8) in Sweden. Secondly, a model for the short time scale simulation of ground loop heat exchangers is described. This model is intended to be used in component-based simulation environments such as TRNSYS and HVACSIM+. It has been used as a research tool in the evaluation of new system types and optimum control strategies. Because determination of ground thermal properties is prerequisite to optimal design of the ground heat exchanger, in the third section of the paper we describe a method that has been developed for estimating the ground thermal properties from on-site measurements of temperature response at a test borehole.

“Hybrid” ground-source heat pump systems that incorporate supplemental heat rejection devices, such as cooling towers have become increasingly popular in the U.S. These systems are of interest for cooling dominated buildings and are briefly described. Finally, we have given an example application of a ground-source heat pump system to a petrol station building using data for operation in the U.K.

Ground-loop Heat Exchanger Modeling

A number of forms of ground coupling have been developed including open-loop and closed-loop systems. Open loop systems use vertical wells similar to those used for extraction of domestic water, from which ground-water is extracted and circulated through one of the heat pump heat exchangers. This water may be returned to the well, or released to another watercourse or well.

Closed-loop ground coupled systems can use horizontal pipe loops or vertical pipe loops in boreholes. Horizontal loops can be in a number of configurations including banks of 'U-tubes' or loosely coiled loops of pipe (so called 'slinky' heat exchangers) lay in narrow trenches. Horizontal systems are buried at a depth of approximately 1m. Their performance is not easy to model, as at this depth the heat transfer is somewhat influenced by soil moisture and therefore by weather conditions.

Closed-loop systems using vertical boreholes have been in use for more than two decades. Closed loop ground heat exchangers of this type consist of a borehole of diameter 75-150mm into which is inserted a loop of pipe with a 'U' bend at the bottom.

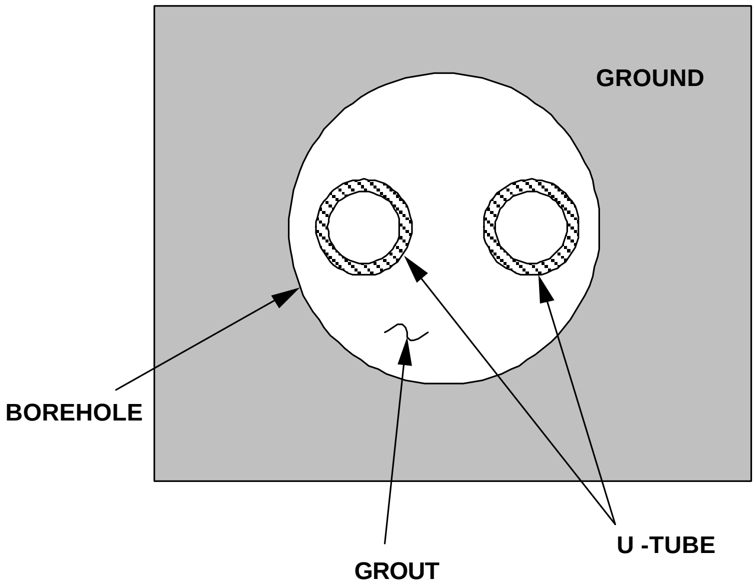


Figure 1: Cross-section of a typical borehole

The borehole is then either back-filled or, more commonly, grouted over its full depth. Grouting is normally required to prevent contamination of the ground water and gives better thermal contact between the pipe and the ground[1]. The pipe used in these systems is typically High Density PolyEthylene (HDPE) of nominal diameter in the range 22-32mm. The depth of the borehole varies between typically 30m and 120m. In this paper we confine our discussion to design and simulation methods and an example application using ground heat exchangers of the vertical closed-loop type.

Design of closed-loop ground heat exchangers is rather different from air coupled heat exchangers in that the primary heat transfer mechanism is conduction rather than convection (unless the groundwater flow is high). 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 over the life of the system. This is particularly important where the building loads are cooling dominated. In such cases the ground temperature may potentially rise over a number of years resulting in a lowering of performance of the heat pump as the fluid temperature rises. A design goal must therefore be to control the rise 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. The design methodology has to be based then on the building loads calculated throughout the whole year, not just the peak heating and cooling loads. Hence more information is required regarding the building loads than for sizing of a conventional system.

Two levels of modeling sophistication are of interest. First, it is desirable to have a design methodology that is satisfactory for sizing ground loop heat exchangers with minimal user input and computational time. Second, it is desirable to have a simulation model that can predict hour-by-hour (or shorter time interval) responses of the ground loop heat exchanger to continuously changing building loads. This approach allows the prediction of system energy consumption and electrical demand.

[1] This is in contrast to the practice in Sweden where the rock is mostly saturated granite and the borehole is not back-filled or grouted but allowed to fill with water.

Since both approaches presented in this paper are based on extensions to the model developed by Eskilson (8), Eskilson's methodology will be discussed first, followed by a description of the design methodology and simulation model.

Eskilson's Methodology

Eskilson's (8) 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 of the individual borehole elements such as the pipe wall and the grout are neglected. 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 for a time step. 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.

This process is graphically demonstrated in Figure 2 for four months of heat rejection.

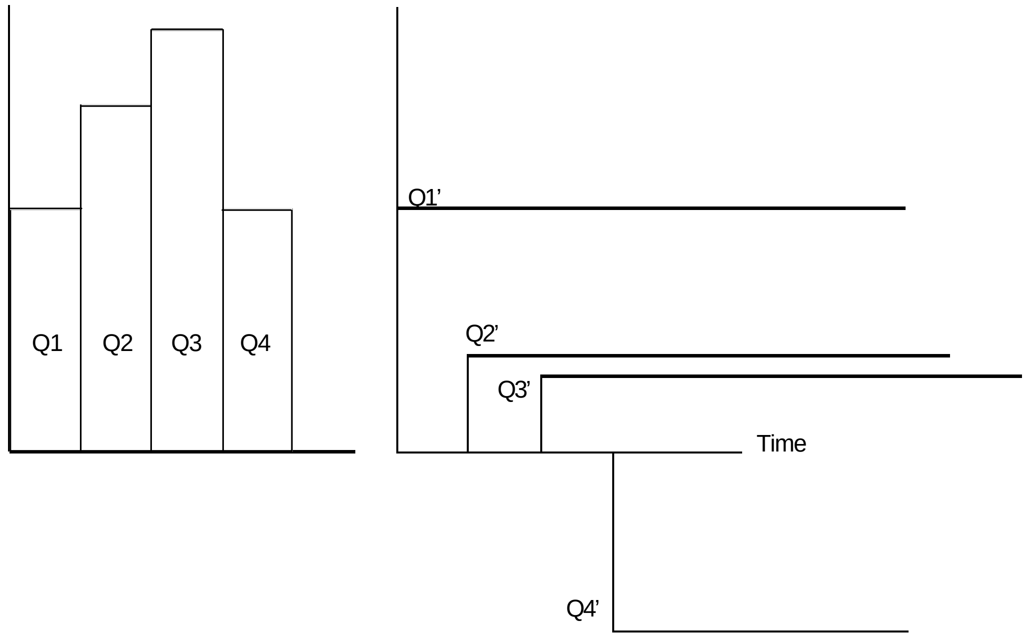


Figure 2: Superposition of piece-wise linear step heat inputs in time. The step heat inputs Q2,Q3 and Q4 are superimposed in time on to the basic heat pulse Q1.

The basic heat pulse from zero to Q1 is applied for the entire duration of the four months and is effective as Q1'=Q1. The subsequent pulses are superimposed as Q2'=Q2-Q1 effective for 3 months, Q3'=Q3-Q2 effective for 2 months and finally Q4'=Q4-Q3 effective for 1 month. Thus, the borehole wall temperature at any time can be determined by adding the responses of the four step functions. Mathematically, the superposition gives the borehole wall temperature at the end of the n^{th} time period as:

$$T_{borehole} = T_{ground} + \sum_{i=1}^{n} \frac{(Q_i - Q_{i-1})}{2\pi k} g\left(\frac{t_n - t_{i-1}}{t_s}, \frac{r_b}{H} \right) \qquad (1)$$

Where:

t = time (s)

t_s= time scale = $H^2/9\alpha$

H= borehole depth (m)

k =ground thermal conductivity (W/m-°C)

$T_{borehole}$ = average borehole temperature in (°C)

T_{ground} = undisturbed ground temperature in (°C)

Q = step heat rejection pulse (W/m)

r_b = borehole radius (m)

i = index to denote the end of a time step. (the end of the 1st hour or 2nd month etc.)

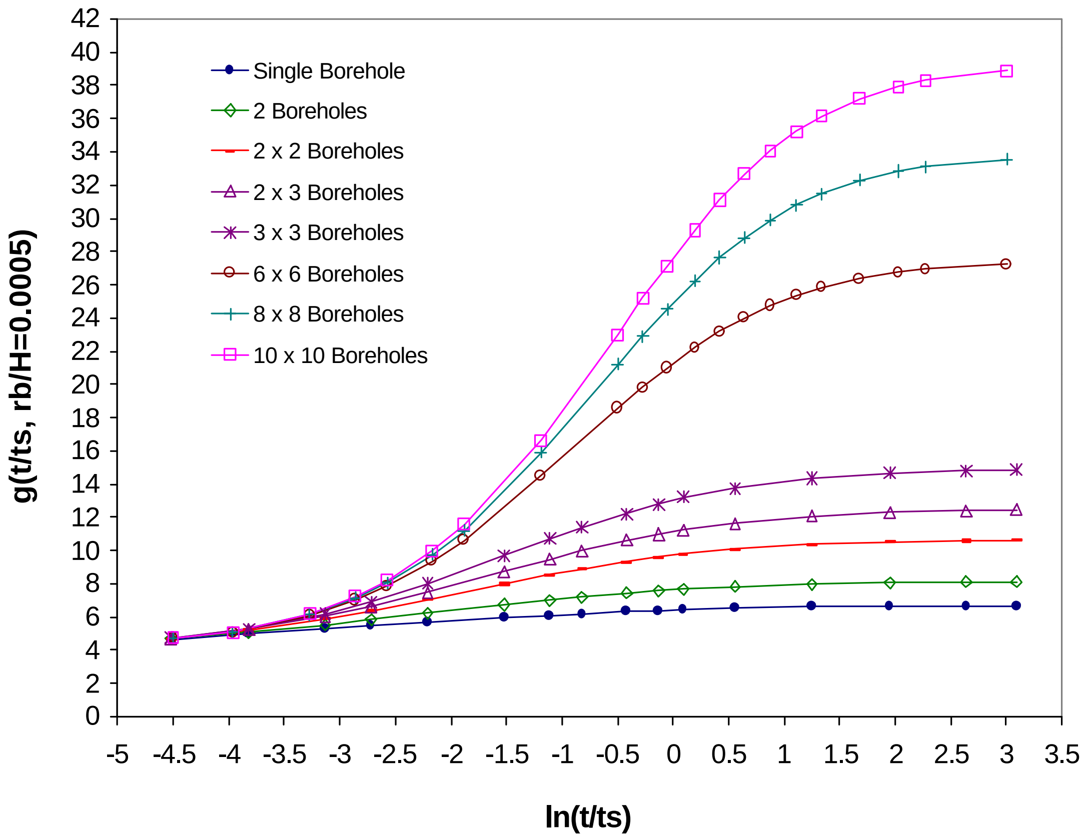


Figure 3: Temperature response factors (g-functions) for various multiple borehole configurations compared to the temperature response curve for a single borehole.

Figure 3 shows the temperature response factor curves (g-functions) plotted versus non-dimensional time for various multiple borehole configurations and compares them to the temperature response factor curve for a single borehole. The g-functions in Figure 3 correspond to borehole configurations with a fixed ratio of 0.1 between the borehole spacing and the borehole depth. 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 $\frac{5r_{Borehole}^2}{a}$. For a typical borehole, that might imply times from 3 to 6 hours. However, for the short time step model, 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 a heavy, saturated soil and a 250 ft (76.2 m) deep borehole, the g-function for the single borehole presented in Figure 3 is only applicable for times in excess of 60 days.)

Design Methodology

The design methodology[2] described in this section has been implemented in a commercially available ground loop heat exchanger design tool (9). The design methodology is based partly on the g-functions developed by Eskilson, partly on a simple heat pump model that represents the ratios of the heat rejected to the ground to cooling provided and heat extracted from the ground to heating provided, and partly on a simple analytical approximation for the response of the ground loop heat exchanger to a single peak heat extraction or rejection pulse. The heat pump model and analytical approximation are discussed below, before the overall procedure is described.

Application of Eskilson's Model

Eskilson's model only determines the temperature at the borehole wall. For sizing purposes, the entering fluid temperature to the heat pump is of interest. In order to determine the EFT, first the average fluid temperature inside the borehole must be determined; then the EFT may be determined. The temperature of the fluid inside the pipes inside the borehole is determined using a thermal resistance. (The thermal capacitance of the pipe, fluid, and grout are neglected.) The borehole resistance is the sum of the convective resistance at the pipe wall, the conductive resistance of the pipe, and the conductive resistance of the grout. The convective resistance is calculated with the Dittus-Boelter correlation. The conductive resistance of the pipe is determined with Fourier's law. The conductive resistance of the grout filling the borehole is determined from the shape factor correlations developed by Paul (10). Further details regarding the borehole resistance calculation may be found in Yavuzturk and Spitler (11). Once the borehole resistance has been determined, the average fluid temperature in the borehole may be determined as:

$$T_f = T_{borehole} + Q_i R_{TOTAL} \tag{2}$$

Where:

$T_{borehole}$ = average borehole wall temperature in (°C)

T_f = average fluid temperature in (°C)

Q_i = current heat rejection pulse (W/m)

Then, once the average fluid temperature in the borehole has been determined, the entering fluid temperature to the heat pump may be found from:

$$T_{entering} = \frac{\dot{q}_{rejection,net}}{2\dot{m}c_p} + T_f \tag{3}$$

Where:

$\dot{q}_{rejection,net}$ = the net heat rejection rate (W),

[2] Design methodologies available for residential ground loop heat exchangers have been reviewed by Cane and Forgas (12). Yavuzturk (13) provides a more up-to-date review of all available methodologies. Another design procedure commonly used in the U.S. is described by Kavanaugh (14).

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

c_p = specific heat of the working fluid (kJ/kg K),

$T_{entering}$ = the entering fluid temperature to the heat pump (°C).

Heat pump model

A very simple water-to-air heat pump model has been developed. In cooling mode, the ratio of heat rejection to the ground to cooling provided is given by:

$$\frac{\dot{q}_{rejection}}{\dot{q}_{cooling}} = a + bT_{entering} + cT_{entering}^{2} \tag{4}$$

Where:

$\dot{q}_{rejection}$ = the heat rejection rate (W),

$\dot{q}_{cooling}$ = the building cooling load met by the heat pump(s) (W),

a,b,c = coefficients determined by an equation fit of manufacturer's catalog data,

$T_{entering}$ = the entering fluid temperature to the heat pump (C).

In heating mode, the ratio of heat extraction from the ground to heating provided is given by:

$$\frac{\dot{q}_{extraction}}{\dot{q}_{heating}} = u + vT_{entering} + wT_{entering}^{2} \tag{5}$$

Where:

$\dot{q}_{extraction}$ = the heat extraction rate (W),

$\dot{q}_{heatng}$ = the building heating load met by the heat pump(s) (W),

u,v,w = coefficients determined by an equation fit of manufacturer's catalog data,

$T_{entering}$ = the entering fluid temperature to the heat pump (C).

The building cooling loads and heating loads are determined in advance by a building simulation program. The loads are assumed to be met by the heat pump or heat pumps, but since the entering fluid temperatures are not known *a priori*, they are determined simultaneously with the heat extraction and rejection rates.

Analytical approximation for the peak pulse

While Eskilson's g-functions are suitable for long (1 week or longer) heat rejection/extraction pulses, they are not intended to be used for shorter periods, such as hourly fluctuations. For most buildings, the cooling or heating load for a peak design day would vary approximately sinusoidally. As an approximation, the peak load is represented as a rectangular pulse with a user-specified duration. Based on comparisons with the more-detailed simulation model presented below, a three-hour duration pulse is suggested.

Using the user-specified peak load on the heat pumps, a peak heat rejection or extraction pulse is determined. The response to the peak pulse is estimated with a simple analytical approximation to the line-source model:

$$\Delta T_{borehole} = \frac{Q_{rejection,\,peak}}{4\pi k}\left\{\ln\left(\frac{4\alpha t}{r_b^2}\right)\right\} \tag{6}$$

Where:

$Q_{rejection,peak}$ = heat rejection rate, above monthly average heat rejection rate (W/m)

= ground thermal diffusivity (m^2/s)

Then, the peak entering temperature may be determined from Equations 2 and 3.

Operation of the Model

The design methodology requires that the user provide the following information:

- monthly heating and cooling loads on the heat pump or heat pumps, typically determined by a building energy analysis program;
- monthly peak heating and cooling loads, again on the heat pumps and typically determined by a building energy analysis program;
- information about the heat pump or heat pumps, from which the relationship between the entering fluid temperature to the heat pump and the heat rejected to the ground for a given cooling load and the heat extracted from the ground for a given heating load can be determined;
- thermal properties of the ground;
- geometric configuration of the ground loop heat exchanger;
- borehole diameter, U-tube diameter, grout thermal properties;
- thermal properties of the working fluid.

Assuming a given borehole depth, and the above information, the average fluid temperature in the borehole at the end of each month, the EFT at the end of each month, and the actual heat rejection rate for each month are determined simultaneously. Then, the responses to the peak pulses are determined for each month, and the resulting peak entering fluid temperatures to the heat pump(s) for each month are determined.

The program also has a sizing mode where the minimum borehole depth that will meet user-specified minimum and maximum peak temperatures is determined by searching with the simulation until the depth is found that is constrained by either the minimum or maximum peak entering fluid temperature.

Simulation Model

The simulation model described here has been presented, in considerable detail, by Yavuzturk and Spitler (11). A brief description will be presented here. The model is primarily aimed at applications in building energy analysis, where it is desirable to be able to predict system energy consumption on an hourly basis. It is also very useful for studying control and operating strategies for hybrid ground source heat pump systems (15), which incorporate supplemental heat rejection equipment, such as cooling towers, fluid coolers, hydronically heated pavement slabs, and shallow ponds.

The model was developed by extending Eskilson's g-functions down to times of less than one hour. Since the numerical model used by Eskilson to determine the g-functions is not suitable for short time steps, a more appropriate numerical model is used to estimate the temperature response of a single borehole for short duration, say one month down to under an hour, heat rejection/extraction pulses. 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. Hence, a two-dimensional, radial-angular, finite volume model has been developed. Complete details may be found in Yavuzturk, et al. (16). The geometry of the circular U-tube pipes is approximated by so-called pie-sectors, shown in Figure 4, over which a constant flux is assumed to be entering the numerical domain for each time step.

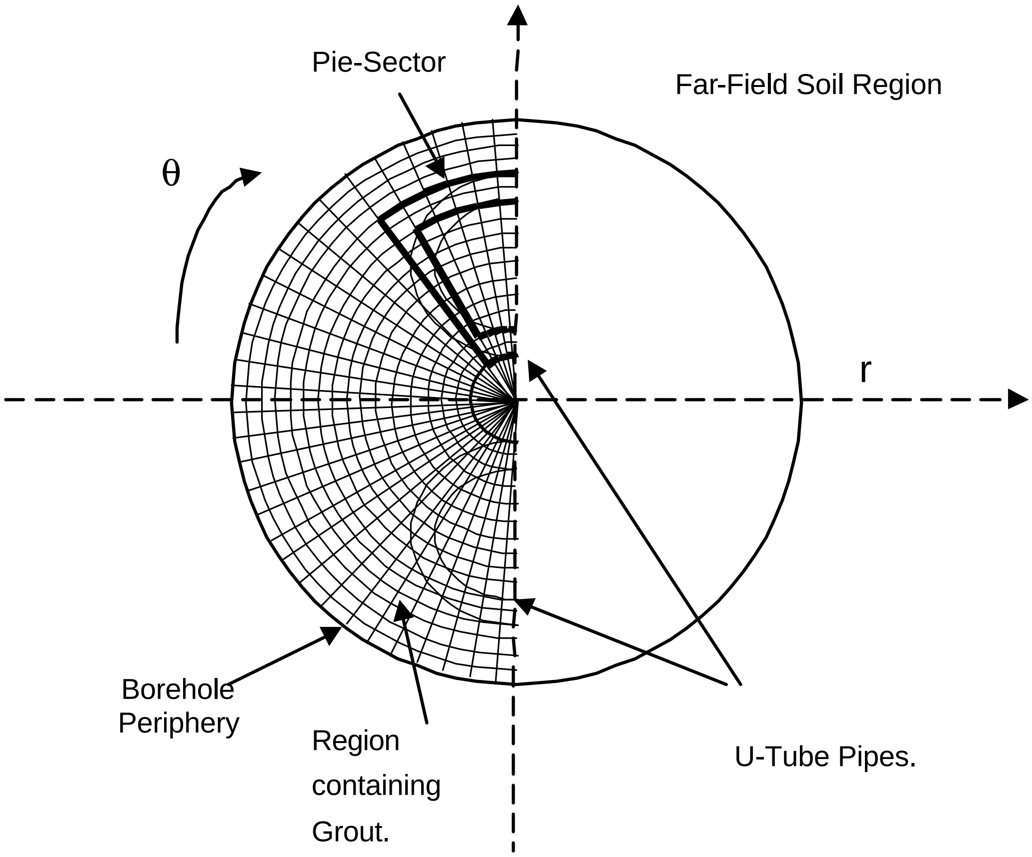


Figure 4: Simplified representation of the borehole region on the numerical model domain using the pie-sector approximation for the U-tube pipes.

The pie-sector perimeter length matches the perimeter of the U-tube pipe . Although not shown in the figure, the grid extends well beyond the edge of the borehole. The numerical model is used to calculate the average fluid temperature in the borehole. This 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. In this way, the correct average fluid temperature is determined using the g-functions and the borehole resistance.

The resulting short time-step g-function curve matches well at the boundary to the long time-step g-functions developed by Eskilson, as shown in Figure 5.

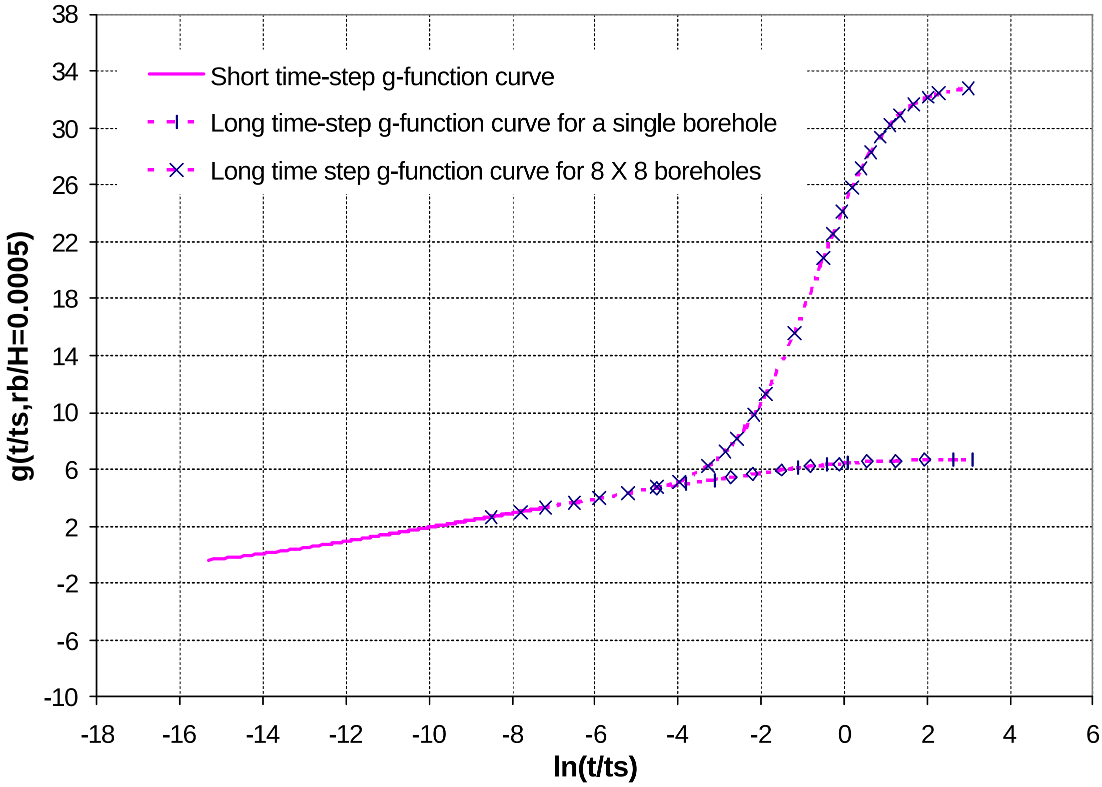


Figure 5: Short time-step g-function curve as an extension of the long time-step g-functions plotted for a single borehole and an 8 X 8 borehole field.

The short and long time-step g-functions are converted to a continuous set of response factors. Although they may be applied directly with hourly time-steps, and hence, hourly heat rejection/extraction pulses, this becomes computationally intensive when 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 one month 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. The load aggregation procedure is given in more detail in the paper by Yavuzturk and Spitler. (11).

In Situ **Measurement of Ground Thermal Properties**

Both ground loop heat exchanger design tools and the simulation model described above rely on some estimate of the ground thermal conductivity and volumetric specific heat. This estimate is critical to the design, yet it is very difficult to make. The required borehole depth or length is highly dependent on the thermal properties of the ground.

The traditional approach to estimating the ground thermal properties has been to first ascertain the type (or types) of soil or rock that surrounds the borehole. Once the type of soil or rock is determined, its thermal conductivity can be estimated from tabulated data, such as that contained in the Soil and Rock Classification for the Design of Ground-Coupled Heat Pump Systems Field Manual (17). Since thermal conductivity values for ground formation types are reported in the literature within a rather broad range of values, a method for more accurately estimating the ground thermal conductivity is highly desirable. A method for experimentally measuring the effective ground thermal conductivity using a test borehole is briefly presented here; it is presented in more depth by Austin, et al. (18).

The ground thermal conductivity can not be directly measured – its value must be inferred from temperature and heat flux measurements. Mogensen (19) described the concept of using such a measurement to estimate the ground thermal conductivity. Subsequently, development of an experimental apparatus began in 1995 at Oklahoma State University and was described by Austin (20). Simultaneously and independently, a similar apparatus was developed by Eklof and Gehlin (21). Gehlin and Nordell (22) report on results from *in situ* thermal response tests conducted using the mobile testing facility at various locations in Sweden to predict ground thermal conductivities.

In order to determine the ground thermal conductivity from the temperature and heat flux measurements, some model of the heat transfer in the ground such as the line source approach (23,19) or the cylinder source approach (24) must be utilized. They are of interest here for possible inverse use—estimating the ground thermal properties from the performance rather than the performance from the ground thermal properties. Specifically, we are interested in imposing a heat pulse of “short” duration

(1-7 days) and determining the ground thermal properties by analysis of the temperature response of the ground. Although the line source and the cylinder source approaches may be used inversely to estimate the ground's thermal conductivity, they require several simplifying assumptions, the effects of which cannot easily be quantified. A detailed numerical model of the borehole reduces the uncertainties associated with these simplifying assumptions by providing a detailed representation of the borehole geometry and thermal properties of the fluid, pipe, grout, and ground. It may therefore be expected to provide a more accurate estimate of the ground thermal conductivity.

Parameter Estimation Methodology

The method presented here uses the Nelder and Mead simplex algorithm (25) as part of a parameter estimation algorithm to estimate the ground thermal conductivity. An alternative parameter estimation based approach has been described by Shonder and Beck (26). The parameter estimation model utilizes a transient, two-dimensional numerical finite volume model of the vertical borehole (16) to estimate the temperature response of the ground to a known time-varying heat flux input. The differences between the experimentally measured temperature response and the estimated temperature response are minimized by adjusting the thermal conductivities of the ground and the grout. Specifically, the sum of the squares of the errors (SSE) is minimized:

$$\mathrm{SSE} = \sum_{n=1}^{N} (T_{\exp} - T_{\mathrm{num}})^2 \tag{7}$$

Where, N = The total number of data points over the duration of the experiment,

T_{exp} = Average of the calibrated input and output temperature at the n^{th} data point,

T_{num} = Average fluid temperature at n^{th} data point as predicted by the numerical model.

The summary information flow diagram for the parameter estimation algorithm is provided in Figure 6.

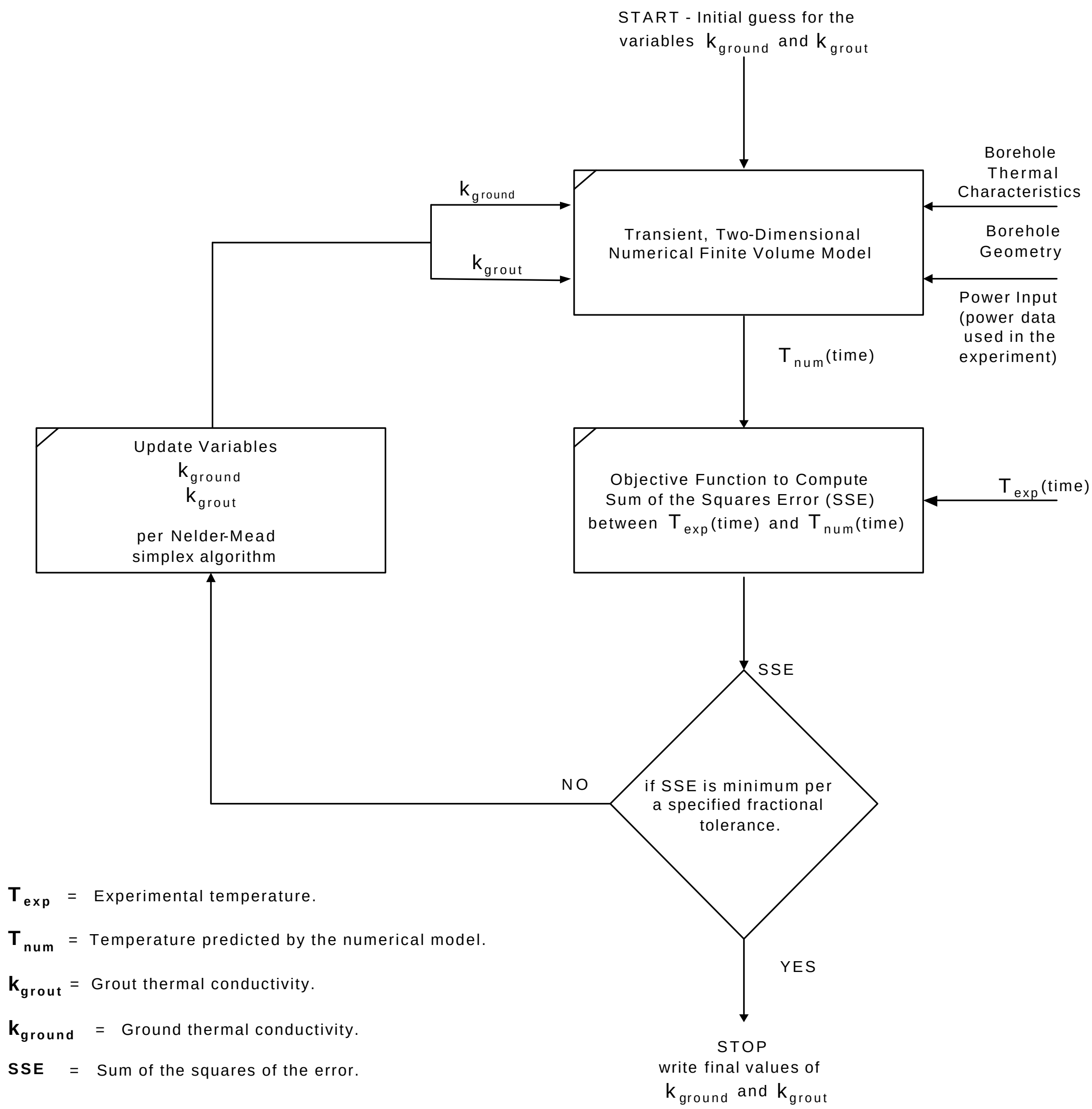


Figure 6: Information flow diagram for the parameter estimation algorithm

One, two, or more parameters might be estimated simultaneously. Although a number of approaches were tried, including estimating up to five parameters (soil conductivity, grout conductivity, soil volumetric specific heat, grout volumetric specific heat, and shank spacing) simultaneously, the most satisfactory approach only estimated the soil conductivity and grout conductivity. The grout conductivity acts as a surrogate for both the grout conductivity and the shank spacing.

Description of the Experimental Apparatus and Test Procedure

The experimental apparatus is housed in a trailer that can be towed to the site and contains everything needed to perform a test – the apparatus, two generators, and a purge tank (20). A simplified schematic of the test system is shown in Figure 7.

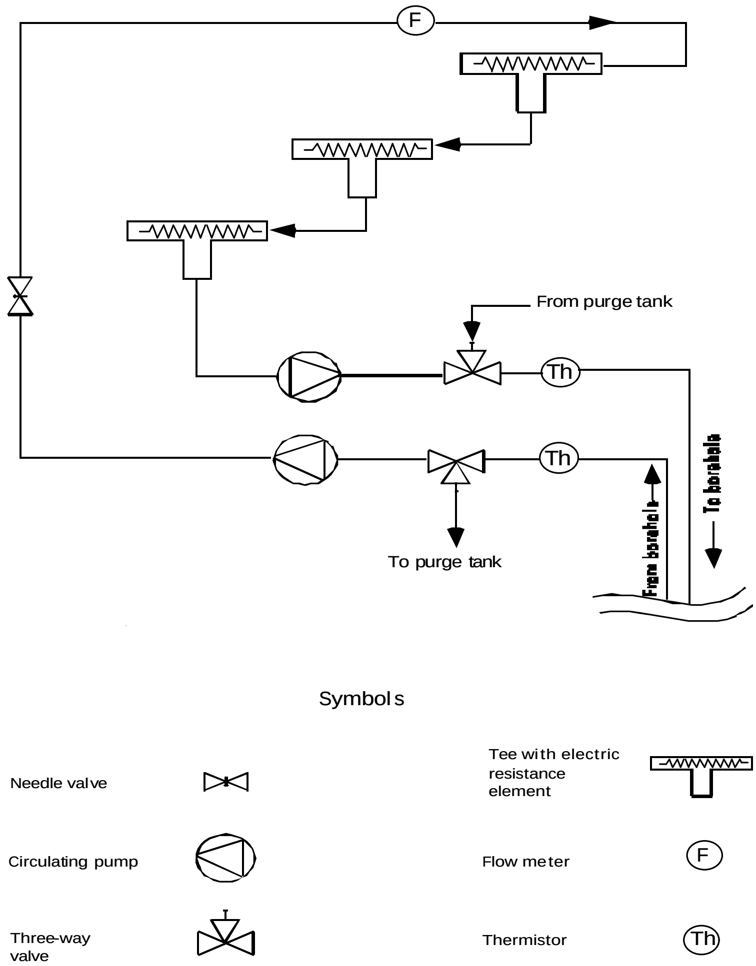


Figure 7: In situ thermal conductivity test system schematic

Once connected to a U-tube that has been inserted into a borehole, and after the system has been purged, a heat flux is imposed on the borehole using the three in-line water heaters, and the temperature response (average of inlet and outlet fluid temperatures, which changes with time) of the borehole is measured. Experimental measurements are made every 2.5 minutes using a data logger, and the power input, the entering/exiting fluid temperatures of the loop and the volumetric flow rate are downloaded to an on-board computer.

A test length of 50 hours was found to be satisfactory for typical borehole installations. A shorter test length is highly desirable, and may be the subject of future research.

Model Validation

For validation of the parameter estimation model predictions, several tests have been conducted where the ground conductivity was established independently.

One test was performed on a borehole that was drilled with a coring bit. The conductivity of 19 representative samples was then measured in a guarded hot plate apparatus (27, 28) to obtain an independent estimate for its thermal conductivity. Another test was performed using a medium-scale laboratory experiment (27, 29) where the geometry and thermal characteristics of a borehole are replicated under controlled conditions. The thermal conductivity of the soil material used in the experiment was determined independently with a calibrated soil conductivity probe.

A comparison between predicted and independently determined thermal conductivity values for both the cored borehole and the medium-scale laboratory tests shows a very reasonable agreement. A maximum deviation of about 2.1% is observed (cored borehole) while the simulated borehole with dry sand and the simulated borehole with saturated sand display a deviation of only about 2.0% and 1.3% respectively. As expected, the errors associated with the predictions of the thermal conductivity of the grout are greater since the second independent parameter is used as a surrogate to account for uncertainties in the borehole.

Sensitivity Analyses

A series of sensitivity analyses have been performed to evaluate the influence of a number of input parameters that cannot be determined exactly, but estimated with some uncertainty. (The term “input parameters” refers here to parameters that are not estimated with the parameter estimation procedure, e.g. far-field temperature, volumetric specific heats, shank spacing, borehole radius) In addition, the duration of the test and experimental errors impact the results, so a sensitivity analysis is performed for both.

A summary of the sources of uncertainties and their effect on the ground thermal conductivity estimation is given in Table 1.

Table 1 Summary of primary sources of uncertainties in the estimation of thermal conductivity of the ground.

Source	Estimated uncertainty in predicted k_{ground}
Length of Test – approx. 50 hours	$\pm$6.5%
Power Measurement. ($\pm$1.5% uncertainty.)	$\pm$1.5%
Estimate of the volumetric specific heat of the ground. ($\pm$5 Btu/ft^3-°F [$\pm$335 kJ/m^3-K])	$\pm$2.6% (average soils) or $\pm$6.3% (extremely dry soils)
Estimate of the borehole radius. ($\pm$0.5 inches [12.7 mm])	$\pm$3.6%
Estimate of the shank spacing. ($\pm$40%)	$\pm$1.6%
The numerical model.	$\pm$1.2%
Estimate of the far-field temperature. ($\pm$1 °F [$\pm$0.6 °C])	$\pm$4.9%
Total Estimated Uncertainty	$\pm$9.6% - 11.2%

The uncertainty in the input parameters has a corresponding uncertainty in the estimated ground thermal conductivity. Since the uncertainties described in Table 1 pertain to parameters that are all independent or nearly independent from each other they may be added in quadrature. Thus, the total estimated uncertainty of the ground thermal conductivity estimations falls within a range of about 9.6% - 11.2% depending on the level of the estimated thermal conductivity, since very low conductivity sands appear to be more sensitive to the estimate of the volumetric specific heat.

Hybrid Systems

The cost of drilling the borehole field for a ground-source heat pump system, although it depends strongly on the local geological conditions, can often be a substantial portion of the system capital cost. This is most likely in buildings where the demand is predominantly for cooling. In situations like this and where the thermal conductivity of the ground is low or drilling conditions are poor, the cost of the borehole field may make a ground-source system uneconomical. However, a compromise between first cost and energy efficiency may be possible by using a smaller borehole field and adding a supplemental heat rejecter into the heat pump water loop. Such systems have been termed 'hybrid' ground-source heat pump systems.

A number of different types of heat rejecters have been suggested for inclusion in the water loop of hybrid systems. Using the short time-step model of the ground heat exchanger described above, along with component models of these supplemental heat rejecters, it has been possible to simulate the performance of some of these systems and evaluate different system sizes and control strategies. These models have been validated using data from a number of experimental supplemental heat rejecters studied at Oklahoma State University.

Perhaps the most obvious candidate for a supplemental heat rejecter in a hybrid system would be a conventional open-circuit cooling tower. In a recent study (15) a number of operating strategies were investigated through simulation of this type of hybrid system using weather data from a number of US locations. One method investigated for controlling the cooling tower was to switch on the cooling tower only when a certain heat pump entering water temperature was exceeded. It was found that this simple strategy does not result in the cooling tower operating during the most advantageous weather conditions.

A second strategy studied involved operating the cooling tower on a simple schedule. This also results in wasting some pump and fan energy by running the tower when little heat transfer can be obtained. The most effective strategy was found to be one where the tower was controlled by the difference between the heat pump entering fluid temperature and the wet bulb temperature. This allows the cooling tower to be used under the most advantageous weather conditions (where the potential is greatest for heat rejection). For the climate conditions that were considered, this control strategy yields the lowest life cycle cost.. With the differential control strategy, a smaller supplemental heat rejection system is operated more frequently than a large unit operated less.

Another possible form of heat rejecter that is well suited for use in a loop with the ground heat exchanger is a pavement heat exchanger. This consists of pipes buried just below either a road or pavement surface. In a recent study (30) it was shown through simulation how a parking area of a medium size office building could be used as a supplemental heat rejecter and the length of the boreholes reduced from 122m to 76m. The borehole field consisted of 100 boreholes and so the reduction in drilling costs would have been substantial enough to make an economic case for this type of system, at the same time having lower operating costs than a system with air cooled condensers. This type of heat rejecter has other benefits when heat rejection is required in winter months as some de-icing of the surface can also be achieved.

Shallow ponds have also been suggested as possible heat rejecters for use in hybrid ground-source heat pump systems. Where ponds are required for either landscaping, irrigation or flood control purposes it is relatively simple and cost effective to introduce additional pipe coils at the bottom of the pond connected into the loop with the borehole field. Development of a model of a shallow pond heat rejecter is described in Chiasson *et al.* (31). Using the same example building as in the pavement heat exchanger study, a similar reduction in borehole depth could be achieved using a shallow pond of approximately 600m^2.

Example UK Application

To provide an example application of ground-source heat pump technology to a building in the UK, we have chosen a petrol station. We have taken the construction details for the simulation from an actual petrol station building that has been recently constructed in Stillwater, Oklahoma. The petrol station is relatively large, having twelve pumps and a building with a floor area of $232m^2$. The building has a large glazed area of $70m^2$ on the south façade, which is well shaded by the canopy. There were no doors or windows in the other walls except the north wall, which has a glass door of $4.6m^2$. The construction is not unlike that found in the UK, having brick and block work insulated cavity walls, flat concrete roof and concrete slab floor.

The building loads have been calculated using the BLAST (32) annual energy calculation program using typical weather data for London. The building requires cooling over an extended season, mainly due to the high lighting and equipment loads. The lighting loads have been taken as 32 W/m^2 and are scheduled on 100% except for the early hours of the morning, when they are scheduled at 60%. Similarly, equipment (plug) loads are taken as 32 W/m^2 and are scheduled on 100% except for the hours between 21:00 and 6:00, when they are scheduled at 50%. Occupancy is scheduled at a maximum of 10 people in the zone for most daylight hours, falling to 1.5 between 23:00 and 5:00. The building loads under these conditions are illustrated in Figure 8.

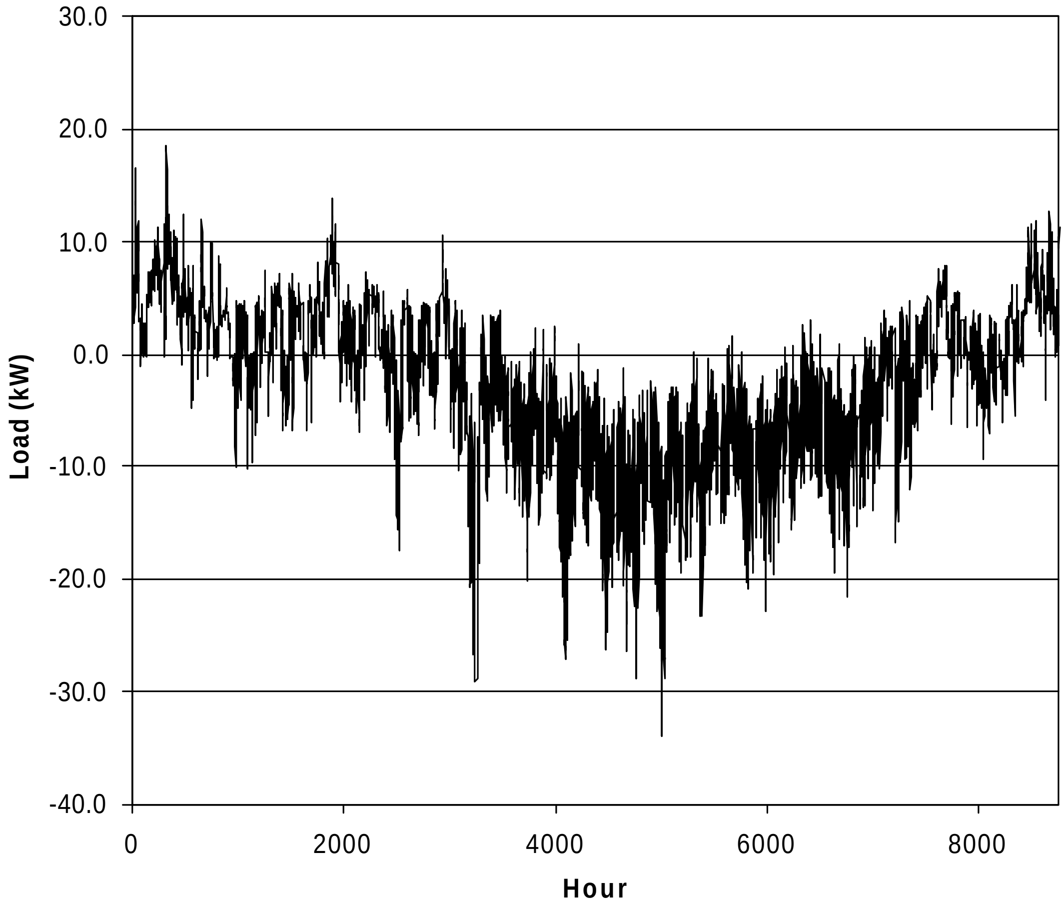


Figure 8: Annual hourly loads for a petrol station building in London.

In order to design a ground-loop heat exchanger for a project such as this one would ideally conduct an *in situ* test to estimate the ground thermal conductivity at the site. As this is a hypothetical project we have used a figure of 1.4 W/m.K which was measured in a thermal conductivity test conducted in London[3] and reported by Witte, et al. (33). This compares with *in situ* measured thermal conductivities elsewhere in the U.K. reported (34) as 3.2 W/m.K (Cornwall – wet rock/granite), 2.7 W/m.K (Exeter area – wet rock) 2.2 W/m.K (Southern Scotland – wet silts and clay). Another important parameter is the undisturbed ground temperature. This can be easily measured at the start of an *in situ* test. This has been taken to be 10°C in this case. A maximum annual entering fluid temperature (i.e. the tem-

[3] At this site, the soil and rock were moist, but not saturated.

perature of the water entering the heat pump condenser in cooling conditions) was selected as 32.0 ^{o}C. Other borehole design parameters are shown in Table 2.

Table 2 Ground Heat Exchanger Design Parameters

Parameter	**Value**
Borehole Diameter	106 mm
Pipe Nominal Diameter	25mm
Grout Conductivity	0.7 W/m.K
Ground Conductivity	1.4 W/m.K
Flow rate per borehole	0.157 l/s
Working fluid	Water
Undisturbed Ground Temperature	10.0 ^{o}C
Maximum Annual Entering Fluid Temperature to the Heat Pump	32.0 ^{o}C
Minimum Annual Entering Fluid Temperature to the Heat Pump	4.5 ^{o}C

The pattern in which the boreholes will be laid out has to be known before the borehole depth can be calculated (different patterns require different ‘g-functions’). This usually depends on the physical constraints of the site. In this case we have used an open rectangle plan of 3x4 boreholes giving a total of ten, spaced at 5m apart. (This would allow drilling to be carried out around the perimeter of the site without interference with other excavations.) Two water-to-water heat pumps were selected with a nominal capacity of 17.6 kW (5 tons) each. Using the design procedure described above (9) a bore-hole length of 79m was calculated.

The ground-source heat pump system has been simulated over a 20-year period using the TRNSYS component simulation environment in order to calculate the loop temperatures and the energy con-sumed by the heat pumps. The temperature of the water leaving the ground loop and entering the heat pumps has been plotted in Figure 9.

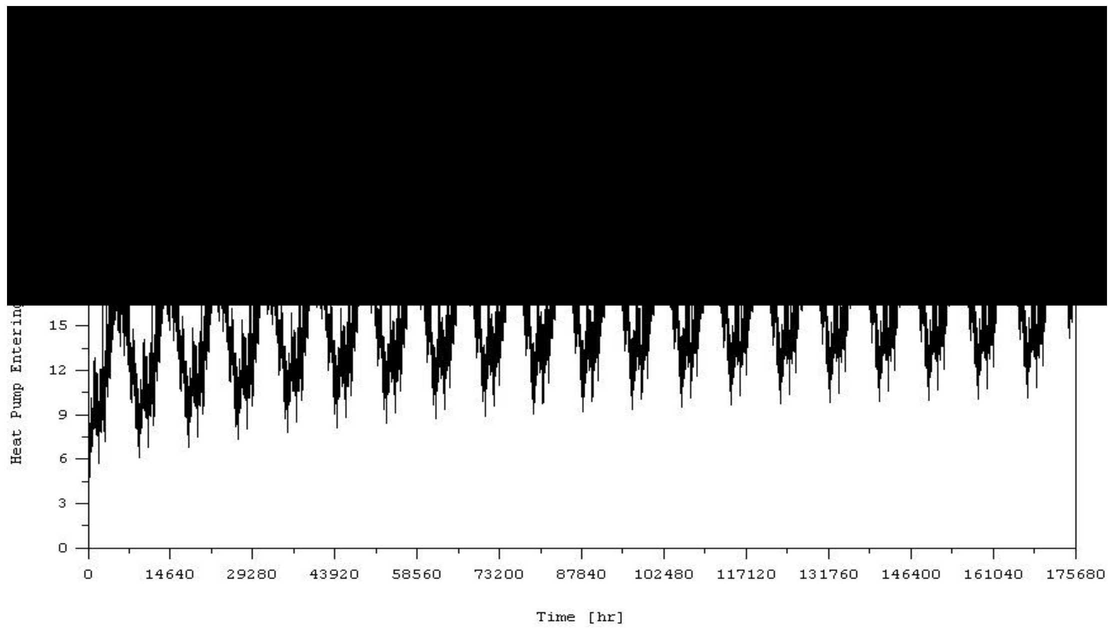
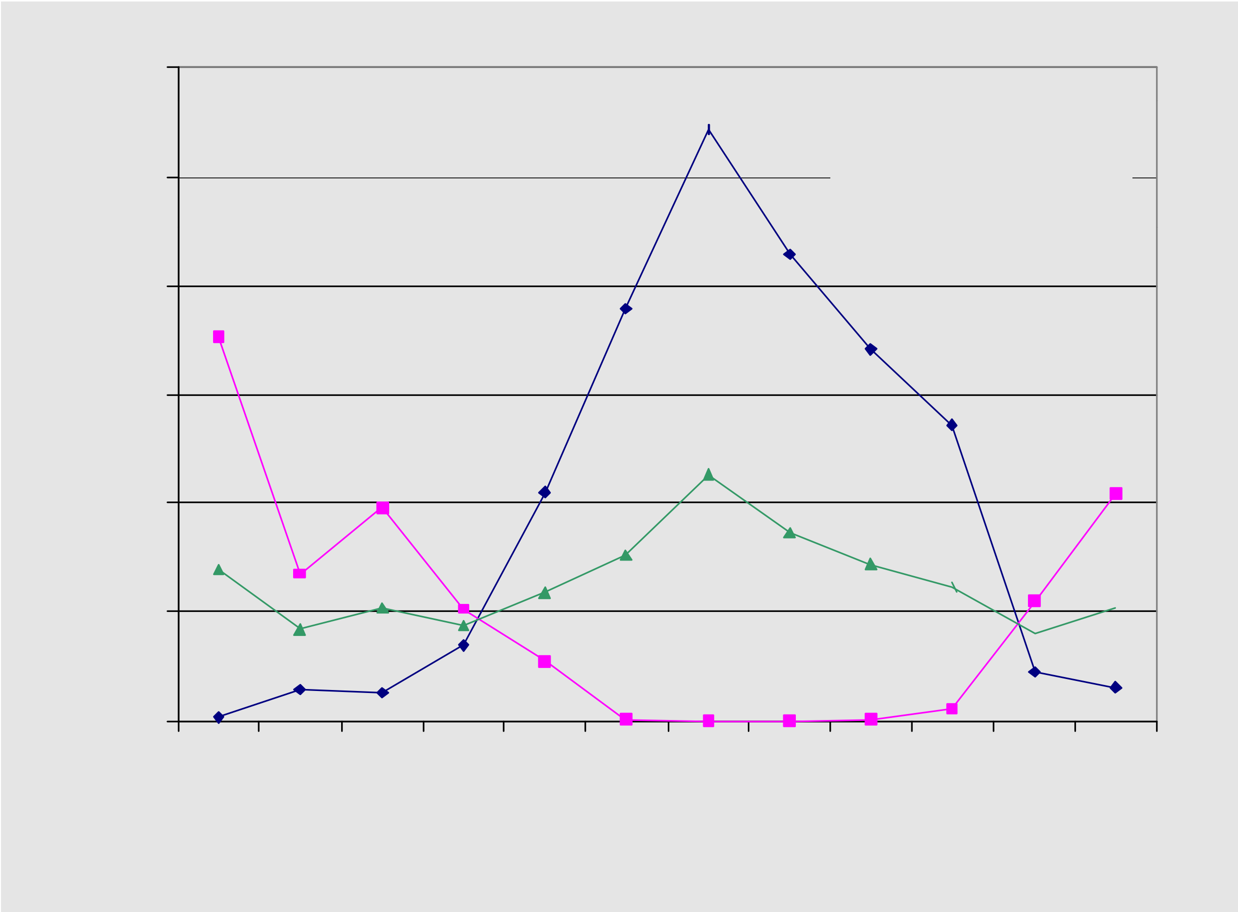


Figure 9: Heat Pump Entering Fluid Temperature over a 20-Year Hourly Simulation.

The peak annual entering fluid temperature can be seen to rise noticeably from year-to-year in the first few years and later approach a value of approximately 33.0 °C. This is slightly higher than predicted by the design program, and the discrepancy may be attributed to approximating the peak load as a rectangular pulse. The effect of the long term heating of the ground, and the need to design the heat exchanger taking this into account, can be clearly seen.

The results of this simulation have been compared to a calculation of the annual energy consumption for the same building but with a DX cooling system with a gas fired boiler and radiator heating system. The energy consumption of this second system was calculated using BLAST. The resulting monthly electrical and gas energy consumption of the two systems are shown in Figure 10.



The electrical energy consumption of the ground-source heat pump system includes an estimate of the pump energy usage. The energy consumption of the ground-source systems has a much lower peak during the summer months. The annual electrical energy consumption for DX cooling system is 23.8 MWh, compared to 15.3 MWh for the ground-source heat pump system.

In a project such as a petrol station there are several other possibilities for energy saving measures using the heat pumps and water loop. It has been shown possible to include much of the refrigerated equipment used in the shop on the water loop. Heat discharged into the loop can also be used for water heating such as in the car washing equipment. Use can also be made of pavement heat exchangers to reduce the load on the borehole field and provide a snow-melting capability. A number of petrol station projects in the US have included different combinations of these measures and demonstrated further overall energy savings (35).

Conclusions

Ground-source heat pump technology has developed over the last two decades to a point where systems are being routinely installed on small and medium-sized projects in many parts of the U.S. Sufficient training assistance has been provided that there are now many small contractors able to design and install such systems competitively. Use of the ground as a heat sink rather than the air is advantageous, in terms of energy savings, throughout the heating and cooling season.

A procedure for the design of vertical closed-loop ground heat exchangers has been presented. The procedure takes account of the cumulative effect of the building loads rejected to and extracted from the ground loop on its long-term performance. This procedure has been developed from the earlier work of Eskilson. It has been shown how, given the borehole parameters, the building loads and an estimate of the ground thermal conductivity and undisturbed temperature, the length of the boreholes can be calculated to maintain the heat pump entering fluid temperature below its design limit over the life of the project.

The thermal conductivity of the ground is the single most important parameter that affects the performance of the ground-loop heat exchanger. Optimal designs can only be achieved with accurate estimates of this conductivity. A method has been described that can be used to accurately estimate the ground thermal conductivity from test borehole thermal response data taken over a period of about fifty hours.

Simulation of the performance of the ground-loop heat exchanger requires a model that can predict the response of the ground to the hourly changing heat extraction and rejection that occurs throughout the year. By extending the ideas of Eskilson, it has been shown how the response can be predicted by considering the loads on the heat exchanger as a series of superimposed pulses of heat. This type of model can be used in short time-step simulations. Using this approach it has also been possible to model a variety of hybrid systems, where different forms of supplemental heat rejecter coupled in series with the ground-loop heat exchanger, are used.

An example application of a ground-source heat pump system to a building in the UK has been illustrated. It was demonstrated that the peak electrical demands of the systems were noticeably lower than an equivalent DX cooling system. Although many ground source heat pump systems are installed in residential buildings in the U.S.A., the market for small and medium-sized commercial and institutional buildings continues to grow. There may be little demand for residential systems of this type in the UK, but considerable potential must exist for this technology in the small/medium commercial sector of the UK and other parts of Europe.

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