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International Ground Source Heat Pump Association Conference & Expo
March 14-16, 2017
Crowne Plaza Denver Airport Convention Center
Denver, Colorado

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Proceedings Book Designed By:
Tim Taylor
IGSHPA Publications Intern
By my count, this is the 30th annual IGSHPA conference since 1987. IGSHPA conferences have usually been dominated by practical, industry-oriented presentations. Over the years, there have also been a significant number of research presentations, though these were seldom accompanied by a paper. Outside of IGSHPA, there are several conference series, such as the IEA Stock conferences, the IEA Heat Pump conferences, ASHRAE meetings and Clima conferences that have varying numbers of ground-source heat pump related papers mixed in with other topics. As a result, no existing conference series reliably draws a high percentage of active GSHP researchers from around the world. I thought this was a shame and it seemed like adding a “research track” with peer-reviewed papers to the existing IGSHPA conference might be a good way to draw together researchers from around the world. It would also have the additional benefit of allowing researchers and practitioners to exchange ideas. I proposed this to the IGSHPA board in October of 2015 and they approved organizing a research track as a “conference-within-a-conference.”

I prevailed upon some well-known researchers in the GSHP field to help me form the Executive Scientific Committee: Michel Bernier, Zhaohong Fang, Signhild Gehlin, and Simon Rees. Together, we wrote the call-for-papers and developed lists of possible International Scientific Committee members and authors. 30 distinguished researchers (see previous page) joined the International Scientific Committee and the first call went out in January of 2016. The conference topics were set as:

- Design of ground heat exchangers
- Performance of alternative GHE designs.
- Modeling and simulation of ground heat exchangers
- Validation of GHE and GSHP system models.
- Thermal response tests for measurement of ground thermal properties
- Optimal control and operation of ground heat exchangers and GSHP systems.
- Modeling and simulation of GSHPs and GSHP systems.
- Measured performance of GSHP systems.
- New system configurations and supporting models.
- Oft-neglected phenomena (moisture transport, freezing/thawing, near-surface effects)

A total of 70 abstracts were submitted, leading to 52 papers submitted, which after the review process finally came to 46 papers scheduled to be presented here in Denver. The review process for each paper was managed by one of the Executive Scientific Committee members and each paper was reviewed by at least two peer reviewers drawn from the International Scientific Committee and other experts. Most of the papers went through two rounds of reviews before being accepted. A big thank you goes out to all the authors, committee members and reviewers!

To help insure that the papers remain widely available, they will be available from both the IGSHPA website and SHAREOK, which is the joint institutional repository for the University of Oklahoma Libraries and Oklahoma State University Libraries. All papers have received digital object identifiers (DOI); these appear on each paper and give a persistent and permanent link to the paper.

I wish everyone a fruitful and productive conference!

Dr. Jeffrey D. Spitler
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An Updated Assessment of the Technical Potential of Geothermal Heat Pump Applications in the United States

Xiaobing Liu
Patrick Hughes
Jeffrey Spitler
Arlene Anderson

ABSTRACT

This paper presents an updated assessment of the technical potential of applying geothermal heat pump (GHP) systems in businesses and homes of the United States. The assessed technical potential includes energy savings, carbon emissions reductions, and consumer energy cost savings. This assessment is based on energy consumption data obtained from the latest survey of the energy consumption of residential and commercial buildings, conducted by the Department of Energy’s Energy Information Administration. It uses energy savings data for GHP systems compared with existing conventional HVAC systems, which were obtained from the results of a series of computer simulations. The impacts of various climate and geological conditions, as well as the efficiency and market share of existing conventional HVAC systems, have been taken into account in the assessment.

INTRODUCTION

Geothermal heat pumps (GHPs), also referred to as ground source heat pumps (GSHPs), have been proved capable of producing large reductions in energy use and CO₂ emissions in buildings while satisfying the demands for space heating (SH), space cooling (SC), and domestic water heating (DWH).

The U.S. Department of Energy (DOE) Geothermal Technologies Office (GTO) is developing a Geothermal Vision Study (Vision Study) to articulate GTO’s investment strategies; discuss geothermal growth scenarios for near future; and address all market segments of the geothermal industry. The GHP is one of the thermal applications of low-temperature geothermal resources included in the Vision Study. This paper provides a brief review of the current status of GHP applications in the United States and presents an updated assessment of the technical potential of GHP applications in both residential and commercial buildings.

CURRENT STATUS

GHPs have been used in all 50 states and the District of Columbia in the United States (EIA 2010). About 52% of domestic GHP shipments went to ten states: Florida, Illinois, Indiana, Michigan, Minnesota, Missouri, New York, Ohio, Pennsylvania, and Texas. Current GHP applications are more concentrated in areas with a cold climate and high population density.

Xiaobing Liu (liux2@ornl.gov) is a R&D staff and Patrick Hughes is a program manager at Oak Ridge National Laboratory. Jeffrey Spitler is a professor in mechanical engineering at Oklahoma State University. Arlene Anderson is a technology manager at the US DoE.
A recent Navigant Research report (2013) indicates that the United States represented 29% of global GHP installations by capacity, with 13,564 MWt (3.9 million tons, or 1.1 million GHP units given the typical GHP unit size is about 12 kWt) installed by 2012. These GHP systems provide space conditioning to roughly 199 million m² (2.14 billion ft²) of residential and commercial buildings in the United States. The current market share of GHPs in the U.S. heating, ventilation, and air-conditioning (HVAC) market is approximately 1% (EIA 2016a). A report issued by Priority Metrics Group (2009) estimated that the GHP market in the United States was about $3.7 billion in 2009, including design, equipment, and installation. It is estimated that the total revenue from sales of domestic GHP units was approximately $319.5 million in 2009 (EIA 2010).

The installed cost of GHPs varies widely, depending on geological conditions, building loads, system designs, and heat pump equipment. A few surveys have been conducted in the United States to collect cost information for GHPs. According to those surveys, the average cost of a commercial GHP system was $20.75/ft² in 2012 (Kavanaugh et al. 2012). The typical price of a GHP system installed in a new home was in the range of $3,000–5,000 per cooling ton (Ellis 2008).

The high initial cost and the lack of public awareness and strong governmental support are believed to be the major barriers preventing rapid adoption of GHPs in the United States. Recent low prices of oil and natural gas (NG) reduce the monetary value of the energy savings, which makes consumers less willing to invest in GHPs. Finally, tax credits for GHP installations will expire at the end of 2016 unless the industry is successful in its efforts to persuade Congress to extend the tax credits. The high initial cost barrier for GHP deployment may be overcome by breakthroughs in the following areas:

- Lower-cost ground heat exchangers and customized drilling techniques/equipment for GHP
- Better design of GHPs as a result of more information on the ground formation
- Volume manufacturing of GHP equipment
- Financial incentives or third party financing
- Integration of GHPs as a part of utility infrastructure in new developments

**METHODOLOGY**

The GHP technical potential is assessed based on (1) energy consumption data obtained from the latest survey of energy consumption in residential and commercial buildings, which is conducted by DOE’s Energy Information Administration (EIA); and (2) energy savings data for GHPs compared with existing conventional HVAC systems, which are calculated based on computer simulation results. These computer simulations account for many factors affecting energy savings, including thermal loads (determined by the location, building envelope, and activity of the building), performance of the existing HVAC systems, local geological conditions (i.e., undisturbed ground temperature and ground thermal conductivity), and the performance of GHPs.

Unlike previous studies (Hughes 2008; Liu 2010), which relied on national or census region–level building energy consumption data, this study uses a database of county-level site energy consumption data in residential and commercial buildings, which was developed by the National Renewable Energy Laboratory (NREL) (McCabe et al. 2016). The county-level data for residential buildings were derived by disaggregating regional-level (states, aggregates of states) site energy consumption data for SH, SC, and DWH—reported in EIA’s 2009 Residential Energy Consumption Survey (RECS; EIA 2013)—by the most recent count of housing units in each county, which is extracted from the National Historical Geographic Information System for a 5-year period (2009–2013). The county-level data for commercial buildings were derived by multiplying the total square footage of each type of commercial building in each county—extracted from the Federal Energy Management Agency’s (FEMA) Comprehensive Data Management System (CDMS)—by the average site energy intensities (kBtu/ft²) for SH, SC, and DWH for each particular type of commercial building in the climate zone in which the county is located. The average site energy
intensities are provided in the 2009 Buildings Energy Data Book (DOE 2009) by principal building activity and climate zone.

The procedure for assessing the technical potential of GHPs is depicted in Figure 1 and explained in the following sections.

![Figure 1](image)

The process for evaluating the technical potential of energy savings by GHPs in a county.

Reference buildings for residential and commercial sectors

A reference building—a 1,644 ft² (153 m²) one-story, slab-on-grade, wood-frame house—is used to represent typical U.S. homes with SH (DOE 2009). There are 18 building types (adopted from FEMA’s CDMS) in the county-level commercial building energy consumption database. These building types are categorized into five groups based on the similarity of the building activities, and each group of commercial buildings is represented in this study by one of DOE’s Commercial Reference Buildings (CRB; DOE 2012). Table 1 lists the categories of the building types and corresponding CRB models. Totally, 13 locations (cities) were selected to represent the major climate zones (IECC 2009) in the Continental United States. Simulations of the reference buildings were used to evaluate the relative differences in energy consumption between the existing conventional HVAC system and the GSHP system (i.e., the energy saving percentage) to satisfy the same SH and SC demands for a particular building in a given climate zone. It was found that the size of a particular type of building does not affect energy saving percentages significantly so the impact of building size on the energy saving percentage is not accounted for in this study. On the other hand, the characteristics of the existing conventional HVAC systems (e.g., efficiency of cooling equipment or energy source for SH) make a significant difference in the energy saving percentage, which has been taken into account in this study as discussed below.

Simulations of GHP system and baseline HVAC systems

The residential GHP system simulated in this study consists of a packaged water-to-air heat pump (WAHP) unit with a two-stage scroll compressor and variable-speed electronically commutated fan-motor, a properly sized and highly energy-efficient loop fluid circulator, and a properly designed and installed vertical-borehole ground heat
The nominal cooling efficiency of the two-stage GHP unit is energy efficiency ratio (EER)\(^1\) 18.2 at full capacity and EER 27 at 76% of full capacity. The nominal heating efficiency of the two-stage GHP unit is COP 4 at full capacity and COP 4.5 at 76% of full capacity\(^2\). The ground heat exchanger is sized to maintain the fluid temperature from the ground loop (the entering fluid temperature to the GHP unit) within the range of 30 to 95°F (~1 to 35°C) for given building loads, ground thermal properties, and undisturbed ground temperatures. The commercial GHP system simulated in this study is a distributed GHP system, which is the most commonly used GHP system in the United States (Liu et al. 2015). The simulated commercial GHP system consists of multiple two-stage WAHPs connected by a common water loop, with each WAHP serving an individual zone in the building. Although a GHP may also provide partial or full DWH (e.g., with a desuperheater or more advanced integrated heat pump), this study does not account for this service.

### Table 1. Commercial Building Categories and Mapping

<table>
<thead>
<tr>
<th>Categories</th>
<th>FEMA CDMS building type</th>
<th>CRB model</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>Office</td>
<td>Services, bank, office/clinic, church</td>
<td>Medium office</td>
<td>3 stories, 53,620 ft(^2) (4950 m(^2)); direct expansion (DX) cooling and gas-fired furnace with electric reheat</td>
</tr>
<tr>
<td>Lodging</td>
<td>Lodging, dormitory, nursing home</td>
<td>Small hotel</td>
<td>4 stories, 40,095 ft(^2) (3725 m(^2)); DX cooling and gas furnace for common area; DX cooling and electric resistance heating for guest rooms</td>
</tr>
<tr>
<td>Education</td>
<td>Grade school, college/university, theater</td>
<td>Secondary school</td>
<td>2 stories, 210,954 ft(^2) (19,600 m(^2)); chilled water and hot water (from gas-fired boiler)</td>
</tr>
<tr>
<td>Store</td>
<td>Retail, wholesale</td>
<td>Small retail</td>
<td>1 story, 22,500 ft(^2) (2090 m(^2)); DX cooling and gas furnace</td>
</tr>
<tr>
<td>Inpatient Hospital</td>
<td>Hospital</td>
<td>Hospital</td>
<td>5 stories with basement, 241,500 ft(^2) (22,436 m(^2)); chilled water and hot water (from gas-fired boiler)</td>
</tr>
</tbody>
</table>

Each CRB model includes an HVAC system that is commonly used in the represented commercial building (Table 1). However, CBECS and RECS data indicate that there are many other existing HVAC systems serving each type of commercial buildings. The existing HVAC systems can be categorized based on the energy source for SH and SC. Table 2 lists different energy sources and their shares in the national site energy consumption for SH and SC of each group of buildings. These national-level contributions are determined based on the microdata of the most recent CBECS (EIA 2016b) and RECS (EIA 2013). As can be seen in Table 2, electricity is the exclusively predominant (with a more than 98% share) energy source for SC in both residential and commercial buildings. NG, heating oil, and propane are the predominant (with more than 80% combined share) energy sources for SH in commercial buildings. For single-family homes, the combined share of NG, heating oil, and propane is relatively lower (66.5%), and electricity contributes 28.1% to the total site energy consumption for SH, of which two-thirds is from heat pumps (mostly air-source heat pumps) and the rest from electric resistance heaters. Since electricity is used almost exclusively for SC, the main difference in terms of energy consumption among the various existing SC equipment types is their cooling efficiencies. The minimum code-compliant efficiencies specified in ASHRAE 90.1-2004 (ASHRAE 2004) were used in the representative CRB model to predict the existing SC equipment’s energy consumptions.

On the other hand, energy consumption for SH depends on both the energy source and the efficiency of the existing SH equipment. Since the minimum code-compliant efficiencies of boilers or furnaces that use NG, oil, or propane are very similar (around 80% Annual Fuel Utilization Efficiency according to ASHRAE 90.1-2004), the non-

---

\(^1\) The EER is the cooling capacity (in British thermal units [Btu]/hour) of the unit divided by its electrical input (in watts) at standard conditions.

\(^2\) The COP and EER are measured at AHRI/ISO/ASHRAE/ANSI 13256-1 rating conditions: for cooling at full capacity, entering fluid temperature is 77°F; for heating at full capacity, entering fluid temperature is 32°F.
electric heating equipment is modeled with the same heating efficiency. Therefore, two baseline SH systems were simulated for each representative commercial building—non-electric SH and electric resistance SH. Three baseline SH systems—non-electric, electric resistance, and air-source heat pump—were simulated for the representative residential building.

<table>
<thead>
<tr>
<th>Reference building</th>
<th>Energy source for SH</th>
<th>Contribution</th>
<th>Energy source for SC</th>
<th>Contribution</th>
</tr>
</thead>
<tbody>
<tr>
<td>Office</td>
<td>Electricity</td>
<td>9.5%</td>
<td>Electricity</td>
<td>99.4%</td>
</tr>
<tr>
<td></td>
<td>NG/oil/propane</td>
<td>90.5%</td>
<td>Other</td>
<td>0.6%</td>
</tr>
<tr>
<td>Lodging</td>
<td>Electricity</td>
<td>17.8%</td>
<td>Electricity</td>
<td>99.2%</td>
</tr>
<tr>
<td></td>
<td>NG/oil/propane</td>
<td>82.6%</td>
<td>Other</td>
<td>0.8%</td>
</tr>
<tr>
<td>Education</td>
<td>Electricity</td>
<td>3.8%</td>
<td>Electricity</td>
<td>98.6%</td>
</tr>
<tr>
<td></td>
<td>NG/oil/propane</td>
<td>96.1%</td>
<td>Other</td>
<td>1.4%</td>
</tr>
<tr>
<td>Store</td>
<td>Electricity</td>
<td>9.5%</td>
<td>Electricity</td>
<td>100%</td>
</tr>
<tr>
<td></td>
<td>NG/oil/propane</td>
<td>90.1%</td>
<td>Other</td>
<td>0.0%</td>
</tr>
<tr>
<td>Inpatient</td>
<td>Electricity</td>
<td>0.5%</td>
<td>Electricity</td>
<td>99.6%</td>
</tr>
<tr>
<td></td>
<td>NG/oil/propane</td>
<td>99.4%</td>
<td>Other</td>
<td>0.4%</td>
</tr>
<tr>
<td>Single family home</td>
<td>Electricity (resistance)</td>
<td>9.4%</td>
<td>Electricity</td>
<td>100%</td>
</tr>
<tr>
<td></td>
<td>NG/oil/propane</td>
<td>66.5%</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>Electricity (air-source heat pump)</td>
<td>18.7%</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

The energy consumptions of both the baseline HVAC systems and the GHPs applied in each of the representative buildings are predicted with eQUEST (Hirsch et al. 2016), a widely used building energy modeling program; it uses the most recent development of the DOE-2 program, which includes a module for simulating GHPs (Liu and Hellstrom 2006). The simulated baseline HVAC systems and the GHPs are sized to satisfy the SH and SC loads of each representative building and are operated with typical controls of these systems.

**Energy savings and other benefits in each county and nationwide**

Since the available county-level energy consumption data do not have information on the contribution of various existing HVAC systems in each county, the national shares of various SH systems (listed in Table 2) are used as weighting factors to account for different existing HVAC systems in each group of buildings. The average site energy saving percentage \((\text{Avg}_\text{ESPct}_SE)\) resulting from retrofitting existing HVAC systems with GHPs for a given type of building in a particular county is calculated based on the site energy consumption of each existing HVAC system \((S_{\text{sys}_SE})\) and the energy saving percentage \((\text{ESPct}_SE)\) of the simulated GHP system compared with each existing HVAC system, as expressed with Eq. (1).

\[
\text{Avg}_\text{ESPct}_SE(j,k) = \frac{\sum_{i=1}^{n} S_{\text{sys}_SE}(i,j) \cdot \text{ESPct}_SE(i,j,k)}{\sum_{i=1}^{n} S_{\text{sys}_SE}(i,j)} .
\]  

where \(i, j,\) and \(k\) are the indexes of baseline HVAC system, building type, and county, respectively.

The maximum achievable county-level annual site energy savings \((S_{\text{ES}})\) from GHP retrofits for a given group of buildings in a county is calculated by multiplying the corresponding total site energy consumption of existing HVAC systems \((SE)\) with \(\text{Avg}_\text{ESPct}_SE\), as expressed with Eq. (2).
\[ SES(j, k) = SE(j, k) \times \text{Avg\_ESPct\_SE}(j, k). \]  

The national annual site energy savings (\textit{National\_SES}) resulting from retrofitting existing HVAC systems with GHPs in all buildings is calculated with Eq. (3).

\[ \text{National\_SES} = \sum_{k=1}^{m} \sum_{j=1}^{n} SES(j, k). \]  

The savings in primary (source) energy, the reduction in CO\textsubscript{2} emissions, and the savings in energy cost are calculated following the same procedure. In these calculations, the annual site energy consumption of each existing HVAC system is replaced with the associated primary energy consumption, CO\textsubscript{2} emissions, or energy costs, which are converted from the site energy consumption data using corresponding conversion factors published by NREL (2007) and the 2014 state-level average electricity and natural gas prices (EIA 2016c, 2016d).

RESULTS AND DISCUSSIONS

Figure 2 shows the annual source energy saving percentages resulting from retrofitting the existing HVAC systems, which are included in the original CRB models, with GHPs at different locations (each representing a climate zone). As shown in Figure 2, the range and magnitude of source energy saving percentages vary widely by building type: 32–59% for a single-family home, -4–50% for offices, 18–41% for lodgings, 17–33% for schools (education), and -2–33% for stores. In general, more energy savings can be achieved at locations in cooler climates. These results indicate that the source energy saving percentages for residential buildings are higher than those for commercial buildings. The simulated residential building is conditioned continuously year-round (like most U.S. homes) and has less internal heat gain (e.g., from lighting and computers) than commercial buildings, which results in a higher ratio between SH and SC demands. It is thought to be the reason for the larger energy saving percentages of residential buildings. For buildings in hot climates (e.g., climate zones 2A and 2B), the energy savings in SH are very small; and the moderate energy savings in SC could be offset by the pumping energy of the GHPs.
Associated with source energy savings, GHP retrofits also reduce carbon emissions. With more renewable power in the mix of electricity production in the future—a trend that has already begun—the emission factor of electricity will decrease, which in turn can further reduce the carbon emissions of GHPs.

The simulation results also indicate that GHPs can reduce annual peak electricity demand for all the investigated buildings except those in very hot climates (e.g., climate zone 2B), or in very cold climates (e.g., climate zone 6B) and with existing non-electric SH system. Although GHPs could result in more electricity consumption (kWh) if they displace fossil fuels for SH, they still reduce summer peak electricity demand because of their higher cooling efficiencies than that of the conventional SC systems.

Figure 3 shows the combined source energy savings potential in both residential and commercial buildings in each county of the United States (not including counties in Alaska and Hawaii). The amount of source energy savings in each county (in Trillion Btu) is color-coded as shown in the legend. As Figure 3 shows, there are substantial energy savings potentials (>0.4 Trillion Btu) in most counties in the United States, and the northeastern region has more counties with high source energy saving potential (>2 Trillion Btu) than other regions.

Table 3 lists the annual site energy savings, source energy savings, carbon emission reductions, and energy cost savings in the residential and commercial sectors, respectively, and the sum of the two sectors. As shown in the table, retrofitting the residential sector with GHPs has three times more potential than retrofitting the commercial sector. Combining both residential and commercial sectors, GHP retrofits have a potential to save 5.7 quadrillion Btu of primary energy, avoid 356.3 million metric tons (Mt) of CO₂ emissions, and reduce energy costs by $49.8 billion, in each year.

The U.S. residential sector consumed 5.5 and 2.7 quads (10¹⁵ Btu or 2.93·10⁸ MWh) of primary energy annually for SH and SC, respectively, in 2009 (EIA 2013). The 2009 building energy data book (DOE 2009) stated that 2.3 and 2.2 quads of primary energy are consumed annually for SH and SC in commercial buildings. The 5.7 quads of primary energy savings from GHP retrofits can reduce the national primary energy consumption for SH and SC by 45%. If 20% of the technical potential can be realized, 1.1 quads of primary energy consumption will be avoided each year, which accounts for 9% of the current annual primary energy consumption in the entire building sector for SH and SC.
Table 3. Technical Potential of GHP Retrofits in the United States

<table>
<thead>
<tr>
<th>Categories</th>
<th>Annual site energy savings (quad Btu)</th>
<th>Annual source energy savings (quad Btu)</th>
<th>Annual carbon emission reductions (million Mt)</th>
<th>Annual energy cost savings (billion $)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Residential</td>
<td>3.3</td>
<td>4.3</td>
<td>271.1</td>
<td>38.2</td>
</tr>
<tr>
<td>Commercial</td>
<td>1.2</td>
<td>1.3</td>
<td>85.2</td>
<td>11.6</td>
</tr>
<tr>
<td>Total</td>
<td>4.5</td>
<td>5.7</td>
<td>356.3</td>
<td>49.8</td>
</tr>
</tbody>
</table>

CONCLUSIONS AND FUTURE WORK

This paper presents an analysis of the technical potential of GHP applications in both residential and commercial buildings in the United States. The analysis indicates that retrofitting existing conventional HVAC systems in U.S. residential and commercial buildings can result in significant energy savings and carbon emission reductions. The residential sector has three times more energy saving potential than the commercial sector. Combining both residential and commercial sectors, GHP retrofits have a potential to save 5.7 quadrillion Btu of primary energy, avoid 356.3 million metric tons (Mt) of CO₂ emissions, and reduce energy costs by $49.8 billion, in each year. Given this huge energy savings potential, GHP could be a key component of national energy and climate change mitigation strategies.

High initial cost is the biggest barrier for GHP deployment, but it may be overcome by breakthroughs in the following areas:
- Lower-cost ground heat exchangers and customized drilling techniques/equipment for GHP
- Better design of GHPs as a result of more information on the ground formation
- Volume manufacturing of GHP equipment
- Financial incentives or third party financing
- Integration of GHPs as a part of utility infrastructure in new developments

ACKNOWLEDGMENTS

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NOMENCLATURE

\[ E_{SPct\_SE} = \text{regional average annual site energy saving percentage of the GHP system (\%)} \]
\[ S_{Sys\_SE} = \text{national annual site energy consumption of baseline system (kBtu)} \]

Subscripts

- \( i \) = index of baseline HVAC system
- \( j \) = index of building type
- \( k \) = index of county
REFERENCES


- A Survey Based on the Swedish Well Database

Kristina Juhlin, BSc  
Signhild Gehlin, PhD

ABSTRACT
The Geological Survey of Sweden (SGU) is the Swedish government agency responsible for groundwater, geological and mineral management in Sweden. SGU provides open access geological data on rock, soil and groundwater conditions. Since 1978 over 600,000 wells (water wells, GSHP boreholes, etc) have been registered in the SGU Well Database, with around 20,000 new registrations per year.

Sweden is one of the leading countries in the world in developing and using ground source heat pump (GSHP) technology. Of the more than 600,000 registered wells, roughly 320,000 wells are registered as GSHP boreholes. The vast majority of these GSHP boreholes are single boreholes for single-family buildings. The number of large GSHP systems with 20 boreholes or more, is estimated to 300-350 sites.

This paper uses data from the SGU Well Database to quantify and analyze the number of vertical GSHP systems reported between 1978-2015, with special focus on GSHP systems with 20 or more boreholes. Results are shown from the development of larger vertical GSHP system installments over the years, number of registrations per year, system size, average well depth, and geographical distribution.

INTRODUCTION
Since the oil crisis in the late 1970s, Sweden has been one of the world leading countries in development and use of ground source heat pumps (GSHPs). Sales statistics from heat pump and ground heat exchanger manufacturers indicate that up till today, some 500,000 GSHP systems, vertical and horizontal, small and large, have been installed in Sweden over the years (Gehlin et al, 2015).

From 1978 The Geological Survey of Sweden (SGU) has administered an open access database on drilled wells in Sweden. The SGU Well Database was a result of a new regulation from 1975 (SFS 1975: 424, SFS 1985: 245). By law, all wells installed must be registered with the Geological Survey of Sweden. The law states that it is the well drillers who are obliged to submit data on the installed wells to SGU. In Sweden there is no law that says that the borehole must be grouted. Therefore almost all wells in Sweden are groundwater-filled, with casing a few meters into the bedrock, due to the cost difference. This is also why the boreholes are referred to as wells in the SGU database. Consultants and others who undertake investigations of wells are also obliged to submit data to the database. The database provides information to well drillers and other people interested in e.g. soil depth, which can be associated with construction of drilled wells.

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As the law to register water wells was established in 1975, wells drilled prior to 1975 are typically not found in the Well Database. SGU acknowledges that although the number of un-registered new wells are believed to decrease with easier registration procedures and increasing awareness of the regulation and the database, there are still a large number of drilled wells missing in the database. By the end of 2015, more than 600 000 wells had been registered in the database, but due to delays in registration or neglected registrations, some 20% of all existing wells in Sweden are believed to be missing in the database (Gierup 2016).

Before the digital era all new registrations were sent to SGU by mail in paper form. The companies typically accumulated all drilling protocols from one year before sending them to SGU. Today well registrations to SGU are usually made digitally in association with the drilling project. SGU built the application for online registration in 2011, but even before that it was possible to register digitally. In the past couple of years around 60% of the new well registrations were done digitally (Gierup 2016). Figure 1 shows the average time passed between the drilling of a well and the date when the well was registered in the database. The diagram is based on all wells registered to SGU, not just energy wells. The delay in time has decreased thanks to the online registration application. Figure 2 shows the number of new wells drilled each year since 1980. The big wave of GSHP system installations in the early 2000’s was due to high oil prices in Sweden (Barth et al. 2012). At that time around 30 000 wells were installed annually. Today around 20 000 wells are registered annually (SGU 2016).

In the SGU Well Database, well types are categorized by their purpose; individual water use (household, cottages, smaller farms), irrigation wells, industrial water wells, observation wells, wells for large agricultural water use, community water use (at least 10 households), energy wells (heat and/or cooling), and other usage (SGU 2015 a). The focus in this paper is on wells drilled for energy extraction or injection. These wells are identified with the capitals ENE in the database. By the end of 2015 around 320 000 wells were registered as energy wells in the database. Only vertical GSHP systems (extracting heat or cold from groundwater or ground) are registered in the database. No horizontal GSHP systems are included.

In 2013 Andersson et al. (2013) estimated the number of large systems with at least 20 active wells per site as 400 systems in Sweden. Compared to the new aggregated SGU Well database data in this study the number of larger vertical GSHP systems in Sweden at the time was overestimated. Around 300 registered facilities with 20 wells or more were identified in the SGU Well Database in this study. However, this number is likely to be underestimated due to neglected and delayed registrations. Given Gierup’s (2016) estimate that 20% of Swedish wells are missing from the database, the Andersson, et al. (2013) estimate of about 400 systems with 20 or more wells seems reasonable.

In this study, the focus has been on vertical GSHP systems with at least 20 wells. This system size was chosen as that would be the size required for larger facilities such as hospitals, schools, commercial buildings, large residential
buildings etc.

**METHOD**

From the SGU’s Well Database some 356,463 energy wells were extracted. Of these wells, 9368 wells belong to systems with 20 or more boreholes, at a total of 292 sites. The following information was extracted for each of these sites:

- Geographical region
- Name of the property
- Number of wells
- Individual well depth (m)
- Total depth of all the wells at the site (m)
- Year of installation

For a majority of the wells there is information on property name, year of installation, number of wells at site, depth of individual well, SWEREF 99 TM coordinates etc. The SWEREF 99 TM are coordinates that give you the exact latitude and longitude and are used in Sweden at a national level, dividing Sweden into twelve regions (Lantmäteriet 2016). This makes the coordinates more precise than using all of Sweden as one region. The only information that was missing from a few systems was the year of registration or the coordinates.

The data collected from the SGU Well Database was compared with information provided by drilling companies, GSHP contractors, and property owners. By comparing the SGU data with the information provided by the companies, some missing data could be added. Some of the systems that were listed by drilling companies were not found in the SGU’s Well Database. In some cases this proved to be explained by the fact that for some larger sites the borefield was registered in portions at different times, and then show up as separate projects in the database.

**RESULTS**

The compiled list with 292 vertical GSHP systems with at least 20 active wells, extracted from the SGU’s Well Database was used for studying the historical development of larger vertical GSHP systems in Sweden, with regard to number of registrations per year, system size, average well depth, and geographical distribution.

**Number of New Installations per Year**

During the first twenty years (1978-1998) of well registrations, between 1-3 new larger GSHP systems were registered annually. Over the last decade, however, between 20 and 30 new larger GSHP systems have been registered annually. The decreasing numbers in 2014-2015 are at least partly due to the delayed registration to SGU Well Database (Figure 3).
Figure 3 shows the growth of the market for larger vertical GSHP systems during the 2000's. With more knowledge in technology, increased interest in environmental certification for a property, and a cheaper energy solution the interest for installing vertical GSHP systems has grown in Sweden (Barth, et al. 2012).

Size of Vertical GSHP System

Not only has the number of larger GSHP systems installed per year increased, but there is also a trend towards increasing system size. The largest vertical GSHP system in Sweden today is located at Karlstad University Campus, and was completed in 2014 (Gehlin, et al. 2015, Akademiska Hus 2014). 203 wells were drilled with a total of 48,000 drilled meters.
In the past years, most larger vertical GSHP systems have had between 20-30 wells per site (Figure 4). The number of wells per site has grown and today the average size of a larger vertical GSHP system registered in the well database is around 60 wells per site (SGU 2015 b).

**Depth of Well Drilling**

In the 1980’s and 1990’s boreholes for GSHP systems were rarely drilled to a depth exceeding 180 m. Since then, drilling equipment and drilling technique have developed rapidly, making possible significant increases in drilled well depth. Today boreholes for larger GSHP systems typically reach 250 m depth, and even 300 m deep boreholes are becoming more frequent for these systems. Figure 5 shows the average borehole depth for GSHP systems in Sweden over time. The average borehole depth for all sizes of GSHP systems has increased from around 120 m in the 1980’s to 192 m in 2015. For GSHP systems with 20 boreholes or more, the average borehole depth has increased from around 170 m in the 1980’s to 230 m in 2015. Boreholes for GSHP systems with 20 boreholes or more account for 0,03 % of the total number of boreholes of all sizes of GSHP systems in the SGU database.

**Figure 4** Development of GSHP system size over the years.

**Figure 5** Development of borehole depth for vertical GSHP systems in Sweden.
Geographical distribution

The geographical distribution of larger vertical GSHP systems in Sweden is uneven. The regions with the largest cities, such as Stockholm County, Uppsala County and Västra Götaland County, have the greatest proportion of larger GSHP systems than the rest of the country (Figure 6).

The largest number of energy wells are located in the more densely populated regions in Sweden, such as the Stockholm area. In the Stockholm region there are over 4,000 wells (over 40% of all GSHP systems) spread over 130 sites with larger vertical GSHP systems. The two regions with the second largest number of vertical GSHP systems, Uppsala and Västra Götaland, have around 30 installations with a total of 850 wells in each region. They each correspond to around 10% of all installed systems.

Figure 7 confirms that the majority of the larger vertical GSHP systems are found around the Stockholm area. However, the largest installations, for example Sweden’s largest Karlstad University Campus with 203 wells, are also found in other regions. The Karlstad University Campus GSHP system does not show as a yellow circle in Figure 7, which may be due to registration of the borefield in separate sections, so that the total system does not register as one large system in the SGU’s Well Database.
While the number of large vertical GSHP systems increases in the more densely populated areas, such as Stockholm, there is also an increasing trend in the other regions (Figure 8a, 8b). In Stockholm County there is now around 130 installed vertical GSHP systems with at least 20 wells per site, and in Jönköping County 11 larger GSHP systems are registered.

**Figure 8a** Registered large GSHP systems in the less populous counties of Sweden.  
**Figure 8b** Registered large GSHP systems in most populous counties of Sweden.

**Type of Building with Installed GSHP System**

When comparing building types for the large GSHP systems, most of the registered systems are serving residential buildings (Figure 9). In the last year residential buildings, mostly apartment buildings with many apartments, have installed more large vertical GSHP systems (Figure 10). Commercial buildings such as hardware stores, local business etc. is the second largest group. The public sector (hospitals, schools etc.) is the smallest fraction.

**Figure 9** Building type served by a large vertical GSHP system.  
**Figure 10** Trend of building type served by large vertical GSHP system over the years.
CONCLUSION

Sweden was one of the pioneering countries in the development and use of geothermal energy in the 1970s and 80s. This new statistical data on larger (>20 wells) vertical GSHP systems in Sweden gives an overview of how larger GSHP systems have been established in different regions in Sweden, and how the GSHP market has developed in recent years.

This study shows that the trends of size, number, and geographical distribution of larger vertical GSHP systems are increasing in the country. The trend of installing large vertical GSHP systems started in the early 2000’s but it is only in the past ten years the number of systems have increased to above 25 new installations per year.

The study also gives an idea of the types of buildings that use large vertical GSHP systems. Residential, public, and commercial buildings have all increased in their use of GSHP systems, and the largest increase is seen for the residential sector.

There are three main sources of error in this study: missing data from early development, registration not submitted, and delayed registration. The processing of the well database data on energy wells also shows that several of the well-known systems with more than 100 boreholes are not registered as one project, but as several portions of the borehole field. With the new web application for registering online, the quality of the data may improve over time.

For further studies, similar data can be collected from other countries well archives, especially in Europe, to compare the evolution and usage of larger vertical GSHP system for that country.

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Assessment of the Gradient of an Objective Function by Analytical Derivation for Optimization-Based Design of Ground-Coupled Heat Pump Systems

Bernard Dusseault Philippe Pasquier Denis Marcotte

ABSTRACT
Optimization-based design of ground heat exchangers requires derivation of the objective function with respect to the design parameters, which is usually done through finite-differentiation of the cost or utility function. The approach is however prone to approximation errors and can result in convergence issues or long optimization time. By deriving analytically the ground heat exchanger transfer function, it is possible to obtain an exact representation of the objective function gradient and avoid numerical instabilities. To illustrate the advantages of using analytical expressions, a common design task is expressed as an optimization problem. It is shown that by using an analytical derivation of the gradient in conjunction with strong Wolfe conditions during a line search may reduce significantly computation time by comparison to a finite-differentiation of the gradient.

INTRODUCTION
Two approaches are commonly used to design a ground-coupled heat pump system. The first one relies on sizing equations, like the one suggested by ASHRAE (Kavanaugh and Rafferty, 2014; Philippe et al., 2010; Bernier et al., 2008), while the second approach consists of iteratively using a simulation method to optimize a cost or utility function. The optimization process can be achieved by trial and error, which can be cumbersome, or automated through nonlinear optimization algorithms (Retkowski and Thöming, 2014; Huang et al., 2015; Hénault et al., 2016). The latter however requires the computation of the derivative of the objective function with respect to the $n$ design parameters in order to find a suitable descent direction.

The objective function gradient is usually computed by finite difference through $n+1$ simulations of the ground-coupled heat pump system. Although being easy to implement, computation of objective function gradient through finite differenciation often leads to inaccurate estimations of the descent direction, which may significantly increase solution time. The objective of this paper is to present two efficient computational approaches to derive the gradient of an objective function corresponding to a common design task encountered by designers.
METHODOLOGY

For a ground heat exchanger (GHE) composed of \( n_b \) boreholes of length \( H \), the mean fluid temperature \( \bar{T}_f \) circulating in the boreholes can be described by:

\[
\bar{T}_f(t) = T_g + \frac{\hat{Q}}{Hn_b}R_b + (f \ast g)(t)
\]

where \( T_g \) is the initial ground temperature, \( \hat{Q} \) is the total ground load exchanged by the GHE and \( R_b \) is the equivalent borehole resistance (Gehlin, 2002; Marcotte and Pasquier, 2008). The last term represents the temperature perturbation at the borehole wall and is obtained by convolving the incremental heat flux signal \( f \) given by:

\[
f(t_i) = \frac{\hat{Q}(t_i)}{Hn_b} - \frac{\hat{Q}(t_{i-1})}{Hn_b}
\]

with the response function \( g \) (Eskilson, 1987). Note that as indicated by Marcotte and Pasquier (2008) and Pasquier and Marcotte (2013), for \( i = 0 \) in Eq. 2, a zero value should be used for \( \hat{Q}(t_{i-1}) \).

**Construction of the Transfer Function \( g \)**

The GHE response function \( g \) synthesize the thermal behavior of the GHE and integrates the thermal properties of the underground, but also the coordinates and length of the boreholes composing the GHE. Although many methods can be used to construct \( g \), the approach chosen in this article is based on the work of Marcotte and Pasquier (2014) because of its efficiency and flexibility. The approach, itself inspired by the works of Lamarche (2009), Cimmino et al. (2013) and Lazzarotto (2014), consists to find sequentially the heat transfer rates \( q_i \), emanating from the boreholes of a GHE using a linear system of equations expressed, for simplicity, in a compact matrix notation. For a parallel arrangement of the boreholes, the system corresponding to the mean temperature at the borehole wall is given by:

\[
\begin{bmatrix}
G & 1 \\
1^T & 0
\end{bmatrix}
\begin{bmatrix}
q_i \\
-\Delta \bar{T}_b
\end{bmatrix} =
\begin{bmatrix}
h \\
1
\end{bmatrix}
\]

(3)

where \( q_i \) is a \( n_b \times 1 \) vector containing the heat load emitted by each borehole for the current time step, \( h \) is a \( n_b \times 1 \) vector containing the historical temperature perturbations, \( 1 \) is a \( n_b \times 1 \) vector of ones and \( G \) is the following interaction matrix:

\[
G =
\begin{bmatrix}
G_{11} & G_{12} & \cdots & G_{1n_b} \\
G_{21} & G_{22} \\
\vdots & \ddots \\
G_{n_b1} & \cdots & G_{n_bn_b}
\end{bmatrix}
\]

(4)

In this work, matrix \( G \) is constructed using the finite line-source (FLS) model of Claesson and Javed (2011) with a heating load of 1 W/m at radial distances \( r_j \) and at an evaluation time corresponding to the current time step value. Thus, the \( G_{ij} \) are given for a borehole of length \( H \), thermal diffusivity \( \alpha \) and buried depth \( D \) by

\[
G_{ij} = \frac{1}{4\pi k} \int_0^\infty \int_{-a}^a e^{-\frac{r^2}{4\alpha t}} Y(Hs, Ds) \frac{Hs}{Hs + d} ds dr
\]

(5)

with

\[
Y(b,d) = 2\text{erf}(b) + 2\text{erf}(b + 2d) - \text{erf}(2b + 2d) - \text{erf}(2d)
\]

(6)
Before solving Eq. 3 for $q_j$ and $\Delta T_j$, the historical vector $h$ has to be built. For time step $m$, the latter is obtained by solving the following convolution product at $t = m\Delta t$:

$$h_j(m\Delta t) = (\tilde{f}_j \ast \tilde{g})(m\Delta t)$$

(7)

where the incremental heat load vector for borehole $j$ is given by:

$$\tilde{f}_j(t) = q_j(t) - q_j(t_{i-1})$$

(8)

and where the transfer function $\tilde{g}$ is the borehole response for the first $m$ time steps obtained under a unit load for each $r_j$ radial distance. Since $b$ has to be calculated at each time step, Eq. 7 must be solved for $\Delta T_j$ sequentially as well by using all the previous values of $q_j$ and $G$. Finally, the transfer function corresponding to the mean temperature at the borehole wall used in Eq. 1 is obtained simply by:

$$g(t) = n_b \Delta T_j(t)$$

(9)

**A Simple Objective Function**

A common task performed by GHE designers consists to identify the total underground loop length keeping the fluid temperature within the heat pump operation range and to keep the GHE cost as low as possible. If the building is a heating dominant one, the total loop length ($n_b H$) chosen by the designer should be sufficient to maintain the fluid temperature just above the heating temperature limit ($T_{\text{min}}$) of the heat pump, while minimizing, for a fixed $n_b$ value, the borehole length. From an optimization perspective, such situation is easily described by an objective function describing the gap between the minimum fluid temperature given by Eq. 1 and $T_{\text{min}}$ through:

$$F = \left( \min(T_j(t)) - T_{\text{min}} \right)^2$$

(10)

Since $F$ is a function of $R_b, H, n_b, T_{\text{min}}$ and the borehole coordinates, minimizing $F$ is in fact a multidimensional problem. For the sake of keeping this case study simple and instructive, only the borehole length $H$ is used as design parameter, thus allowing to reduce the problem to a unidimensional one with a well-defined global minimum. The next section will now focus on finding the gradient of this objective function according to $H$ with three different approaches.

**Analytical derivation of the gradient**

The gradient of $F$ with respect to the design parameter is obtained by computing the derivative of $F$ with respect to $H$. This leads to:

$$\frac{\partial F}{\partial H} = 2\left( \min(T_j(t)) - T_{\text{min}} \right) \frac{\partial \min(T_j(t))}{\partial H}$$

(11)

To evaluate the derivative of $T_j$ according to $H$, Eq. 1 should be derived as well. Now, dropping the min and notation to simplify the writing, the derivative is evaluated at the time during where the minimum in the fluid temperature occurs:

$$\frac{\partial \min(T_j(t))}{\partial H} = -\frac{\dot{Q}}{H^2 n_b} - \left( \frac{\partial f}{\partial H} \ast \tilde{g} \right)(t) + \left( f \ast \frac{\partial \tilde{g}}{\partial H} \right)(t)$$

(12)

with

$$\frac{\partial f}{\partial H} = -\frac{\dot{Q}(t)}{H^2 n_b} - \frac{\dot{Q}(t_{i-1})}{H^2 n_b}$$

(13)
and

\[ \frac{\partial g}{\partial H} = \eta_b \frac{\partial \Delta \tilde{T}_b}{\partial H} \] (14)

Now, the derivative of the transfer function with respect to \( H \) is needed to solve Eq. 12 and it is necessary to derive Eq. 3 as well. Differentiating Eq. 3 and solving for \( -\partial \Delta \tilde{T}_b / \partial H \) (see Petersen and Pedersen, 2012) leads to:

\[
\begin{bmatrix}
\partial q / \partial H \\
-\partial \Delta \tilde{T}_b / \partial H
\end{bmatrix} = \begin{bmatrix}
G & 1 \\
1 & 0
\end{bmatrix}^{-1}
\begin{bmatrix}
\partial G / \partial H \\
0
\end{bmatrix}
\begin{bmatrix}
G & 1 \\
1 & 1
\end{bmatrix}^{-1}
\begin{bmatrix}
h \\
1
\end{bmatrix}
+ \begin{bmatrix}
G & 1 \\
1 & 0
\end{bmatrix}^{-1}
\begin{bmatrix}
-\partial h / \partial H
\end{bmatrix}
\]

(15)

Since all the terms in \( G \) were obtained using the FLS model of Cleasson and Javed (2011), their formulation has to be derived as well according to \( H \) and this gives:

\[
\frac{\partial G_{ij}}{\partial H} = \frac{1}{4\pi k} \int \frac{e^{-r^2 z^2}}{\sqrt{\pi \omega}} \left( \frac{Y(Hs, Ds)}{Hs^2 z^2} + \frac{Z(Hs, Ds)}{Hs} \right) dz
\]

(16)

with

\[
Z(b, d) = 2erf(b) + 2erf(b + 2d) - 2erf(2b + 2d)
\]

(17)

Finally, Eq. 15 to 17 are solved sequentially to construct \( \frac{\partial g}{\partial H} \) which requires computation of

\[
\frac{\partial h_{ij}}{\partial H} (m\Delta t) = \left( \tilde{f}_j * \frac{\partial G_{ij}}{\partial H} \right) (m\Delta t)
\]

(18)

**Finite-differentiation of the gradient**

The finite-difference method involves a simple and well known algorithm commonly used for the calculation of gradients. In this article, a forward finite difference scheme is used to approximate the derivative of the objective function \( F \) according to \( H \) between \( H \) and \( H + \varepsilon \) :

\[
\frac{\partial F}{\partial H} = \frac{F(H + \varepsilon) - F(H)}{\varepsilon}
\]

(19)

**Semi-analytical derivation of the gradient**

Evaluating sequentially Eq. 15 to 18 can be cumbersome and to avoid this task, an hybrid approach relying both on finite-difference and analytical derivation is also presented. The semi-analytical derivation of Eq. 10 consists to solve Eq. 3 for \( \Delta \tilde{T}_b \) with \( H \) and \( H + \varepsilon \) to evaluate \( \frac{\partial \Delta \tilde{T}_b}{\partial H} \) (Eq. 14) by finite difference through:

\[
\frac{\partial \Delta \tilde{T}_b}{\partial H} = \frac{\Delta \tilde{T}_b(H + \varepsilon) - \Delta \tilde{T}_b(H)}{\varepsilon}
\]

(20)

Once the solution of Eq. 20 is obtained, Eq. 11 to 14 are solved to obtain the objective function gradient. Note that the terms involving \( \tilde{Q}(t_i) / Hn_b \) are derived analytically. We will show that with this approach, the implementation is simpler but still retains the precision of the analytical approach.
A SIMPLE CASE STUDY

To assess the precision of the gradient computed by the three methods presented earlier, Eq. 1 has been used to simulate over a 10 year-period the mean fluid temperature circulating in a GHE made of 25 boreholes embedded in a soil having a ground thermal conductivity $k$ of 2.5 W/mK, a thermal diffusivity $\alpha$ of 1.25e-6 m²/s and a ground temperature $T_g$ of 10 °C. The boreholes have an equivalent resistance $R_b$ of 0.1 mK/W, are located on a regular grid with spacing of 10 and 5 m at a depth $D$ of 2 m. The ground loads illustrated in Figure 1 were used to solve Eq. 1

The objective function and its gradient as a function of borehole length

Figure 2a below presents a sweeping of the possible values of $F$ and $dF/dH$ for $H$ values comprised between 30 m to 300 m. We can see that these functions vary by several orders of magnitude with a clear minimum around $H = 146$ m. Figure 2b also illustrates easily the discrepancies that exists between the gradient computed by finite differentiation and the gradient computed with an analytical or semi-analytical approach.

When looking at Figure 2b, one can see that smaller differences also exist between the analytical and semi-analytical methods. The magnitude of these discrepancies is grasped easily when looking at Table 1 which illustrates $F, dF/dH$ and the associated computation time for two different $H$ values. The first value was chosen to be as close as possible from the optimum of $F(145.363)$ while the second $H$ value (250 m) corresponds to a poor design. As shown in Table 1, the objective function values are equal for a given $H$. This is due to the fact that no matter which method is used for the calculation of the gradient, the objective function is always obtained the same way using the sequential method based on Marcotte and Pasquier (2014). However, the gradients computed by the analytically-based methods are up to 30% higher. For all six scenarios, 100 runs were completed to get a proper evaluation of the calculation times of the gradients by all three methods. As shown in Table 1, the analytical derivation method is systematically faster than the other two by at least 2 % with our current MATLAB-based code. This is due to the shear fact that equivalent calculations that do not involved epsilons are compiled faster in our code.
Figure 2: a) Objective function $F$ for various borehole lengths and b) gradients evaluated with the methods tested in this work with $\epsilon = 10^{-6}$.

Table 1. Objective Function Gradient for $H = 145.363$ and $250.000$ m ($\epsilon = 10^{-6}$)

<table>
<thead>
<tr>
<th>Method Used</th>
<th>$H$ (m)</th>
<th>Objective Function ($^\circ$C²)</th>
<th>Gradient Value ($^\circ$C²/m)</th>
<th>Computation Time (s)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Finite Difference</td>
<td>145.363</td>
<td>7.835 $\times 10^{-10}$</td>
<td>-5.94 $\times 10^{-6}$</td>
<td>0.386</td>
</tr>
<tr>
<td>Semi-analytical Derivation</td>
<td>145.363</td>
<td>7.835 $\times 10^{-10}$</td>
<td>-4.141 $\times 10^{-6}$</td>
<td>0.400</td>
</tr>
<tr>
<td>Analytical Derivation</td>
<td>145.363</td>
<td>7.835 $\times 10^{-10}$</td>
<td>-4.621 $\times 10^{-6}$</td>
<td>0.369</td>
</tr>
<tr>
<td>Finite Difference</td>
<td>250.000</td>
<td>24.809</td>
<td>0.2445</td>
<td>0.379</td>
</tr>
<tr>
<td>Semi-analytical Derivation</td>
<td>250.000</td>
<td>24.809</td>
<td>0.3149</td>
<td>0.399</td>
</tr>
<tr>
<td>Analytical Derivation</td>
<td>250.000</td>
<td>24.809</td>
<td>0.2797</td>
<td>0.372</td>
</tr>
</tbody>
</table>

OPTIMIZATION-BASED DESIGN USING A LINE SEARCH STRATEGY

An accurate estimation of gradient is almost useless when used without an optimization algorithm able to minimize the objective function. Therefore, to illustrate the advantages of using analytical gradients for optimization-based design, the various gradients were integrated within an optimization routine. Here, since the design problem involves only $H$, the problem reduces to a unidimensional line search. A line search algorithm using the strong Wolfe conditions on curvature, as implemented in minFunc (Schmidt, 2005), was used to identify the near-optimum $H$ value.

Table 2 and Table 3 provide the results for two different initial solutions of $H$ using a $\epsilon$ value of $10^{-6}$ and $10^{-10}$ respectively. The optimization algorithm was limited to 5024 iterations to limit computation time, although it didn’t prove to be necessary. All presented values of $H$ were limited to seven digits.

Looking at the results in both Table 2 and Table 3, three conclusions can be drawn. First, although the convergence rate all methods are influenced by $H_0$, when the initial guess is too high compared to the optimal value, only the analytical-based derivation methods converge every time. Secondly, both the methods that implies a finite differentiation varies significantly in number of iterations depending on starting conditions. This is due to the fact that the convergence of these two methods relies heavily on the $\epsilon$ value selected. This is especially true when the gradient to be evaluated is low and steady, associated with high initial values of H according to figure 2b, and when the epsilon...
is small. This implies that either a convergence analysis must be done or a safe low initial guess must be used in the optimization for the finite difference and semi-analytical methods in this particular case.

Thirdly, both the proposed analytical and the semi-analytical derivation methods converge way quicker than the finite difference method and in a lower number of iterations. On average, the semi-analytical method is 1.75 times faster while the analytical derivation method is 2 times faster. This is due to the fact than using a more precise gradient of the objective function allows for a much more rapid convergence. Indeed, Figure 3 illustrates the $H$ values used during the line search for the methods presented earlier and one can easily see the advantage of using gradients derived analytically.

Figure 1b shows the simulated fluid temperature for $H$ equals to 50, 145.363 and 250 m. The horizontal red line underneath the fluid temperature corresponds to the minimum fluid temperature of -2°C that should be reached by the fluid to minimize the objective function $F$. It is clear that for $H$ values of 50 and 250 m, the fluid temperature is far from -2°C. However, for $H$ corresponding to the optimum, the minimum occurs during the last winter and matches almost perfectly the red line.

Figure 3 – Evolution of $H$ during the optimization-based design; initial guess $H = 50$ m, $\varepsilon = 10^{-6}$.

Overall, both analytical-based methods proved to be much faster and effective compared to the finite difference. The analytical derivation is more precise and allows for the fastest convergence and lowest number of iterations but requires more implementation time since it needs the calculation of a derived version of the FLS model and associated matrix. The semi-analytical method is almost as easy to implement as the common finite difference while allowing for a much faster rate of convergence. However, since it still requires to perform a finite difference on the transfer function of the GHE, the semi-analytical approach still requires the user to select more carefully its initial $H$ solution and epsilon before launching the optimization process.

<table>
<thead>
<tr>
<th>Method Used</th>
<th>Initial solution</th>
<th>$H$ (m)</th>
<th>Objective Function ($°C^2$)</th>
<th>Iterations</th>
<th>Computation Time (s)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Finite Difference</td>
<td>50</td>
<td>145.3626</td>
<td>3.58 $E^{-9}$</td>
<td>39</td>
<td>21.94</td>
</tr>
<tr>
<td>Semi-analytical Derivation</td>
<td>50</td>
<td>145.3644</td>
<td>8.03 $E^{-9}$</td>
<td>18</td>
<td>10.41</td>
</tr>
<tr>
<td>Analytical Derivation</td>
<td>50</td>
<td>145.3636</td>
<td>4.07 $E^{-10}$</td>
<td>18</td>
<td>9.93</td>
</tr>
<tr>
<td>Finite Difference</td>
<td>250</td>
<td>145.7234</td>
<td>1.07 $E^{-12}$</td>
<td>29</td>
<td>16.16</td>
</tr>
<tr>
<td>Semi-analytical Derivation</td>
<td>250</td>
<td>145.3655</td>
<td>3.03 $E^{-8}$</td>
<td>15</td>
<td>8.72</td>
</tr>
<tr>
<td>Analytical Derivation</td>
<td>250</td>
<td>145.3633</td>
<td>2.95 $E^{-11}$</td>
<td>22</td>
<td>12.20</td>
</tr>
</tbody>
</table>
Table 3. Line search results for two initial solutions with $\varepsilon = 10^{-10}$.

<table>
<thead>
<tr>
<th>Method Used</th>
<th>Initial solution</th>
<th>$H_0$ (m)</th>
<th>$H$ (m)</th>
<th>Objective Function ($^\circ$C$^2$)</th>
<th>Iterations</th>
<th>Computation Time (s)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Finite Difference</td>
<td>50</td>
<td>51.0672</td>
<td>51.0672</td>
<td>4.5218 E$^2$</td>
<td>36</td>
<td>20.82</td>
</tr>
<tr>
<td>Semi-analytical Derivation</td>
<td>50</td>
<td>145.3637</td>
<td>145.3637</td>
<td>9.69 E$^{10}$</td>
<td>23</td>
<td>14.10</td>
</tr>
<tr>
<td>Analytical Derivation</td>
<td>50</td>
<td>145.3636</td>
<td>145.3636</td>
<td>4.07 E$^{10}$</td>
<td>18</td>
<td>9.93</td>
</tr>
<tr>
<td>Finite Difference</td>
<td>250</td>
<td>-</td>
<td>-</td>
<td>Did not converge within 5 minutes</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Semi-analytical Derivation</td>
<td>250</td>
<td>145.3647</td>
<td>145.3647</td>
<td>9.99 E$^9$</td>
<td>40</td>
<td>24.54</td>
</tr>
<tr>
<td>Analytical Derivation</td>
<td>250</td>
<td>145.3633</td>
<td>145.3633</td>
<td>2.95 E$^{11}$</td>
<td>22</td>
<td>12.20</td>
</tr>
</tbody>
</table>

CONCLUSION

In this article, we demonstrated that using a finite-difference based algorithms to evaluate the gradient of an objective function can lead to some approximation errors. When using such gradient in an optimization algorithm, even a simple unidimensional line search problem may have convergence issues or longer convergence time. By deriving analytically an objective function with respect to the design parameter of interest, a formulation that provides an accurate estimation of the gradient can be found. Using an analytical gradient in conjunction with strong Wolfe conditions may reduce computation time significantly.

NOMENCLATURE

- $D$: Buried depth of the boreholes (m)
- $f, f'$: Step increment vector (W/m)
- $F$: Objective function ($^\circ$C$^2$)
- $g, g'$: Analytical model response ($^\circ$C·m/W)
- $G$: Matrix whose element $(i, j)$ is the temperature perturbation caused at borehole $i$ by the heat emitted by borehole $j$ ($^\circ$C)
- $h$: Vector containing the temperature perturbations due to the historical part ($^\circ$C)
- $H$: Length of the boreholes (m)
- $k$: Thermal conductivity of the ground (W/(m·°C))
- $n_b$: Number of boreholes in the GHE (-)
- $q_i$: Vector of heat flux (W/m)
- $\dot{Q}$: Total ground load exchanged by the GHE (W)
- $r$: Distance between boreholes $i$ and $j$ (m)
- $R_b$: Borehole equivalent thermal resistance (m·°C/W)
- $t$: Time (s)
- $T_f$: Mean fluid temperature ($^\circ$C)
- $T_g$: Initial temperature of the ground ($^\circ$C)
- $T_{min}$: Minimum fluid temperature ($^\circ$C)
- $\alpha$: Thermal diffusivity (m$^2$/s)
- $\Delta T_b$: Transfer function corresponding to the mean temperature at the borehole wall ($^\circ$C/W/borehole)
- $\varepsilon$: Increment used to perform finite difference (m)

Subscripts

- $b$: Borehole
- $g$: Ground
- $i, j$: Position
- $m, t$: Time step
REFERENCES


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Analytical Solution for Optimal Mass Flow Rate in Primary Circuit of Ground-Coupled Heat Pump Systems

Damien Picard  
Filip Jorissen  
Lieve Helsen

ABSTRACT

Ground source heat pump (GSHP) systems extract heat or cold from the ground by circulating a heat carrier fluid (HCF) in a ground heat exchanger and inject this energy in buildings. This paper shows that there exists an optimal HCF flow rate which minimizes the energy use of such systems. The paper proposes an analytical solution for the optimal flow rate as a function of measurable variables, system parameters and data that can easily be derived from manufacturer data sheets. The analytical solution is validated using a detailed simulation model representing an existing GSHP system of 99 boreholes with a depth of 30m.

INTRODUCTION

Ground source heat pump (GSHP) systems extract heat or cold from the ground by circulating a heat carrier fluid (HCF) in a ground heat exchanger and inject this energy in buildings. Despite relatively high investment costs and thanks to their high energy efficiency, GSHP systems have proven their economic viability with about $10^5$ units sold every year in Europe between 2005 and 2013 (Nowak et al., 2014). Numerous studies and tools have been proposed to optimize the design of GSHPs in order to reduce the investment costs. Only a few of these studies propose (optimal) control strategies for the mass flow rate in the installation, while ASHRAE (2007) reports that pump energy represent 4 to 21% of the total energy demand of GSHP systems. This section firstly summarizes the findings from the literature about optimal flow in (ground source) heat pump systems and secondly, it describes the current paper objective and structure.

To the best authors’ knowledge, Li and Lai (2013) were the first and only authors who proposed an analytical solution for optimal HCF flow rate and for optimal borehole length. Li and Lai applied an entropy minimization technique to a ground heat exchanger with single U-tube but without considering the heat pump. In their case, an optimal flow rate exists due to 1) a rising entropy generation from pressure drops when the flow rate increases and 2) a decreasing entropy generation due to smaller ground and HCF temperature differences when the flow rate increases. A major drawback of their method is that the analysis does not include the heat pump performance which depends on the HCF flow rate and temperature, while it plays a crucial role in the system performance. Furthermore, an entropy optimum does not necessarily coincide with an energy or economic optimum since entropy generated due to pressure drops has a different energetic and economic value than entropy generated due to heat transfer.

An energy optimization of the air flow rate in heat pump systems was proposed by Granryd (2010) for an air-to-air system. The author found an analytical solution that maximizes the COP$_2$ by using simple (empirical) correlations to express 1) the heat transfer and the pressure drop in the heat pump heat exchangers and, 2) the heat pump thermal power ($Q_{\text{cond}}$), as a function of the air velocity in the condenser and evaporator. The maximum COP$_2$ is then obtained by setting its derivative towards the air velocity to zero. A COP$_2$ optimum exists due to the increase of the heat pump COP$_1$ and the increase of pressure losses for increasing air

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1The coefficient of performance 2 (COP$_2$) is defined as the delivered useful energy (the condenser heat to the sink) divided by the electrical power use of the heat pump compressor and its fan or pump at both the source and sink sides.

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Figure 1: System description: (a) Schematic description of a ground-source heat pump system, (b) T-s diagram describing the working principle of the heat pump, (c) illustration of the dependency on the HCF flow rate for the heat pump COP and the pumping power $P_{pump}$.

flow rates in the evaporator. The author also showed that not only the COP$_1$ but also $Q_{cond}$ depends on the flow rate and that the flow rate that maximizes $Q_{cond}$ is not the same as the one maximizing COP$_2$. The simplified optimal solution shows good agreement with detailed simulations of the heat pump system.

The optimal HCF flow rate has also been investigated using simulation tools. Iolova and Bernier (2006) performed a simulation-based comparative study for a school in TRNSYS between a GSHP system using a variable speed drive (VSD) pump and one using constant flow rate. The system is composed of several heat pumps connected in parallel to a borefield. In case of the constant flow rate pump, the borefield pump is always on, regardless of whether the heat pump is on or off. In case of the VSD, each heat pump evaporator has a valve that blocks the flow when the heat pump is off. The VSD pump ensures a constant pressure drop over the system. They concluded that the variable flows systems saves up to 82% of the pumping energy use and 18.5% of the total GSHP system energy use. The fact that inefficient systems with constant flow rate still exist today stresses the need of simple expressions to calculate the optimal HCF flow rate in GSHP systems.

This paper proposes a simplified analytical solution for the optimal HCF flow rate of a GSHP system taking both the borefield and the heat pump into account. The optimal solution is a function of measurable variables and system parameters and data that can easily be obtained from manufacturer data sheets. Section 2 describes the steady state models used in Section 3 to derive the analytical solution and Section 4 validates the obtained expression with detailed simulation models.

MODEL DESCRIPTION

Fig. 1 (a) shows the considered system: a ground source heat pump extracts heat from a borefield with average ground temperature $T_b$. Thermal power $Q_{cond}$ is supplied to a building at a supply temperature $T_{cond,out}$. The mass flow rate of the heat carrier fluid (HCF) at the source side is $\dot{m}$, resulting in an inlet and outlet evaporator temperature $T_{eva,in}$ and $T_{eva,out}$. The considered electrical power uses are compressor power ($P_{comp}$) and source circulation pump power ($P_{pump}$). The mass flow rate at the sink side is assumed to be constant. Fig. 1 (c) shows that such a system has an optimal mass flow rate due to the increase of the heat pump COP (see Section 2.1) and the increase of $P_{pump}$ with $\dot{m}$.

The following sections describe the heat pump model (Section 2.1), the borefield model (Section 2.2) and the pump model (Section 2.3) that were used to derive an analytical solution for the optimal HCF mass flow rate $\dot{m}$.

HEAT PUMP MODEL

A heat pump is a device that converts heat from a low temperature source to heat at a higher temperature, by compressing a refrigerant using a compressor that is typically driven by an electric motor. The refrigerant evaporates in a first heat exchanger (evaporator), which requires heat at a low temperature. This refrigerant is compressed (state 1 in the T-s diagram Fig. 1, b) to a higher pressure and temperature (state 2). The refrigerant then condenses to a liquid state (state 3) while rejecting heat at a higher temperature in the second heat exchanger (condenser). The refrigerant then expands over an expansion valve and enters the evaporator (state 4).

The energy efficiency of the heat pump depends on the refrigerant pressure difference between the condenser and the evaporator. The pressure difference is determined by the required temperature difference which depends on both the source and sink temperature and on the mass flow rates. The temperature difference is controlled by the expansion valve. While the exact control method of the valve is typically a manufacturer secret, the valve needs to ensure a small amount of superheat in state 1 such that...
no liquid refrigerant enters the compressor. Furthermore, state 4 should be at a lower temperature than \( T_{\text{eva.out}} \) and state 2 should be hotter than \( T_{\text{cond.out}} \). These temperatures are further dependent on the HCF flow rate (see blue lines in Fig. 1(b). Therefore, the HCF mass flow rate \( \dot{m} \) in the evaporator indirectly influences the heat pump performance as it changes both \( T_{\text{eva.in}} \) and \( T_{\text{eva.out}} \).

The heat pump performance data provided by manufacturers are typically the COP and compressor electrical power \( P_{\text{comp}} \) as a function of the evaporator inlet temperature \( T_{\text{eva.in}} \), the condenser outlet temperature \( T_{\text{cond.out}} \) (or inlet \( T_{\text{cond.in}} \)), and (optionally) the evaporator mass flow rate \( \dot{m} \). Fig. 2 shows performance data for the Carrier water/air ground source heat pump type GZ048 (full load) for \( T_{\text{cond.in}} = 21.11^\circ \text{C} \) Carrier (2016). From Fig. 2 it is clear that the performance depends on both \( T_{\text{eva.in}} \) and \( \dot{m} \). Based on the discussions in previous paragraphs, a strong relation between \( T_{\text{m}} = T_{\text{eva.in}} + T_{\text{eva.out}} / 2 \) and the heat pump performance is expected. The heat pump performance is, however, not provided as a function of \( T_{\text{m}} \) but it can be computed from the other variables. Transforming Fig. 2 using \( T_{\text{m}} = T_{\text{eva.in}} + \frac{\dot{Q}_{\text{eva}}}{c_p \dot{m}} \) with \( c_p \) the HCF specific heat capacity, confirms this hypothesis since the curves for the different \( \dot{m} \) are now more or less coinciding (see Fig. 3). This relation has been verified for different \( T_{\text{cond.in}} \) for Carrier heat pump models G024 to G072 and for Daikin SmartSource 026 Daikin (2016).

The steady state behaviour of the heat pump can now be modelled using a linear fit of \( P_{\text{comp}} \) and \( \dot{Q}_{\text{eva}} \) (assuming full load and constant condenser inlet temperature):

\[
T_{\text{m}} = \frac{T_{\text{eva.in}} + T_{\text{eva.out}}}{2}
\]

\[
\dot{Q}_{\text{cond}} = \alpha + \beta T_{\text{m}}, \quad P_{\text{comp}} = \gamma + \psi T_{\text{m}} \Rightarrow \dot{Q}_{\text{eva}} = \dot{Q}_{\text{cond}} - P_{\text{comp}} = \alpha - \gamma + (\beta - \psi)T_{\text{m}} := \eta + \varepsilon T_{\text{m}}
\]
BOREFIELD MODEL

A borefield is a heat exchanger in the ground composed of one or multiple boreholes. Boreholes are drilled in the ground to a depth typically between 15 to 180 m ASHRAE (2007) with a diameter between 76 and 178 mm Chiasson (2007). A single U-shaped, double U-shaped or (less frequent) coaxial pipe is inserted in the borehole in order to circulate the heat carrier fluid (HCF). The pipe diameter varies between 20 and 40 mm ASHRAE (2007) and the mass flow rate of the HCF is usually chosen such that the flow is slightly turbulent, or a more conservative flow rate of 0.1 l/s per pipe may be used. The borehole is filled with grout, which is usually a mixture of bentonite and sand. The grout and the ground thermal conductivities and thermal capacity, as well as the borehole diameter, the pipe arrangement and material and the mass flow rate determine the so-called borehole resistance $R_b$. $R_b$ is defined by Hellström (1991) as the thermal resistance per length borehole between the average temperature of the heat carrier fluid $T_m$ and the borehole wall temperature $T_b$. According to ASHRAE (2007), $R_b$ values for single U-pipe range from 0.08 to 0.4 K.m/W but typical values rather range from 0.09 to 0.16 K.m/W for single U-pipe and from 0.05 to 0.08 K.m/W for double U-pipe boreholes Hellström and Sanner (2000).

If the average temperature of the ground at the borehole wall $\bar{T}_b$ is known, the most simple borehole model is obtained by disregarding the grout dynamics and by assuming a linear variation of the temperature along the pipe Lamarche et al. (2010). The ground is assumed to exchange heat $\dot{Q}_{eva}$ at $T_m$. The drawback of this simplification is that $T_{eva,in}$ (which equals the borefield supply temperature) can become higher than $\bar{T}_b$ for low flow rates, which is a violation of the second law of thermodynamics. We therefore assume that $\dot{Q}_{eva}$ is exchanged with $T_{eva,in}$ instead. $T_m$ can now be expressed as a function of $\dot{m}$ by using the energy balance equation in the borefield:

$$R^*_b := \frac{R_b}{L_{tot}} := \frac{R'_b + R_{conv}}{L_{tot}} \quad (3)$$

$$\dot{Q}_{eva} R^*_b = \bar{T}_b - T_{eva,in} \quad (4)$$

$$\dot{Q}_{eva} = \dot{m} c_p \left( T_{eva,in} - T_{eva,out} \right) \quad (5)$$

$$\Leftrightarrow T_m = \bar{T}_b - \left( R'_b + \frac{1}{2 \dot{m} c_p} \right) \dot{Q}_{eva} \quad (6)$$

with the HCF heat capacity $c_p$ and the total borehole(s) length $L_{tot}$, $R'_b$ is the borehole resistance between the pipe inner wall and borehole wall. The flow dependent convective resistance $R_{conv}$ is calculated separately. $R_b$ is usually obtained experimentally by means of a thermal response test. If the thermal properties of the grout and ground and the exact geometry of the borehole are known, $R'_b$ can be computed using the multipole method Hellström (1991). $R_{conv}$ in a circular pipe is computed from Equation (7) 2:

$$R_{conv} = \frac{1}{\pi \lambda_f / \text{Nu}} \quad \text{with} \quad \begin{cases} 
\text{Nu} = 0.023 R_e^{0.8} Pr^{0.35} & \text{if turbulent (Dittus-Boelter correlation)} \\
\text{Nu} = 3.66 + 4.36 \left( \frac{Pr}{Pr_{ref}} \right)^{1/2} & \text{if laminar (Thirumaleshwar (2009))}
\end{cases} \quad (7)$$

with the HCF thermal conductivity $\lambda_f$, the Reynolds number $Re = \frac{\dot{m} v f}{\rho f}$, the Prandtl number $Pr = \frac{\nu f}{\alpha}$, the HCF velocity $v_f$, the inner pipe diameter $d_{p,in}$, the HCF dynamic viscosity $\nu f$ and thermal diffusivity $\alpha$. Fig. 4 shows that $R_{conv}$ is only weakly dependent on $\dot{m}$ but that a transition from turbulent to laminar increases the resistivity from about 0.02 to 0.13 m.K/W.

2For the laminar case, the average value between the correlation for constant heat flux and correlation for constant wall temperature is taken.
PRESSURE LOSSES AND CIRCULATION PUMP

In this section, the circulation pump and typical pressure losses due to the circulation of the HCF in the heat pump and borefield are discussed.

The pressure drop over the heat exchangers depends on the heat pump type and size and can often be found in the manufacturer data sheets. Typically, small units have a pressure drop in the order of 4 kPa at minimal flow ($\theta_{eva,max} = T_{eva,in} - T_{eva,out} \approx 5K$). Larger units have a pressure drop in the order of 13 kPa at minimal flow ($\theta_{eva,max} = 4K$). The pressure losses associated to the borefield happen in the borehole, the horizontal connection pipes, the collector and the various bends, valves and connection elements. Typically, the total pressure drop is in the range of 0.5 to 1 bar, but it can widely vary and detailed pressure drop calculations should be carried out for more accurate results.

A circulation pump has an efficiency $\eta_{pump}$ that varies between 55 and 85% at nominal speed Bernier and Bourret (1999), depending on its size. Since the pump load has a quadratic pressure drop characteristic, similarity laws predict that the efficiency is not a function of the pump speed. In this paper we therefore assume $\eta_{pump}$ to be constant.

Assuming a constant pump efficiency $\eta_{pump}$ and a cubic relation between pump power and flow rates, and using a certain flow rate $\dot{V}_0$ and corresponding pressure drops, $P_{pump}$ can be expressed as

$$a_p = \frac{\Delta p_{borefield} + \Delta p_{heat pump}}{\eta_{pump} \dot{V}_0^2}$$

$$P_{pump} = a_p \left( \frac{\dot{m}}{\rho} \right)^3$$

with the borefield and the heat pump pressure drops $\Delta p_{borefield}$ and $\Delta p_{heat pump}$.

OPTIMAL SOLUTION

In this section, an analytical solution for the optimal HCF mass flow rate is derived by maximizing the system coefficient of performance ($COP_2$) (Eq. 10). Constant condenser inlet temperature and full load condition are assumed for the heat pump.

$$\frac{1}{COP_2} = \frac{P_{pump} + P_{comp}}{Q_{cond}}$$

The optimization problem can be re-written by substituting the model equations in Eq. 10. First $T_m$ is obtained as a function of $\dot{m}$ and some parameters using Eq. 2 & 6:

$$T_m = \tilde{T}_b - \left( R_b^* + \frac{1}{\dot{m} \rho_{cp}} \right) (\eta + \varepsilon T_m)$$

$$\Leftrightarrow T_m = \frac{\dot{m} (\tilde{T}_b - R_b^* \eta) - \frac{\rho_{cp}}{\dot{m}} \xi}{\dot{m} (R_b^* \varepsilon + 1) + \frac{\rho_{cp}}{\dot{m}} \nu}$$

with $\xi := R_b^* \varepsilon + 1$, $\nu := \frac{\rho_{cp}}{\dot{m} \rho_{cp}}$, $\xi := \tilde{T}_b - R_b^* \eta$, $\lambda := \frac{\rho_{cp}}{\dot{m}}$.

By inserting Eq. 2 & 9 in Eq. 10 and developing it with Eq 11, following optimization problem is obtained:

$$\frac{1}{COP_2} = \frac{a_1 \dot{m}^4 + a_2 \dot{m}^3 + a_3 \dot{m}^2 + a_4 \dot{m} + a_5}{\alpha + \beta T_m}$$

$$= \frac{a_1 \dot{m}^4 + a_2 \dot{m}^3 + a_3 \dot{m}^2 + a_4 \dot{m} + a_5}{a_5 + a_6}$$

with $a_1 = \frac{\rho_p \dot{m}^4 + \gamma + \psi T_m}{\alpha + \beta T_m}$, $a_2 = \frac{\rho_p \dot{m}^3}{\alpha + \beta T_m}$, $a_3 = \frac{\rho_p \dot{m}^2}{\alpha + \beta T_m}$, $a_4 = \frac{\rho_p \dot{m}}{\alpha + \beta T_m}$, $a_5 = \frac{\rho_p}{\alpha + \beta T_m}$.

The optimal solution is computed from the roots of the derivative of this function (the denominator is removed from the
The optimal analytical solution is validated using a simulation model based on an existing GSHP system. The system consists of a borefield with 99 boreholes of 30 m deep (double U-type), a Wilo Cronoline IL 80/220 4.4 circulation pump, and heat pumps. The total GSHP system pressure drop at nominal flow rate (14.85 kg/s) is 170 kPa. It is connected to 14 Carrier GZ048 heat pumps that are operated at nominal condenser flow rate with an inlet temperature of 21.1 °C. All parameter values used for the simulation and for the analytical solution are summarized in Table 1. The system is modelled in Modelica using the borefield model from Picard and Helsen (2014) and the pump model from Wetter et al. (2015). The heat pump model uses a 3-dimensional linear table interpolation of the manufacturer performance data Carrier (2016), Fig. 3. The Modelica model is simulated using Dymola 2017.

For the study, the optimal solution can be obtained by plotting Eq. 13 or by using a line search method to find the roots of Eq. 17.

### RESULTS, VALIDATION AND DISCUSSION

The optimal analytical solution is validated using a simulation model based on an existing GSHP system. The system consists of a borefield with 99 boreholes of 30 m deep (double U-type), a Wilo Cronoline IL 80/220 4.4 circulation pump, and heat pumps. The total GSHP system pressure drop at nominal flow rate (14.85 kg/s) is 170 kPa. It is connected to 14 Carrier GZ048 heat pumps that are operated at nominal condenser flow rate with an inlet temperature of 21.1 °C. All parameter values used for the simulation and for the analytical solution are summarized in Table 1. The system is modelled in Modelica using the borefield model from Picard and Helsen (2014) and the pump model from Wetter et al. (2015). The heat pump model uses a 3-dimensional linear table interpolation of the manufacturer performance data Carrier (2016), Fig. 3. The Modelica model is simulated using Dymola 2017.

The parameters used for the analytical solution are summarized in Table 1. The goodness of the linear fits for the considered heat pumps are shown in Fig. 3 (black dashed line). Q_{\text{cond, lin}} and COP_{\text{lin}} are computed from Q_{\text{cond, lin}} and P_{\text{comp, lin}}. Fig. 3 shows that Q_{\text{cond, lin}} is a good approximation of Q_{\text{cond}} but P_{\text{comp, lin}} shows a less linear behaviour. This results in a slight underestimation of COP_{\text{lin}} but a good estimation of the COP.

The following experiment is carried out: the borefield is initialised by assuming a uniform ground temperature of 15 °C and then operating the pump and the heat pump at nominal flow rate for 17.4 days. The borefield cools down to around 8.7 °C. The mass flow rate is then changed to a different fixed value (see Fig. 5). Once the heat pump has generated 200 kWh of thermal energy, the COP's are reported. The reason for this approach is to allow objective comparison of a transient system. The analytical solution is obtained by using the same parameter values and a borehole wall temperature of 8.7 °C.

Fig. 5 compares the results from the simulation model (blue dashed line) and the analytical model (red dotted line) for different mass flow rates. Fig. 5 (a) and (b) are expressed as a function of the system mass flow rate \( \dot{m}_{bf} \) while Fig. 5 (c) and (d) correspond to a single heat pump. Fig. 5.a shows that the analytical model underestimates the HCF temperatures. This is caused by the assumption that the HCF exchanges heat with the borefield at \( T_{\text{eva, in}} \) instead of using an exponential HCF temperature variation Lamarche et al. (2010). The error gets amplified by the decrease of \( Q_{\text{cond}} \), \( Q_{\text{eva}} \) and \( P_{\text{comp}} \) for smaller \( T_m \) resulting in an underestimation of COP_{\text{lin}}.
Table 1: Parameter values used for the validation.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
<th>Unit</th>
</tr>
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<tr>
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</tr>
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</tr>
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<td>$R_{conv}$</td>
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<td>$L_{tot}$</td>
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<tr>
<td>$c_p$</td>
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<td>[J/(kg$^\circ$C)]</td>
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<tr>
<td>$\nu$</td>
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<tr>
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<tr>
<td>$\varepsilon$</td>
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<td>[W/K]</td>
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</table>

HCF: 20% glycol Heat pump

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
<th>Unit</th>
</tr>
</thead>
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<tr>
<td>$\rho$</td>
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<td>[m$^2$/s]</td>
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<tr>
<td>$\lambda$</td>
<td>0.505</td>
<td>[W/K]</td>
</tr>
<tr>
<td>$\rho_0$</td>
<td>1033</td>
<td>[kg/m$^3$]</td>
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<td>$c_{p0}$</td>
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<td>[J/(kg$^\circ$C)]</td>
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</tr>
<tr>
<td>$\lambda_0$</td>
<td>0.505</td>
<td>[W/K]</td>
</tr>
</tbody>
</table>

Figure 5: Results comparison between analytical model (dotted red) and simulation (dashed blue) model: (a) in and outlet evaporator HCF temperatures; (b) (left axis) heat pump COP$_1$ and system COP$_2$ (right axis) head losses; (c) condenser thermal power; (d) compressor electrical power.

It should be noted that optimal COP$_2$ is generally found at lower flow rate as illustrated by Southard et al. (2014) where the GSHP COP$_2$ was increased by 18% when the differential pressure set point on the ground loop was reduced from 1.4 to 0.6 bar. The fact that the optimal flow rate for the validation exercise is rather high is explained by its assumptions: i) the heat pumps operate at full load, ii) a relatively small head loss of the ground heat exchanger was used. Running the heat pumps at part load would lead to a lower optimal flow rate as the relative influence of the pumping energy on the COP$_2$ increases. Other assumptions for the validation are i) constant $T_{cond,in} = 21.11$ $^\circ$C, and ii) a cubic relation between pumping power and mass flow rate is assumed.

Further work should experimentally validate the proposed analytical solution for different heat pump load ratio's and different GSHP systems and it should confirm that the optimal flow rate of a GSHP system depends on the heat pump load, the ground temperature, and borefield and heat pump characteristics. Notice that Eq. 2 needs to be re-computed for each heat pump part load ratio as the heat pump characteristics change accordingly. The optimal mass flows can then be derived for each part load ratio.

CONCLUSION

This paper shows that there exists an optimal heat carrier fluid flow rate for ground source heat pump (GSHP) systems that minimizes its energy use. The paper proposes an analytical solution for computing the optimal flow rate as a function of measurable variables and system parameters and data that can be obtained from manufacturer data sheets. The optimal solution is based on a steady-state
approximation of the borehole and a linear approximation of the thermal and electrical power of the heat pump as a function of the average evaporator temperature. It was found that expressing the heat pump powers and COP as a function of the average evaporator temperature instead of its inlet temperature reduces their mass flow rate dependencies. The obtained analytical solution shows good agreement with the optimal solution obtained by a detailed simulation model representing an existing GSHP system of 99x30m boreholes.

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REFERENCES

First Measurements of a Monitoring project on a BTES system

Patricia Monzó        Alberto Lazzarotto        José Acuña

ABSTRACT
Performance of Borehole Thermal Energy Storage (BTES) systems depends on the temperature of the secondary fluid, circulating through the ground-loop heat exchangers. Borehole systems are therefore designed in order to ensure that inlet and outlet temperatures of the secondary fluid are within given operational limits during the whole lifetime of the system. Monitoring the operation of the bore fields is crucial for the validation of existing models utilized for their design. Measured data provides valuable information for researchers and practitioners working in the field. A first data-set from an ongoing monitoring project is presented in this article. The monitoring system comprises temperature sensors and power meters placed at strategic locations within the bore field. A distributed temperature sensing rig that employs fiber optic cables as linear sensors is utilized to measure temperature every meter along the depth of nine monitored boreholes, yielding data regarding both temporal and spatial variation of the temperature in the ground. The heat exchanged with the ground is also measured via power meters in all nine monitored boreholes as well as at the manifold level. The BTES system is located at the Stockholm University Campus, Sweden, and consists of 130 boreholes, 230 meters deep. After more than a year of planning and installation work, some selected measurements recorded in the BTES during the first months of operation are reported in this article.

INTRODUCTION

BTES systems actively and intentionally create an underground heat (and/or cold) store by means of a number of closed-loop borehole heat exchangers (BHEs) that use the surrounding rock or soil as a storage medium. As the heat carrier fluid circulates through the BHEs, heat is transferred, mainly by conduction, from the fluid to the surrounding ground and vice versa. The annual mean temperature of the ground storage volume varies according to the net heat that is exchanged over a year.

Performance of BTES systems depends on temperatures of the secondary fluid circulating in BHEs, which should be kept within a certain temperature range during the life of the system. Because the rate of year-to-year temperature increase or decrease levels out during the first years of operation, a life of 20-25 years is often assumed for the design calculation, even though the ground heat exchangers are expected to last much longer. To ensure the secondary fluid temperature to be within the operational temperature limits, BTES systems require accurate design tools to predict the long-term thermal response of the bore field over its life time.

Measured data of large bore fields is lacking in the academy. Mostly the response of small scale experimental setups (Acuña, 2013, Luo et al., 2013, Cimmino and Bernier, 2015) have been published. Data from in-situ monitoring of full scale installations (Hern, 2004, Montagud et al., 2011, Naiker and Rees, 2011, Sanner et al., 2016) are of a great value for validation of the methods implemented in bore field design tools. During the last years, the interest from industrial actors has resulted in a better understanding of the thermal response of bore fields. Examples where monitoring of bore fields is discussed are presented in (Mikhaylova et al., 2015, Michalski et al., 2016). Examples of validation of methods and design tools against measured data are illustrated in (Cullin et al., 2015, Sanner et al., 2016). Monitored data is also

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used to evaluate system performance and to investigate optimization schemes. Further developments of accurate bore field design tools, where aspects such as thermal interaction between neighboring boreholes, neighboring bore fields, influence of groundwater flow, and influence of other surrounded heat sources, may be strongly supported by accurate monitoring data of bore fields for long-term periods.

In this paper, a few data sets from an ongoing monitoring project on a new BTES installation are reported. The installation started operating on April, 15 2016. The measurements are presented for a two weeks period from September 2016. The monitoring system used consists of temperature sensors and power meters located at strategic positions within a bore field, as described in (Monzó et al., 2016). The work reported in this paper is one of the initial steps of a project that intends to evaluate and optimize the actual performance of the BTES system and use the measured data for validation of some bore field modelling approaches.

**DESCRIPTION OF THE BTES**

The BTES installation is designed to provide annually about 4 GWh of cooling and about 3 GWh of heating for one newly constructed building and for a group of existing buildings at Stockholm University Campus, Sweden. As described in (Monzó et al., 2016), the BTES is connected to a cluster of heating and cooling systems. In the summer, the heat rejected by two large chillers is stored in the ground. In the winter, space heating is provided partly by the heat rejected from the chillers and is partly supplied from the ground using a heat pump. During mild seasons, cooling is achieved from the BTES, and represents a minor portion of the total cooling demand. The BTES is sized to keep the temperature of the secondary fluid entering the heat pumps between 2.5°C and 31°C. The secondary fluid is an aqueous solution of bioethanol (20% by weight). The BTES system operational power limits are 1400 kW for heat extraction and 2500 kW heat injection. Supplementary heating from a district–heating network is used when hourly peaks exceed the design limits. Dry cooler units are used when hourly peaks exceed design limits for cooling loads. The annual net design heat exchanged in the underground storage volume is around 1400 MWh of cooling.

![Plan view of the BTES - Bore field layout.](image-url)
The BTES system comprises 130 vertical and inclined groundwater filled BHEs, of 230 meter length, arranged in an uneven pattern that covers a total storage volume of around 5,850,000 m³. The ground thermal conductivity reported during the design phase is around 3.9 W/(m K) with a volumetric heat capacity of the soil of around 2.2 MJ/(m³ K). A storage ground capacity of around 3,570 MWh/K is obtained for the above mentioned storage volume. Thus, the expected temperature change in the storage volume due to the net annual heat exchanged (1400 MWh) over the first year is around 0.4 K. Figure 1 illustrates the layout of the bore field and the arrangement of the boreholes within their corresponding manifolds. Manifolds illustrated with solid green lines represent measurement manifolds containing monitored boreholes. The position of monitored boreholes within the field is depicted by red circles in Figure 1. The projection of the inclined boreholes on the ground surface is also illustrated in Figure 1 using solid lines.

MONITORING SYSTEM

The monitoring system consists of power and temperature measurements. Power meters measure the volume flow (l/s) and fluid temperatures (°C) at the supply and the return pipes, which corresponds to one set of measurement, in both measurement boreholes and their corresponding manifold. For further information about their and location within the bore field, readers are referred to (Monzó et al., 2016). The power meters are of type Dynaflox DEM-series meters, with a pressure loss less than 0.02 MPa at a permanent flow. The working temperature range is between –5°C and +95°C. Two PT1000 resistance thermometers with precision of ±0.1K (according to manufacturer) are used to measure temperatures in the supply and return pipes. Power is measured every minute.

A distributed temperature sensing (DTS) instrument (type ORYX DTS) measure the temperature along optical fiber cables placed in the groundwater within the boreholes. The physical principle behind this measurement technique is based on the fact that glass fibers are sensitive to temperature changes. In DTS instrument a laser beam is sent through a fiber cable and the backscattered signal is collected and translated to temperature. The DTS instrument measures averages of temperature in both time and space. In our installation temporal sampling is set at 10 minutes interval and spatial sampling is set at 1 meter. The longest fiber cable loop installed at this site is about 3.5 km long. With this setting, a temperature resolution less than 0.1°C can be achieved according to the instrument manufacturer. The instrument has a single laser and four channels through which it is connected to the fiber cables. The laser is only active on one channel at a time, therefore an average temperature value for every meter of fiber cable is recorded every 40 minutes in each channel. For practical reasons, fiber cables are installed separately in each borehole (upward and downward). Since the DTS instrument requires a continuous fiber cable, fiber cables are spliced (welded) to achieve the arrangement in each DTS channel as shown in Table 1.

<table>
<thead>
<tr>
<th>Table 1. Arrangement of measurement boreholes in each DTS channel</th>
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<tbody>
<tr>
<td><strong>Channel</strong></td>
</tr>
<tr>
<td>---</td>
</tr>
<tr>
<td>1</td>
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<tr>
<td>2</td>
</tr>
<tr>
<td>3</td>
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<tr>
<td>4</td>
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</table>

Figure 2 shows, as an example, the splice-fiber scheme and the temperature measurement recorded on 2016-07-01 17:45:14 along the length of the fiber cable connected to channel 3. As presented in Table 1, two measurement boreholes in two manifolds are measured in channel 3. The fiber optic cable is installed so that the laser travels upwards and downwards in borehole 6 and 12 (outside the pipes). Each cable has four fibers, but only two of them are used in the set-up described in this paper, fiber 1 and 2 illustrated in Figure 2(a). The fibers are spliced so that the cables relative to the measurement boreholes are connected in series. This takes place at junction boxes 2 to 5 in the example shown in Figure 2, while fibers 1 and 2 connect to each other in junction box 1 (end-loop splice). This fiber set-up is also known as duplex arrangement (Hausner et al., 2011). Temperatures are measured twice along the cable.
A post-processing step is necessary to identify the DTS measurements relative to the length of the fiber cables corresponding to each measurement borehole in each channel. The intervals of interest are identified by knowing the fiber cable path of the channel and by using some sections of the cable for reference. These sections are some extra meters of fiber cable reserved before and after the junction boxes, which serve as references when placed in thermal baths with a known temperature. In Figure 2(b), the length of the fiber cable corresponding to borehole 6 and 12 are highlighted along with the sections used as reference.

In December 2015 the fiber cables were connected to the DTS equipment. Subsequently, the DTS equipment was connected to a local data acquisition system and data has been recorded since then. During the connection of the fiber cables to the DTS equipment, we realized that the fiber cable in one of the selected boreholes, borehole 24 in manifold 4, was not transmitting. The fiber cable probably broke during the construction work at the site. Currently, the temperature along the depth of 9 of the 10 instrumented boreholes is monitored. The connection of the power meters to a local data acquisition system was accomplished in June 2016 and since then power data is being recorded. Because of flooding of one measurement well where power meters are placed, the meters relative to borehole 6 and manifold 1 were damaged and they do not transmit accurate values at the moment. Unexpected low measurements of volume flow rates were observed in borehole 38, which are not reported in this paper. The power meters use an M-bus connection to transfer the data while the DTS unit communicates through a serial connection.

Once the data acquisition system was set up, a careful mapping of the instruments was carried out. The raw power data received through the local data acquisition system (one set of measurements for each power meter) were not initially tagged and manual identification was required to map the measurement to the relative meter. For this purpose, advantage was taken of the fact that boreholes and manifolds are equipped with flow control valves and are labelled at the site according to the numbering presented in (Monzó et al., 2016). The mapping process consisted of closing the flow control valve of each of the measurement boreholes and manifolds, one at a time. For each of these time intervals one set of measurement recorded by our data acquisition system returned a zero value for the volume flow, indicating the correspondence between data and the relative meter. This procedure allowed the identification and tagging of each set of power measurements. Mapping of the DTS measurements required both the identification of the specific
boreholes connected in each channel and the section of the fiber cable relative to each borehole. The monitored boreholes were closed, one by one, for periods of about 10 hours, so that no heat was injected in the closed boreholes. During these time spans, the measured temperature profiles along the borehole length showed that one of the temperature profile had a distinct behavior compared to the others enabling the identification of the boreholes in each channel. The mapping of the sections of interest in the fiber cables was done as previously exemplified for channel 3.

FIRST MEASUREMENTS

After performing some operational tests during February and March 2016, the BTES system started its operation on April 15 2016 with a total heat injection of around 3400 MWh since then. Since the monitoring set-up was fully operational in September 2016, the authors have chosen a two weeks period from September 2016 to be reported in this paper.

Power Measurements

Measurements of volume flows $V$ (l/s) and temperatures ($^\circ$C) at the supply, $T_{in}$, and return pipes, $T_{out}$, are obtained from the power meters installed in measurement manifolds and boreholes. The power $q$ (W) exchanged in the pipe loop results from $q = V \cdot \rho \cdot C_p \cdot (T_{in} - T_{out})$. The density, $\rho$ (kg/m$^3$), and the specific heat capacity, $C_p$ (J/(kg K)) of the secondary fluid at the working temperature is calculated continuously using analytical correlations from experimental measurements presented in (Ignatowicz et al., 2016).

Figure 3 shows examples of the volume flow, inlet and outlet temperatures and power measured at the monitored manifolds from Sept 8 to Sept 21. Figure 4 presents measurements at the borehole level. The period shown in Figure 3 and 4 corresponds to a heat injection period. Two operation strategies related to the outdoor temperatures can be distinguished in Figure 3 and Figure 4. “Strategy 1” from Sept 8 to Sept 15 characterized by large heat injection when the average daily temperature is around 17.5$^\circ$C, and “Strategy 2” from Sept 16 to Sept 21 when the average outdoor daily temperature was lower (13.3$^\circ$C) and less heat was injected into the ground. On a daily basis, two operation schemes can also be observed: a “day time scheme” and a “night time scheme”. The former occurs approximately from 8 am to 9 pm when buildings are occupied and outdoor temperatures are higher. The latter occurs approximately from 9 pm to 8 am, when there is no building occupancy and outdoor temperatures are lower. Slightly less heat is injected into the BTES during weekends (especially at nights), since the decreased occupancy of the buildings reduces the demand of cooling loads.

During the “day time scheme” between Sept 8 to Sept 15, a fluid temperature difference of 3 to 4K is maintained, and the capacity of the system is adjusted by changing the volume flow. In the “night time scheme” the temperature difference decreases to values around 1.5 to 2K and the volume flow is kept at its minimum rate. In Strategy 2 (Sept 16 to Sept 21), during “day time scheme”, the fluid temperature difference is around 3K and the volume flow is varied to satisfy the heat injection. While for the “night time scheme” the temperature difference is reduced to around 1K and the volume flow is kept to its minimum rate. On Sept 19, a peak injection occurred during early hours of “day time scheme”. The heat injection in the second period (Sept 16 to Sept 21) is reduced by around 30% in comparison to the first period (Sept 8 and Sept 15). On a daily basis, the heat injected (in both observed periods) varies around 50% between day time and night time schemes. It can also be observed that the heat injection into the BTES is softened from the intermittent on/off operation of chillers by means of a buffer tank that decouples the BTES from the remaining systems in the cluster, as shown in (Monzó et al., 2016).

Figure 3 also shows that the mass flow distribution among the manifolds has been balanced in order to have higher flow rates in manifolds with a larger number of boreholes. Similarly, the more boreholes per manifold, the larger amount of heat injected by manifold. When comparing inlet temperatures in manifolds at a given time, a maximum difference of around 1.5K is observed at this reported period. Heat losses in piping between central pipe and measurement wells may partly explain the differences in inlet temperatures. The comparison of the outlet temperatures
in the manifolds during this period also shows that the largest deviation is about 1.5K. The discrepancies in inlet and outlet temperatures are consistent. However, because of the relatively high inlet temperatures differences, further investigation will be carried out at site to pinpoint possible biases in the measurements.

Figure 3 Volume flow, inlet and outlet temperatures and power at the measurement manifolds during Sept 8 to 21.

Measurements from power meters in monitored boreholes are shown in Figure 4, with the exception of borehole 6 and 38. Variation around ±5% in volume flows is observed in the investigated period, which illustrates that the mass flow are relatively balanced among the boreholes. Due to the operational strategies applied, the volume flow per borehole varies approximately between 0.2 to 0.75 l/s. For a secondary fluid temperature of 16°C, a fluid to pipe thermal
resistance of 0.02 to 0.0051 (m K)/W results for this range of observed flow rates. Temperature differences between the fluid and the borehole wall change by about 0.15 degrees for the heat injection (20 to 50 W/m) occurring during the reported period. This change in the volume flow influences the thermal and the hydraulic performance of the borehole. Further investigation to assess the benefit of alternatives strategies will be performed in future work.

At any instant during this period, a maximum difference of 1.4K is observed between inlet temperatures in measurement boreholes. A maximum difference between manifolds and boreholes of approximately 0.6K is observed. Borehole inlet temperatures are expected to have a value similar to the inlet to their relative manifolds. Further examination at site will be devoted to amend these issues. The power injection per unit length (W/m) illustrates that the heat injection loads are balanced among the boreholes. The power is about the same in all monitored boreholes with exception of slightly lower values at borehole 106. The power injection per unit length (W/m) during the first period (Sept 8 and Sept 15) reaches values of around 40 to 50 W/m for “day time scheme”, while it decreases to 20 W/m at “night time scheme”. During the second period (Sept 16 and Sept 21), the power injection per unit length (W/m) is around 20 W/m at “day time scheme” and around 5 W/m at “night time scheme”.

Temperature Measurements

Work on calibration of the DTS measurements is ongoing. Figure 5 shows raw data of the groundwater temperature measured along the length of the nine monitored boreholes at two specific time steps. The first time chosen occurs before the operation of the BTES was started, on February 1, 2016, while fluid circulation was taking place in all boreholes. The measurements show that the temperature of the groundwater outside the pipes tended to be evened out along the depth of all borehole. The second time is chosen on Sept 10, 2016, a moment when the groundwater temperature along the boreholes increases in comparison with the profiles observed in the first period. The latter reflects the effect of the injection of heat into BTES during the first cooling season, since April 15, 2016. It can be also observed that the increase is non uniform along the depth. The new profiles present a temperature difference of around 2K between the top and the bottom of the boreholes. The upper part of the boreholes might be exchanging a larger share of heat than the bottom part. Temperatures discontinuities along the borehole depth are observed at a few sections, for instance at about 75 m in BH106. These local features will be analyzed in the near future.

![Figure 5 Temperature measurements along the borehole length at two particular times.](image)

CONCLUSION

A monitoring set up and a few data sets from a BTES system, consisting of 130 groundwater filled BHEs, are presented in this paper. The BTES system started operating on April 15, 2016. Relevant activities related to the final implementation of the BTES monitoring set-up are described. The set-up comprises groundwater temperature
measurements along the depth of nine selected monitored boreholes and power measurements at all monitored boreholes and at their relative manifolds.

The power measurements show large variation of volume flows in the ground loop, which might play a major role on regulation of the heat load to the BTES. Raw data regarding groundwater temperature along the depth of all the monitored boreholes is shown while circulating the secondary fluid before the operation of the BTES as well as during the end of the cooling season while injecting heat into the ground. At all boreholes, the groundwater temperature is evened out along the depth when only circulation occurs and increases during heat injection. A temperature difference of around 2K from the top to the bottom of the borehole is observed. These first data sets show the potential of the described monitoring system and provides an insight into detailed information that will successively be published regarding how this BTES system is operated and how the temperatures within the BTES are affected in accordance with operational strategies. The work reported in this paper is one of the initial steps of a project that intends to evaluate and optimize the actual performance of the system and use the measured data for validation of some bore field modelling approaches.

ACKNOWLEDGMENTS

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REFERENCES


Energy Performance Evaluation of a Recycled Water Heat Pump System in Cool and Dry Climate Zone

Piljae Im, PhD  Xiaobing Liu, PhD

ABSTRACT
This paper presents performance evaluation results for a recycled water heat pump (RWHP) system, which uses the recycled water from a municipal water system as a heat sink and heat source for heat pumps. The temperature of the recycled water, system heat flow, and efficiency were analyzed based on measured data from December 2014 through August 2015. The annual energy consumption of the RWHP system was compared with that of a baseline system—a conventional variable-air-volume system using a water-cooled chiller and a natural gas–fired boiler, both of which meet the minimum energy efficiencies allowed by ASHRAE 90.1-2013. The analysis results indicate that, on an annual basis, the RWHP system has avoided 50% of source energy consumption, reduced CO₂ emissions by 41%, and saved 34% in energy costs compared with the baseline system.

INTRODUCTION
In 2009, 26 projects were competitively selected and funded with American Recovery and Reinvestment Act (ARRA) grants to demonstrate innovative ground source heat pump (GSHP) technologies. Denver Museum of Nature and Science (DMNS) in Denver, Colorado, was one of the 26 demonstration project sites. The new facility is a five-story, 140,000-gross-square-foot addition to the museum. The innovation of the demonstrated GSHP technology at this site is that this system uses water from the city’s underground municipal recycled (non-potable) water system as the heat sink and heat source for the heat pump in lieu of the borehole field used in conventional GSHP systems. This project is believed to be the first of its kind in the United States. The demonstrated technology, called recycled water heat pump (RWHP), has potential to be applied in other urban areas, given that there are existing RW distribution systems in many cities. For example, the existing RW system in Denver is over 70 miles long and is still expanding, and currently 171 water districts in 11 states in the United States have existing RW systems. Currently, the RW is mainly used for landscape irrigation and pond water-level management.

This case study was conducted based on the available measured performance data from December 2014 through August 2015, utility bills for the building in 2014 and 2015, construction drawings, maintenance records, personal communications, and construction costs. The annual energy consumption of the RWHP system was compared with that of a baseline scenario—a conventional variable-air-volume (VAV) system using a water-cooled chiller and a natural gas–fired boiler, both of which meet the minimum energy efficiencies allowed by ASHRAE 90.1-2013 (ASHRAE 2013). The comparison was made to determine energy savings, operating cost savings, and CO₂ emission reductions achieved by the RWHP system. A cost analysis was also performed to evaluate the simple payback of the RWHP system. More detailed information for this case study is given by Im and Liu (2015).
BUILDING AND SYSTEM DESCRIPTION

The host building is a new addition, the Education and Collection Facility building, to the existing DMNS building. This 140,000 ft² building has five stories (including two stories underground). The RWHP system uses seven 30-ton modular water-to-water heat pumps (WWHPs). Each WWHP uses R-410A refrigerant, has two compressors, and can independently provide either hot water (HW) or chilled water (CHW) to the building. A master controller modulates the operation mode of each WWHP to satisfy the varying heating and cooling demands of the building. The RWHP system can provide HW and CHW simultaneously. As shown in Fig. 1, the RWHP system has five water loops: the RW loop, source water loop, CHW loop, HW loop, and precooling loop. Each loop has its own circulation pump and associated controls. Since RW may not always be available because of routine maintenance or other reasons, the RWHP system also has two steam heat exchangers (HXs) to provide supplemental heating, and a cooling tower that serves as a backup heat sink when RW is not available or not sufficient to keep the source water temperature within a desired range. The precooling loop provides cold water to the five air-handling units (AHUs) in the building (one for each floor) and serves as the heat source for a separate WWHP unit dedicated to domestic hot water (DHW) heating. Water in this loop is cooled by the source water loop and the WWHP for DHW. The supply cold water temperature is designed to be between 45 and 75°F, with a maximum 10°F temperature rise in the return water. The heat rejected from the precooling loop goes to the source water loop and becomes a heat source for the modular WWHPs (for producing CHW and HW). Table 1 is a list of data points related to the RWHP system, which are indicated in Fig. 1.

Fig. 1. A schematic of the recycled water heat pump system with monitored data points shown.

A new pipeline for the RW was constructed between the existing city RW pipeline and the target building. Two 8-inch PVC pipes (one for supply and the other for return) were installed side by side in a 48-inch-wide trench. The supply line was insulated with 2 inches of foam insulation to reduce heat transfer between the supply and return lines. Trench depth varied from 6 feet to 15 feet through the 3,300-foot-long pipeline (in each direction). Remote water sampling stations were required on the lines between the conduit and building entry points. Isolation valves and a meter were installed at the conduit connection. Other isolation valves were installed at points before the lines enter
the building. RW is pumped through a plate frame HX (referred as “RW HX” hereinafter; shown in Fig. 1) by two 15-hp redundant variable-speed pumps (referred as the “RW pump” hereinafter) to exchange heat with the source water.

Table 1. RWHP System Monitoring Data Points

<table>
<thead>
<tr>
<th>Label</th>
<th>Description</th>
<th>Label</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>TRWS</td>
<td>Recycled Water Supply Temperature</td>
<td>THL3b</td>
<td>HW Loop Temp after Steam HX 309</td>
</tr>
<tr>
<td>TRWR</td>
<td>Recycled Water Return Temperature</td>
<td>FHL</td>
<td>HW Loop Flow</td>
</tr>
<tr>
<td>FRW</td>
<td>Recycled Water Flow</td>
<td>WHLP</td>
<td>HW Loop Pump Power</td>
</tr>
<tr>
<td>WRWP</td>
<td>Recycle Water Pump Power</td>
<td>TCL1</td>
<td>CHW Loop Temperature to WWHPs</td>
</tr>
<tr>
<td>TSL1</td>
<td>Source Loop after RW HX Temperature</td>
<td>TSL2</td>
<td>CHW Loop Temperature from WWHPs</td>
</tr>
<tr>
<td>TSL2</td>
<td>Source Loop after Cooling Tower Temperature</td>
<td>TCL2</td>
<td>CHW Loop Temperature from WWHPs</td>
</tr>
<tr>
<td>TSL3</td>
<td>Source Loop after Pre-cool Loop Temperature</td>
<td>WCLP</td>
<td>CHW Loop Pump Power</td>
</tr>
<tr>
<td>TSL4</td>
<td>Source Loop after Steam HX Temperature</td>
<td>WPCLP</td>
<td>Pre-Cooling Loop Pump Power</td>
</tr>
<tr>
<td>TSL5</td>
<td>Source Loop after WWHP Temperature</td>
<td>TPCL1</td>
<td>Pre-Cooling Loop Temperature prior to DHW WWHP</td>
</tr>
<tr>
<td>FSL</td>
<td>Source Loop Flow</td>
<td>TPCL2</td>
<td>Pre-Cooling Loop Temperature after DHW WWHP</td>
</tr>
<tr>
<td>WSLP</td>
<td>Source Loop Pump Power</td>
<td>FPCL</td>
<td>Flow pre-Cooling Loop</td>
</tr>
<tr>
<td>WHP1-7</td>
<td>Heat Pump Power (WWHPs 1 through 7)</td>
<td>WCLP</td>
<td>Pre-Cooling Loop Pump Power</td>
</tr>
<tr>
<td>THL1</td>
<td>HW Loop Temperature to WWHPs</td>
<td>WCT</td>
<td>Cooling Tower Power</td>
</tr>
<tr>
<td>THL2a</td>
<td>HW Loop Temp before Steam HX 308</td>
<td>SDHWHP</td>
<td>DHW Heat Pump Status</td>
</tr>
<tr>
<td>THL2b</td>
<td>HW Loop Temp before Steam HX 309</td>
<td>TDHWHP</td>
<td>DHW Heat Pump Supply Temp</td>
</tr>
<tr>
<td>THL3a</td>
<td>HW Loop Temp after Steam HX 308</td>
<td>TAO</td>
<td>Ambient Temperature</td>
</tr>
</tbody>
</table>

The control sequence for the RW pump is different in heating and cooling seasons. During cooling season, the RW pump is turned on when the leaving water temperature from the RW HX (TSL1 in Fig. 1) is above 55°F and the leaving water temperature from the modular WWHP (TSL5 in Fig. 1) is at least 2°F higher than the RW supply temperature (TRWS in Fig. 1). During heating season, the RW pump is turned on when TSL1 is below 55°F and the TSL5 is at least 2°F lower than TRWS. When it is turned on, the speed of the RW pump is modulated to maintain the temperature differential of the RW at 10°F across the RW HX.

RESULTS

Recycled Water Temperature

Hourly RW temperatures during the period from January through August 2015 are plotted in Fig. 2 along with the hourly outdoor air (OA) temperatures (OATs). As shown in this figure, the RW temperature was relatively stable throughout the monitored period, whereas the OAT fluctuated to a much larger degree during the same period.

Although the monthly average RW and OA dry bulb temperatures during the cooling season (June through August) were close to each other, the OAT fluctuated in a much larger range during each month. The maximum OA dry bulb temperature (indicated as “OAT”) was 94.5°F during the 8 month period, whereas the maximum RW temperature was 83.2°F during the same period. On the other hand, the monthly average OA wet bulb temperature at Denver is always below 60°F during the same period, which indicates that a wet cooling tower would be very effective to cool the source water in this climate. The minimum RW temperature (38.8°F) was much higher than the minimum OAT (7.6°F). A closer look at the OA and RW temperatures from July 15 through 17 reveals that whereas the RW supply temperature (indicated as “TRWS”) was higher than the OAT in the nighttime, it was lower than the OAT in the daytime when cooling demands were high (see the upper chart in Fig. 3). Since a lower heat sink temperature will lead to higher cooling efficiency in a heat pump, using RW as a heat sink during the daytime can result in lower cooling energy consumption than using OA. Furthermore, as shown in the lower chart of Fig. 3, TRWS was higher than OAT all the time during typical days in winter (January 20–22), which indicates RW is a better heat source than OA.
Fig. 2. Hourly OA temperature vs. RW temperature.

Fig. 3. Outdoor air (OA) dry bulb temperature, recycled water (RW) temperature, and water-to-water heat pump (WWHP) loads during typical days in the cooling season (top) and heating season (bottom).
Heat Flow Analysis

Heat flows from the various heat sinks and sources in the source water loop were analyzed to quantify their contributions. Figure 4 shows the monthly heat flows, which are grouped into two categories: heat extracted from the various heat sources (with positive values) and heat rejected to various heat sinks (with negative values). As can be seen in this figure, the heat flows demonstrated two different patterns. During heating season (i.e., December 2014 through May 2015), most heat (~70%) was extracted from the precooling loop and the rest was extracted from the RW and the steam HX. While most of these additions were used by the WWHPs to generate HW, a fair amount of heat was rejected to the cooling tower and the RW. The purpose of the heat rejection was to make the source water cool enough to precool the air in the AHUs. In contrast, during cooling season (June through August, 2015), roughly an equal amount of heat was rejected by the WWHPs (i.e., the condensing heat from the cooling operation of the WWHPs) and the precooling loop. Only a small amount of heat was extracted from the RW. Most of the heat (~60-70%) added to the source water was rejected to the ambient air by the wet cooling tower and about 20–30% of the heat was rejected to the RW. The rest was used by the WWHPs when they ran in heating mode.

System COP Analysis

Because the RWHP system provided simultaneous cooling and heating, the “effective COP,” which accounts for both heating and cooling operation, was calculated to evaluate the performance of the WWHPs and of the entire RWHP system. Figure 5 presents the effective COP for the WWHPs and the RWHP system for each 10°F bin of the OAT. It shows, in general, the effective COP increases with the increase in OAT. The effective COP of the WWHPs was about 5.6–6.0 when the OAT is higher than 70°F (i.e., when most modules of the WWHPs ran in cooling mode). This is consistent with the manufacturer’s catalog data, which indicates that the cooling COP of the WWHP ranges within 5.9–6.2 under similar operating conditions (i.e., 75°F water entering condenser and 45°F water leaving evaporator). The effective COP of the entire RWHP system—which accounts for the supplemental heat input from the steam HX in the source loop as well as the power consumptions of the cooling tower, the RW pump, and the source loop pumps—rose from 2.6 to 4.4 with the increase in OAT, which is coincidental with the increased simultaneous heating and cooling operation.
Energy and Cost Savings Potential

To estimate the energy saving potential of the RWHP system, the energy consumption of a conventional VAV system using a water-cooled chiller and a natural gas boiler was calculated as a baseline for providing the same heating and cooling outputs as the RWHP system. The energy efficiency of the chiller and the boiler used in the baseline were the minimum values allowed by ASHRAE 90.1-2013. The following are the major assumptions for calculating the baseline energy consumption and energy savings:

- It is assumed that there is no difference in the power consumption of the source loop pumps, CHW loop pumps, and HW loop pumps between the baseline and the RWHP systems.
- The water-cooled chiller has a nominal cooling COP of 5.54. A generic performance curve for water-cooled chillers was adopted from the DOE-2 program (Hirsch et al. 2016) and used to calculate the chiller power consumption for providing the same hourly cooling output as the RWHP system (including outputs from both the precooling and CHW loops).
- The natural gas–fired boiler has a thermal efficiency of 80% and provides the same hourly heating outputs as both the WWHPs and the steam HXs in the HW loop.
- The cooling tower power consumption of the baseline system is calculated based on the average heat rejection efficiency of a typical cooling tower, which depends on the average wet-bulb temperature in each month.
- Average utility rates obtained from 2014 and 2015 utility bills were used for energy cost calculations. The average electricity rate is $0.076/kWh and the natural gas rate is $4.84/MMBtu.

The source energy consumption and carbon emissions of the two systems were calculated based on the measured consumption of electricity and natural gas by the RWHP system and the simulation-predicted electricity and natural gas consumption of the baseline system. The site-source energy conversion factors and the emission factors of electricity and natural gas suggested by Deru and To cellini (2007) were used in the calculations. Table 2 shows the annual performance comparison between the baseline and the RWHP systems. The analysis shows that the total annual energy cost savings would be $16,295 (34% savings), and CO2 emission reductions associated with the energy savings would be approximately 41%.

For the 8 months encompassed in this study, the RWHP system saved 2,507 MMBtu of source energy (a 47% savings) and $11,386 in energy costs (a 37% reduction) compared with the baseline system. Considering the costs associated with using RW ($0.078 per 1,000 gallons of RW passing though the heat exchanger) and additional cooling tower make-up water for the baseline system, which are about $807 and $500, respectively, the operating cost savings...
of the RWHP system is $10,157 (33% savings).

Because the available measured data cover only 8 months, the energy use of the two systems during the rest of a 1-year period (September through December, 2014) was estimated to assess the annual energy savings potential of the RWHP system. (The results are listed in Table 2.) The estimation procedure included two steps:

- Derive correlations between the monthly energy use of each major component of the two systems and the monthly average OAT based on available data from January through August in 2015
- Estimate the energy use of the two systems with the derived correlations and the historical OAT data from September through December 2014.

Table 2. Comparison of Annual Performance between Baseline and RWHP Systems

<table>
<thead>
<tr>
<th></th>
<th>Baseline system</th>
<th>RWHP system</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Electricity</td>
<td>Natural gas</td>
</tr>
<tr>
<td>Annual HVAC related site energy (kWh)</td>
<td>331,509</td>
<td>4,742 MMBtu</td>
</tr>
<tr>
<td>Annual HVAC related source energy (MMBtu)</td>
<td>8,191</td>
<td>4,098</td>
</tr>
<tr>
<td>Source energy savings (MMBtu)</td>
<td>–</td>
<td>4,094</td>
</tr>
<tr>
<td>% of source energy savings</td>
<td>–</td>
<td>50.0%</td>
</tr>
<tr>
<td>Energy cost by fuel type ($)</td>
<td>$25,195</td>
<td>$22,953</td>
</tr>
<tr>
<td>Total energy cost ($)</td>
<td>$48,148</td>
<td>$30,108</td>
</tr>
<tr>
<td>Recycled water use ($)</td>
<td>–</td>
<td>$2,398</td>
</tr>
<tr>
<td>Additional make-up water ($)</td>
<td>$654</td>
<td></td>
</tr>
<tr>
<td>Annual cost savings ($)</td>
<td>–</td>
<td>$16,295</td>
</tr>
<tr>
<td>% of cost savings</td>
<td>–</td>
<td>33.8%</td>
</tr>
<tr>
<td>CO₂ emissions (lb) by fuel type</td>
<td>543,675</td>
<td>578,565</td>
</tr>
<tr>
<td>Total CO₂ emissions (lb)</td>
<td>1,122,339</td>
<td>666,725</td>
</tr>
<tr>
<td>CO₂ emission reductions (lb)</td>
<td>–</td>
<td>455,515</td>
</tr>
<tr>
<td>% of CO₂ emission reductions</td>
<td>–</td>
<td>40.6%</td>
</tr>
</tbody>
</table>

The normalized cost of the RWHP system (including the AHUs and the ductwork system inside the building) is $25,210/ton of installed cooling capacity, or $37.8/ft² of building floor space. With the achieved annual energy cost savings, the simple payback for this system is about 58 years. This long payback period is due to the high cost of constructing a 3,300 ft long two-way pipeline to access the RW. The pipeline cost is $1.1 million, which accounts for about 20% of the total system cost. If the length of the pipeline were 1,000 ft, the simple payback would be reduced to 11 years.

CONCLUSION

Energy Performance and Cost-Effectiveness

- The measured RW temperatures from the demonstration site during the encompassed period show that RW was more favorable than OA for effective operation of the WWHPs. The maximum OAT was about 94.5°F during the cooling season, whereas the maximum RW temperature was about 83.2°F. The lowest RW temperature was about 38.8°F during the heating season, whereas the lowest OAT was below 10°F.
- Effective COPs of the WWHPs and of the entire RWHP system, which account for the simultaneous cooling and heating, were calculated based on the measured data. The effective COP of the WWHPs ranged from 4.6 to 6.0, whereas the effective COPs of the RWHP system ranged from 2.6 to 4.4 during the 8 month
investigative period of this study. The system COP increased with the increase in OAT, which is in part a result of the increased simultaneous heating and cooling demands.

- The demonstrated RWHP system saved 4,094 MMBtu of source energy (a 50.0% savings) and $16,295 in energy costs (a 33.8% savings) annually, compared with a conventional VAV system using a water-cooled chiller and a natural gas–fired boiler, both of which meet the minimum energy efficiencies allowed by ASHRAE 90.1-2013. The energy savings also resulted in a 41% reduction in CO2 emissions.
- The simple payback period of this system is about 58 years. However, if the length of the pipeline for accessing the RW were shortened from 3,300 ft to 1,000 ft, the simple payback would be reduced to 11 years.

**Lessons Learned**

- The run-around heat recovery through the precooling loop was very effective and significantly reduced the demand for external heat sources (e.g., the RW and the steam HX). The wet cooling tower rejected more heat than the RW because of the relatively dry air in the Denver area.
- Contributions of the steam HXs were very small, and it is likely that the RWHP system could work well without the boiler and the steam HXs, or with just a smaller water heater as a backup. Such a configuration would reduce the complexity and the associated cost of the RWHP system.
- The RWHP system would be economically more competitive if the RW were closer to the building.

**ACKNOWLEDGMENTS**

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Can ground source heat pumps perform well in Alaska?

Robbin Garber-Slaght  Rorik Peterson

ABSTRACT
The long heating season and cold soils of Alaska provide a harsh testing ground for ground source heat pumps (GSHPs), even those designed and marketed for colder climates. Fairbanks, Alaska has 7,509°C heating degree-days (13,517°F HDD24) and only 40°C cooling degree-days (72°F CDD24). This large and unbalanced heating load creates a questionable environment for GSHPs. In addition, soil temperatures average around freezing (0°C/32°F); the soil may be permafrost year-round, just above freezing, or in an annual freeze-thaw cycle. In 2013 the Cold Climate Housing Research Center (CCHRC) installed a GSHP at its facility in Fairbanks. The heat pump replaced an oil-fired condensing boiler heating a 464 m² (5,000 ft²) office space. The ground heat exchanger was installed in a marginal area underlain with permafrost near 0°C (32°F). The intent of the installation was to observe and monitor the system over a 10-year period in order to develop a better understanding of the performance of GSHPs in ground with permafrost and to help inform future design. The system enjoyed one season of better-than-expected performance, averaging a COP of 3.7 its first winter. By the third winter, the COP had dropped to an annual average of 3.2 and ice had started to develop in the area around the heat extraction coils. A combination of physical monitoring and numerical modeling is used to evaluate the heat pump system.

INTRODUCTION
The Cold Climate Housing Research Center and the Alaska Center for Energy and Power (ACEP) completed a study on the state and use of GSHP technology in cold climates in 2011 (Meyer, et al.). They found that even with high capital costs of GSHPs, a ground source system with a minimum COP of 2.5 can be cost effective in several parts of Alaska. The study also found that the use of GSHPs is increasing in cold climates as the technology improves, but determined that there is a lack in long term studies in cold climates. Of particular interest is the long term effect of unbalanced heat extraction on the soil surrounding the ground heat exchanger (GHE) and the degradation in the efficiency of the heat pump system.

Wu, et al. (2013) point out that the soil temperatures around a GHE in a heating dominated climate can degrade over time, which lowers the efficiency of the GSHP. There are several approaches to address this problem: increasing the GHE size, installing a secondary heating source, and using thermal storage (Wu et al., 2013). You, et al. (2016) also suggest several ways to improve GHE performance. Increasing the size of the GHE or changing the layout of boreholes can mitigate thermal imbalance slightly, but is not effective for a larger imbalance. Modifying the heat pump system itself to include auxiliary heating sources or using the heat pump only at certain times of the day when other heating options are not available is a practical and effective way to help the heat pump maintain efficiency (You et al., 2016).

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Prior to this CCHRC study, the longest GSHP study in Alaska lasted for 1.5 years and concluded that the soil temperatures with GHE installed at 1.2 m (4 ft.) of depth recovered early in the summer season (Nielsen & Zarling, 1983). A Southcentral Alaska study that lasted one year found that deeper (2.7 m, 9 ft.) heat extraction coils did not recover temperature completely in the summer months, whereas the shallower (1.5 m, 5 ft.) coils recovered well (Mueller & Zarling, 1996).

Phase changes from liquid to solid in the soil have the potential to improve energy transfer and prolong the life of the GHE. Eslami-nejad & Bernier (2012) found that by saturating soils around the GHE boreholes, energy from the freezing phase change was added to the heat pump system. Yang, et al. (2015) found that freezing in the GHE area can enhance the heat transfer performance and can help downsizing the heat exchanger.

Razaei, et al. (2012) studied the effects of surface treatments on the soil temperatures in a GHE. A layer of tire derived aggregate 0.2 m (0.6 ft.) deep has effects on the soil temperatures down to 4 m (13 ft.). The surface layer of aggregate had the potential to increase the energy absorbed from the GHE by 17% over no surface treatment and the aggregate performed better in cold climates.

THE CCHRC HEAT PUMP

CCHRC’s Research and Testing Facility (RTF) is located on the campus of the University of Alaska Fairbanks (UAF). Fairbanks has 7,509°C HDD (13,517°F HDD) and 40°C CDD (72°F CDD); the 99.6% design temperature is -41.9°C (-43.5°F) (ASHRAE, 2013). Fairbanks is in a zone of discontinuous and warm permafrost. The area surrounding the RTF is an open field that was cleared of native vegetation more than 60 years ago and is made up of moist silt (Shannon & Wilson, Inc., 2002). The permafrost underlying the RTF has been degrading since the land was first cleared. In 2006, the top of the permafrost layer on the site ranged from 3 to 7.3 m (10 to 24 ft.).

The CCHRC RTF is 2,044 m² (22,000 ft²) with 3 distinct heating sections. The heat pump was sized to heat the 464 m² (5,000 ft²) office space on the east side of the building with a design heat load of 17.5 kW (60,000 BTU/hr.). Heat is distributed to the office space via in-floor hydronic tubing that is embedded in concrete. The office space has 9 thermostatic controlled zone valves. The heat pump system replaced a 22.3 kW (76,000 BTU/hr.) oil fired condensing boiler and a wood fired masonry heater as the main source of heat for this portion of the building; the masonry heater is still used for supplemental space heating and ambiance.

The soils around the RTF have been extensively surveyed in the past 20 years for road and building construction. This survey information was used to inform the design of the GHE. Test boreholes drilled on the site in 2006 prior to the construction of the RTF found the site underlain with a sloping layer of permafrost. The top of the layer started at 3 m (10 ft.) on the south side of the building and sloped down to 9.1 m (30 ft.) on the north side. Data collected under the building since 2006 shows that the top of the permafrost has further degraded 0.6 m (2 ft.). A test borehole in 2012 prior to installing the ground loop did not find permafrost within 9.1 m (30 ft.) of the surface.

A soil thermal conductivity test was conducted in October 2012, one year prior to the installation of the heat pump. A 34 m (112 ft.) long horizontal loop at 2.7 m (9 ft.) of depth was installed 12 days prior to the testing. The test duration was 48 hours. The soil thermal conductivity was found to be 1.42 W/m·K (0.82 Btu/hr·ft·°F). The thermal diffusivity was estimated to be 0.055 m²/day (0.59 ft²/day).

Originally, CCHRC wanted to demonstrate both deep wells and horizontally trenched GHEs. However, test bores for Thompson Drive (about 183 m (200 yards) from the ground loop field) construction in 2001 discovered frozen schist bedrock from 19.5 m (64 ft.) down to 45.7 m (150 ft.) (the bottom of the boreholes). The frozen schist was -6.7°C (20°F), which was deemed too cold to use in this demonstration. A horizontal GHE was designed based on the technology available in Fairbanks at the time. Since directionally drilling was not an option, horizontal slinky coils were developed. Six 30.5 m (100 ft.) long by 1 m (3 ft.) wide slinky coils with an 0.5 m (18 in.) pitch were installed 1.8 m (6 ft.) apart (see Figure 1). Overall, 1,463 m (4,800 lineal ft.) of 1.9 cm (3/4 in) HDPE was installed at 2.7 m (9 ft.) depth to create the in-ground heat exchanger. The GHE size and depth were determined based on knowledge of past installations in the area, in conjunction with ground thermal conductivity test data, and information...
The depth of the GHE is greater than recommended by the Mueller & Zarling (1996) and Nielsen & Zarling (1983) Alaskan studies. The depth was chosen to be below the line of seasonal frost and above the top of the permafrost. In addition, the 2.7 m (9 ft.) depth is the typical installation depth for residential horizontal GHE in the Fairbanks area. Part of the finite element analysis portion of this study is to determine an optimum depth for the ground heat exchanger.

In addition to determining an optimum heat exchanger depth, this study is also looking into whether different ground coverings are more advantageous for energy recharge. Three different coverings are being evaluated: dark rocks, sand, and grass. Each treatment covers 2 slinky loops (see Figure 1). The soil temperatures under the coverings are monitored as are the temperatures of the fluid returning from the coils.

The heat pump itself is a residential 21 kW (6 ton) water to water unit, chosen based on previous experience with the model in Fairbanks. It is connected to the existing in-floor heat delivery system. The heat pump heats a 303 liter (80 gallon) buffer tank of water to a temperature determined by outdoor air temperature. The minimum temperature for the buffer tank is 26.7°C (80°F) and the maximum is 42.8°C (109°F). The GHE side of the heat pump is charged with a 20% methanol, 80% water mixture. The building hydronic side of the heat pump is charged with water. All energy flows to and from the heat pump are monitored.

Figure 1  CCHRC’s heat pump. The schematic includes the location and type of data collection sensors used in the project. The temperature sensors across the GHE also extend vertically down to create a soil temperature profile. The center vertical temperature string measures into the ground at 0.3 m, 0.6 m, 1 m, 2.1 m, 2.9 m, 3.5 m and 4.1 m (1 ft., 2 ft., 3.5 ft., 7 ft., 9.5 ft., 11.5 ft., and 13.5 ft.). The other four temperature strings are spaced at 0.25 m, 0.75 m, 1 m, 1.5 m, 2 m, 2.7 m, 3.2 m, and 3.7 m (0.8 ft., 2.5 ft., 3.3 ft., 5 ft., 6.5 ft. 9 ft., 10.5 ft., and 12 ft.). There are 50 in-ground temperature sensors in total.
CCHRC GSHP RESULTS

Temperatures recorded in and around the GHE show a steady cooling of the ground over the three years the heat pump has been in use. The temperatures in the vicinity of the heat extraction coils are colder than the baseline temperatures in the adjacent field. The temperature at the depth of the coils shows 0°C (32°F) most of the winter, the baseline temperatures are 3 to 4°C (5.4 to 7.2°F) higher. The temperature has not dropped below freezing as energy of phase change is extracted from the surrounding soils, freezing the soils before the temperature drops further. Thus far the soil around the loops has risen above freezing each summer. Figure 2 shows the ground temperatures over the three years. The temperatures at the depth of the GHE (2.7 m, 9 ft) have remained close to the freezing line since December 2015. They have not dropped much below the freezing point in three years, but could if permafrost develops in the GHE area.

Figure 2  Whiplash curves of the center of the GHE for 3 years. Temperatures do not drop below freezing but as permanent ice develops, colder temperatures may result. Each year starts in September and ends in August.
Permafrost. Permafrost tubes in the GHE show some ice creation within the slinky coil in the center of the GHE. The ice does not spread beyond the slinky coil, as far as can be determined from the permafrost tubes. Ice developed around the center slinky coil in February 2016 and grew to 1.4 m (4.5 ft.) of ice on, above, and below the slinky coil (there is an undetermined amount of error on the permafrost tubes). The ice had melted by July 2016.

Surface treatments. Temperatures in the GHE show the effects of the differing surface treatments. Deeper down in the GHE the effects of the surface treatments are harder to discern, especially around the slinky coils where the heat extraction has an overwhelming effect on the ground temperatures. However, the dark gravel is keeping that section of the GHE warmer than the sand or the grass. The temperatures in the manifold (Figure 3) show that the fluid returning from the gravel loops is always slightly warmer than the other two surface treatment loops. The differences in the surface treatments are noticeable in the fall of 2015, with the gravel 0.5 C° (0.9 F°) warmer than the sand loops and 1 C° (1.8 F°) warmer than the grass loops. As the winter progressed the gravel loops stayed warmer than the other loops but there was less of a difference. The sand loops ended the winter season with the coldest temperatures.

Figure 3
Temperatures of the fluid returning from the GHE. The temperatures in the beginning of 2014 do not have the effects of the summer on the surface treatments as the system was not complete until September 2013. Holes in the data are due to no fluid returning to the building. Each surface treatment has 2 coils returning to the building.

Efficiency. The efficiency of the heat pump varied over the course of each heating season. It tended to be higher in autumn when the GHE was the warmest and fell over the course of the winter. However, as the heating demand of the building lessened, the COP improved as the heat pump delivered lower temperature heat to the building. Figure 4 shows the trend for the COP. The COP for the heat pump is trending slightly lower over time, with a 14% decline in the annual average over 3 years. Although the average decline from year 2 to year 3 was less significant at 3%.
**GHE NUMERICAL MODEL**

A numerical finite-element model was constructed to simulate the purely conductive heat transfer with water content phase change behavior of the ground soil surrounding the GHE using Comsol Multiphysics 5.2a with a PARDISO solver. Physical properties (e.g. porosity) were based on field measurements and the corresponding thermal properties (e.g. heat capacity) used Kersten correlations (Andersland and Ladanyi, 2004). Phase change behavior was numerically implemented using a pseudo-heat capacity method over a finite yet narrow temperature range immediately below 0ºC (32ºF). Model simulations were run for 10 years starting from an initial soil temperature distribution recorded just after installation. Time stepping is adaptive, and a more restrictive maximum step of 3,600 seconds was added to assure phase change behavior was not “stepped over” by the pseudo-heat capacity implementation.

Both a two- and three-dimensional models were built. The two-dimensional model was a symmetric vertical cut from the ground surface extending 30 m (98.4 ft.) down and 2 m (6.6 ft.) wide; with the line of symmetry directly through the center-line of the GHE. This geometry was arrived at using the three-dimensional model which indicated that the heat flux and all associated physical phenomena were primarily in the vertical direction except in the immediate vicinity of the heat-extraction coils. Here, there is some off-vertical heat that extended less than one meter horizontally. Therefore the 2-meter wide (4 m (13.1 ft.) due to symmetry) system was able to adequately capture the thermal behavior at a substantial computational saving.

The model parameters and boundary conditions were specified to approximate the conditions at the test location as close as possible without introducing substantial and unnecessary complexity. In some instances, analytical functions were used to approximate actual conditions. For example, the air temperature at the top boundary of the model domain was modeled as a sinusoidal function of period 1 year with an amplitude, phase offset, plus a constant component to match the climatic average at the testing location. The heat pump thermal demand (from the ground) was modeled by approximating a truncated sinusoidal function with zero demand in the warmer months and was based on the actual heat supplied to the building in year 2 of the study. The same profiles were used for each year of

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**Figure 4** Heat pump COP over time. The COP calculated for May 2015 is an outlier that is not explained with the existing data, however some of the zone valves for the in-floor delivery were stuck open which could potentially explain odd data.
the 10-year simulations. A sensitivity analysis of building demand (e.g. temperature) was not completed.

**NUMERICAL MODEL RESULTS**

Figure 5 shows the modeled temperature at several depths (in meters) over the initial 3 year period of the simulations (from the 2-D model at the center of the GHE). The depths closely approximate the temperature sensor depth locations in the field study to within 5 cm. There is a gradual decline in temperature near the heat pump energy extraction depth (2.7 m, 9 ft.), with more consistent periodic temperatures at the shallower and deeper depths. This repeatability from year to year is due to the fact that the same air temperature profile (climatic average) was used for each year of the simulation. The simulated temperatures at depth agree fairly well with the recorded temperatures (the annual average is within 0.09°C (0.16°F) at 3.5 m (11.5 ft.) and 0.23°C (0.41°F) at 2.9 m (9.5 ft.) in the third year of the model and study) considering the approximations of the heat demand and surface boundary conditions. Model simulations were run for 10 years, and stabilization was generally observed after 5 years as the longer-term modeled temperatures shown in Figure 5 indicate.

![Figure 5](image_url)

**Figure 5** Modeled temperature over time. Days start in January and run out for seven years. The depth slightly above the GHE coils at 2.7 m (9 ft.) shows a slight below-freezing temperature at its coldest; around -1°C (30.2°F), while the actual data recorded are around 0°C (32°F) indicating the soil is at the freezing point. The simulation causes the model to indicate slightly below freezing temperatures during phase change; any temperature around 0 to -1°C indicates phase change.

The heat pump COP varies as a function of the heat absorption reservoir temperature, and the equipment manual provides this performance information for temperatures from -7°C to 21°C (20°F to 70°F). In order to simulate the varying performance as the ground temperature varied throughout the simulation, a least-squares linear fit to the performance data was obtained. This was then used to specify the expected COP. Actual recorded COP values were slightly higher than those predicted by the simulation, and may be due to several different factors such as the building heat supply temperature. The performance data used for the model was based on an assumed constant building supply temperature of 38°C (100°F).

To assess the expected performance of the heat pump if the GHE had been installed at a different depth, the heat
removal term in the model (the depth of the GHE coils) was placed at different depths in the model domain. Depths ranging from 1 to 4 m (3.3 to 13 ft.) (every 0.5 m, 1.6 ft.) were examined. Figure 6 shows the yearly average expected COP of the heat pump system if the GHE coils had been installed at these depths. The simulations indicate that deeper depths result in higher COP, with a diminishing return after 2 to 2.5 m (6.6 to 8 ft.). The performance at all depths stabilizes after about 5 years. It is interesting to note that the shallowest depth (1 m, 3.3 ft.) has slightly higher average COP than 1.5 m (4.9 ft.) after 10 years (although both depths have significantly low COP values). This is likely due to some small amount of passive surface energy recharge during the warmer summer months. Although this effect is small, the implication seems to be that surface energy recharge is not significant in the natural state at depths below around 1 meter, regardless of the three different surface treatments. Active recharge is required at depths below around 1 meter in order to get the natural heat to penetrate to those depths, such as reverse operation of the GHE coils, or a separate heating system such as a solar-thermal system.

Figure 6 Modeled annual COP. Each bar represents a different depth of the GHE coils from the surface.

CONCLUSIONS

Data from the first 3 years of heat pump operation show a 14% decline in COP; however the drop in COP is only 3% from year 2 to year 3, potentially indicating stabilization in the near future. The numerical modeling suggests that the decline will level out around year 5, and that the COP will not be much lower than 3.4. The original ACEP/CCHRC study found that a GSHP with a COP of 2.5 or greater would be cost effective in Fairbanks (Meyer et al., 2011). Three years of operation is certainly not long enough to see all the changes the heat pump will create in the soil thermal regime. This heat pump demonstration will be monitored for at least another 7 years to verify the degradation in the soil temperatures and the COP. If the loss in efficiency becomes too burdensome to the cost to operate the heat pump, methods to bring the ground loop into better thermal balance may be instituted.

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Examination of Ambient Temperature Variations Effects on Predicted Fluid Temperatures in Vertical Boreholes

Massimo Cimmino

ABSTRACT
A fully-coupled model of geothermal bore fields is presented, taking into account the axial variations of borehole wall temperatures and heat extraction rates along the borehole lengths, fluid temperature variations inside the U-tubes and depth-variation of ground temperature due to seasonal ambient temperature variations. The model incorporates an analytical finite line source model to calculate borehole wall temperatures. A steady-state analytical solution is used to calculate fluid temperatures from arbitrary borehole wall temperature profiles. Ground temperatures are calculated from a one-dimensional analytical solution to heat conduction in semi-infinite media with varying surface temperature. A system of equations is built and solved in the Laplace domain. The specific case of boreholes connected in series is discussed. Different assumptions for time- and depth-variation of ground temperatures are compared. Results show that time-varying uniform ground temperatures is accurate for simulations, with differences in predicted fluid temperatures within 1.5% of time-varying non-uniform ground temperatures.

INTRODUCTION
The design of geothermal bore fields coupled to geothermal heat pump systems relies on the accurate simulation of fluid temperatures in the boreholes. At short time scales, heat transfer processes in geothermal boreholes are affected by the thermal capacities of the circulating fluid and the grout material as well as the travel time of the fluid inside the boreholes. The heat interaction between the boreholes and the ground is then radially one-dimensional. At larger time-scales, axial conduction effects due to the finite borehole length and thermal interaction between boreholes become significant. The ground surface temperature is often considered constant and equal to the undisturbed ground temperature in analytical models, since the penetration depth of seasonal ground temperature variations due to varying ambient temperatures covers only a small portion of the borehole length. For short boreholes, however, the ground temperature variation covers a greater portion of the boreholes and may significantly impact on the outlet fluid temperatures. This study examines the effect of ambient temperature variations on the predicted fluid temperatures in geothermal boreholes.

Fluid and ground temperature variations in geothermal bore fields are typically calculated by the temporal superposition of thermal response factors, or g-functions. Eskilson (1987) evaluated thermal response factors from
numerical finite difference simulations. These thermal response factors represent the dimensionless temperature drop in a bore field with constant total heat extraction rate. g-Functions are implemented in several ground heat exchanger design tools and energy simulation programs (Hellström and Sanner 1994; Spitler 2000; Fisher, et al. 2006; Liu and Hellstrom 2006).

Finite line source solutions (Zeng, et al. 2002; Lamarche and Beauchamp 2007; Claesson and Javed 2011) have been proposed to estimate g-functions and simulate geothermal bore fields. Since the finite line source solution assumes a uniform heat extraction or injection rate along the boreholes, contrary to the uniform borehole wall temperature used by Eskilson (1987), thermal response factors obtained from the finite line source solution tend to overestimate Eskilson’s g-functions. Cimmino et al. (2013) and Cimmino and Bernier (2014) proposed a method, based on the finite line source, to consider the variation of the heat extraction rates along the borehole lengths and obtain thermal response factors for uniform borehole wall temperatures. A similar approach was used by Lazzarotto (2016) and Lazzarotto and Björk (2016) for tilted boreholes. The finite line source method has also been applied to geothermal boreholes with groundwater advection (Molina-Giraldo, et al. 2011) and boreholes connected in series (Marcotte and Pasquier 2014).

The condition of uniform borehole wall temperature may lead to inaccuracies when simulating geothermal bore fields. Cimmino (2015) proposed a method based on the finite line source solution to calculate thermal response factors of bore fields with equal inlet fluid temperature for all boreholes. The finite line source solution was coupled to a steady-state solution of the fluid temperature profiles in the U-tube pipes. It was shown that the value of thermal response factors are sensitive to the borehole thermal resistance. Furthermore, the condition of constant ground surface temperature may also lead to inaccuracies when predicting fluid and ground temperatures in geothermal systems (Bandos, et al. 2009; Bandos, et al. 2011; Rivera, et al. 2015; Zarrella and Pasquier 2015; Rivera, et al. 2016).

![Figure 1](image)

**Figure 1** Assumptions used in the calculation of ground temperatures $T_g$, borehole wall temperatures $T_b$ and fluid temperatures $T_f$

While it is recognized that ambient air temperature variations influence ground temperature conditions in geothermal bore fields, it is not clear how much returning fluid temperatures from the boreholes are affected, especially for short boreholes. This study aims to quantify the impact of ambient air temperature variations on
predicted fluid temperature and its relation with the borehole length and buried depth. Different assumptions for the time- and depth-variation of ground temperatures along the boreholes due to ambient temperature variations are considered, as shown on fig. 1: (1) time-varying non-uniform ground temperature, (2) time-varying uniform ground temperature and (3) constant uniform ground temperature. The relative difference between each assumption on the predicted fluid temperatures are assessed from a simulation case study of a residential geothermal heat pump system. Predicted fluid temperatures are compared for different values of borehole buried depth and borehole length. The validity of each assumption is reported.

MATHEMATICAL MODEL

The analytical model of Cimmino (2016) is adapted to simulate single U-tube vertical geothermal boreholes in series and is coupled to an analytical solution of the ground temperature variation due to varying ambient air temperatures.

Ground Model

Each borehole is divided into $n_q$ segments of equal lengths. Each of the borehole segments are modeled using the finite line source solution, using one line source per borehole segment. The borehole wall temperature of each segment is obtained by the temporal and spatial superpositions of the finite line source solution. The borehole wall temperature along each of the segments of a borehole $i$ are given by:

$$\mathcal{L}\left( T_{bi}(t) \right) = \mathcal{L}\left( T_g(t) \right) - \sum_{j=1}^{n_q} \mathcal{L}\left( \Delta h_{ij}(t) \right) \cdot \mathcal{L}\left( Q_{bi}(t) \right) / \left( 2\pi k_s H / n_q \right) \quad (1)$$

where $T_{bi}(t) = [T_{b,i,u}(t)]$ is a column vector of the borehole wall temperature of each borehole segment $u$ of borehole $i$, $T_g(t) = [T_{g,u}(t)]$ is a column vector of the average ground temperature along the length of each borehole segment $u$, $Q_{bj}(t) = [Q_{b,j,v}(t)]$ is a column vector of the heat extraction rates of each segment $v$ of borehole $j$, $\Delta h_{ij}(t) = [\Delta h_{ij,u,v}(t)]$ is a square matrix of the segment-to-segment response factor increments between segments of borehole $j$ and borehole $i$. $\mathcal{L}(f(t))$ represents the direct Laplace transform of a function $f(t)$. Information on the use of numerical Laplace transforms for the calculation of borehole wall temperatures is given in (Cimmino, et al. 2013; Cimmino and Bernier 2014; Cimmino 2015; Cimmino and Bernier 2015). Segment-to-segment response factor increments are given by the finite line source solution (Claesson and Javed 2011; Cimmino and Bernier 2014).

Surface Effects

The ground temperature is calculated from the analytical solution to transient heat conduction in semi-infinite media with varying surface temperature, provided by Carslaw and Jaeger (1959). The analytical solution is given by:

$$\mathcal{L}(T_g(z,t) - T_{g,0}) = \mathcal{L}(T_{air}(t) - T_{g,0}) \cdot \mathcal{L}(F_g(z,t)) \quad (2)$$

$$f_g(z,t) = \frac{z}{\sqrt{\pi \alpha_s t}} \exp \left( -\frac{z^2}{4\alpha_s t} \right) \quad (3)$$

where $T_{g,0}$ is the initial uniform ground temperature and $T_{air}(t)$ is the varying ambient air temperature, assumed equal to the ground surface temperature. The ground temperature is averaged over each segment:

$$\mathcal{L}\left( T_g(t) - 1_{n_q} T_{g,0} \right) = \mathcal{L}(T_{air}(t) - T_{g,0}) \cdot \mathcal{L}(F_g(t)) \quad (4)$$
\( F_g(t) = \begin{bmatrix} F_{g1}(t) \\ F_{g2}(t) \end{bmatrix} \) is a column vector of the functions \( F_{g,i}(t) \) and \( 1_{nq} \) is a \( n_q \times 1 \) all-ones column vector. The average ground temperature over the borehole lengths is calculated in the same manner:

\[
\mathcal{L}(T_g(t) - T_{g0}) = \mathcal{L}(T_{ai}(t) - T_{g0}) \cdot \mathcal{L}(S_{g}(t))
\]

where \( T_{g}(t) \) is the average temperature over the length of the boreholes.

**Borehole Model**

Fluid temperatures in the boreholes are calculated from the steady-state line source approximation (Hellstrom 1991). An analytical method was recently proposed by Cimmino (2016) for boreholes with any number of U-tubes and varying borehole wall temperatures. The solution is adapted here for single U-tube boreholes. Borehole thermal resistances give the relation between the fluid and borehole wall temperatures and the pipe heat transfer rates:

\[
T_{f,i}(z,t) - 1_T b_i(z,t) = R q_i(z,t)
\]

where \( T_{f,i}(z) = \begin{bmatrix} T_{f,i,1}(z) \\ T_{f,i,2}(z) \end{bmatrix} \) is column vector of the fluid temperature in each pipe of borehole \( i \), with \( T_{f,i,1}(z) \) the temperature in the downward flowing pipe and \( T_{f,i,2}(z) \) the temperature in the upward flowing pipe, \( q_i(z,t) = \begin{bmatrix} q_{i,1}(z,t) \\ q_{i,2}(z,t) \end{bmatrix} \) is a column vector of the heat transfer rates of each of the pipes in borehole \( i \), and \( R = \begin{bmatrix} R_{11} & R_{12} \\ R_{21} & R_{22} \end{bmatrix} \) is the matrix of borehole thermal resistances, with coefficients calculated from the line source approximation (Hellstrom 1991).

The inverse of eq. (8) and the thermal energy balance of the fluid along the pipe length yields a system of differential equations:

\[
\frac{\partial T_{f,i}}{\partial z}(z,t) = A T_{f,i}(z,t) - A 1_T b_i(z,t)
\]

where \( A = \begin{bmatrix} A_{11} & A_{12} \\ A_{21} & A_{22} \end{bmatrix} = \begin{bmatrix} -S_{11}/\dot{m}_f & -S_{12}/\dot{m}_f \\ S_{21}/\dot{m}_f & S_{22}/\dot{m}_f \end{bmatrix} \) is the coefficient matrix of the system of differential equations, with \( R^{-1} = \begin{bmatrix} S_{11} & S_{12} \\ S_{21} & S_{22} \end{bmatrix} \) the inverse of the borehole resistance matrix.

The fluid temperature profiles are given by the matrix exponential of \( A z \). The fluid temperature profiles in borehole \( i \) with uniform borehole wall temperature along each of the borehole segments is given by (Cimmino 2016):

\[
T_{f,i}(z,t) = E(z) T_{f,i}(0,t) - \sum_{v=1}^{nq} \int_{(n-1)H}^{nH} E(z-z') A 1_T b_{i,v}(t) dz'
\]

where \( E(z) = \begin{bmatrix} E_{11}(z) & E_{12}(z) \\ E_{21}(z) & E_{22}(z) \end{bmatrix} = \exp(A z) = \begin{bmatrix} \exp(D z) \end{bmatrix} V^{-1} \) is the matrix exponential of \( A z \), \( D \) is a diagonal matrix of the eigenvalues of \( A \) and \( V \) is the matrix of the column eigenvectors of \( A \).

The relation between inlet and outlet temperatures is obtained by imposing equal fluid temperatures \( T_{f,i,1} = \)
\( T_{f,i,2} \) at the bottom end of the borehole \( z = H \):

\[
E_{\text{out}}(H)T_{f,\text{out},i}(t) = E_{\text{in}}(H)T_{f,\text{in},i}(t) + E_b(H)T_{b,i}(t)
\]

(11)

where \( E_{\text{out}}(z) = E_{12}(z) - E_{22}(z) \) and \( E_{\text{in}}(z) = E_{21}(z) - E_{11}(z) \) are coefficients obtained from the matrix exponential. \( E_b = [E_{b,v}] \) is a line vector of coefficients for each of the borehole segments, given by:

\[
E_{b,v}(z) = [1 \, -1]VD^{-1}\left(\exp\left(D(H - (v - 1) \cdot H/n_q)\right) - \exp\left(D(H - v \cdot H/n_q)\right)\right)V^{-1}A_{12}
\]

(12)

**Full Model**

The borehole model is coupled to the ground model through energy balances along each segment of each borehole. Fluid temperature variations along a borehole segment may be calculated from eq. (10). The heat extraction rate of the borehole segment is then equal to the net heat gain along all pipes:

\[
Q_{b,i}(t) = -\Delta E_{\text{in}}T_{f,\text{in},i}(t) + \Delta E_{\text{out}}T_{f,\text{out},i}(t) - \Delta E_bT_{b,i}(t)
\]

(13)

where \( \Delta E_{\text{in}} = [\Delta E_{\text{in},u}] \) and \( \Delta E_{\text{out}} = [\Delta E_{\text{out},u}] \) are column vectors of coefficients, with \( \Delta E_{\text{in},u} = \dot{m}c_f\left(E_{\text{in}}\left(\frac{u}{n_q} \cdot H\right) - E_{\text{in}}\left(\frac{(u-1)}{n_q} \cdot H\right)\right) \) and \( \Delta E_{\text{out},u} = \dot{m}c_f\left(E_{\text{out}}\left(\frac{u}{n_q} \cdot H\right) - E_{\text{out}}\left(\frac{(u-1)}{n_q} \cdot H\right)\right) \), and \( \Delta E_b = [\Delta E_{b,uv}] \) is a matrix of coefficients, with:

\[
\Delta E_{b,uv} = \begin{cases} \\
0 & \text{for } u < v \\
\dot{m}c_fE_{b,v}(uH/n_q) & \text{for } u = v \\
\dot{m}c_f\left(E_{b,v}(uH/n_q) - E_{b,v}((u - 1)H/n_q)\right) & \text{for } u > v
\end{cases}
\]

(14)

As the boreholes are connected in series, the outlet fluid temperature from borehole \( i \) is equal to the inlet temperature into the next borehole \( i + 1 \). Finally, the total heat extracted from the bore field is equal to the sum of the heat extracted from all boreholes:

\[
Q(t) = B_{\text{in}}T_{f,\text{in}}(t) + B_{\text{out}}T_{f,\text{out}}(t)
\]

(15)

where \( Q(t) = [0_{n_b-1 \times 1} \quad Q_{\text{tot}}(t)] \) is a column vector, with \( 0_{n_b-1 \times 1} \) a \( n_b - 1 \times 1 \) all-zeros column vector and \( Q_{\text{tot}} \) is the total heat extraction rate from the bore field, \( B_{\text{in}} = [0_{n_b-1 \times 1} \quad I_{n_b-1} \quad -\dot{m}c_f \quad 0_{1 \times n_b-1}] \) and \( B_{\text{out}} = [-I_{n_b-1} \quad 0_{n_b-1 \times 1} \quad \dot{m}c_f \quad 0_{1 \times n_b-1}] \) are coefficient matrices, with \( I_{n_b-1} \) the \( n_b - 1 \times n_b - 1 \) identity matrix, and \( T_{f,\text{in}} = [T_{f,\text{in},i}] \) and \( T_{f,\text{out}} = [T_{f,\text{out},i}] \) are column vectors of the inlet and outlet fluid temperatures into each borehole in the bore field.

Eqs. (1,4,11,13,15) can be assembled to obtain a system of linear equations in the Laplace domain, which gives the relation between the heat extraction rates and borehole wall temperatures of all borehole segments, and the inlet and outlet fluid temperatures of all boreholes.

**RESULTS**

The proposed model is used to study the effects of ambient air temperature variations on the predicted fluid temperatures. Three assumptions of decreasing complexity for the calculation of ground temperatures are compared: (1) time- and depth-varying ground temperature, as obtained from eq. (4), (2) time-varying uniform ground temperature \( T_g(t) = \bar{T}_g(t) \), as obtained from eq. (6), and (3) constant uniform temperature \( T_g(t) = T_{g,0} \).
A geothermal system was simulated using the three previously defined assumptions for ground temperature variations with different number of equidistant boreholes on a straight line ranging from \( n_b = 1 \) to \( 50 \), using the same total heat extraction rate and total borehole length in the bore field. All boreholes are positioned on a single line and connected in series. The total heat extraction rate, shown on fig. 2, is obtained from the simulation of a single family home (Kegel, et al. 2012). A constant heat pump coefficient of performance (COP) of 3 in both heating and cooling modes and a heat pump capacity of 10 kW were assumed to obtain the heat extraction rates from the building loads. The peak heat extraction and heat injection rates are 6.67 kW and 5.61 kW respectively. The total yearly thermal energy extracted from and injected into the ground are 19190 kWh and 2243 kWh, respectively. The ambient air temperatures are obtained from TMY2 weather data for Montreal, Canada. The calculated ground temperature profiles from eq. (4) are shown on fig. 2 for different times of the year. The simulation parameters are presented in Table 1.

The maximum absolute predicted dimensionless outlet fluid temperature difference between each of assumptions (2) and (3) and assumption (1) are obtained for fields of \( n_b = 1, 2, 3, 4, 5, 6, 7, 10, 15, 20, 25, 30, 35, 40 \) and 50 boreholes and for buried depths \( D = 2, 3 \) and \( 4 \) m. The dimensionless outlet fluid temperature difference is defined as:

\[
\Delta \theta(t) = \frac{T_{f, out}^{(i)}(t) - T_{f, out}^{(1)}(t)}{T_{g,0} - T_{f, out}^{(1)}(t)}
\]  

where \( T_{f, out}^{(i)}(t) \) is the outlet fluid temperature of the \( n_b \)-th borehole calculated using assumption \( (i) \) defined above. The maximum predicted dimensionless outlet fluid temperature differences are shown on fig. 3 as a function of the portion of the borehole located above the thermal penetration depth \( D_T = \sqrt{\alpha_s t_T} \), where \( t_T = \frac{\pi}{2} \) years is the characteristic thermal penetration time. It is shown that the maximum absolute difference in predicted outlet fluid temperatures calculated using assumptions (1) and (3) increase linearly with increasing portion of the borehole above the penetration depth. A 2% difference is obtained when approximately 8% of the borehole is above the penetration depth. The maximum absolute difference in predicted outlet fluid temperatures calculated using assumptions (1) and (2) are much lower: the maximum absolute difference is less than 1.5% for all studied cases.
difference also decreases after reaching a maximum value between 35% and 50% of the borehole above the thermal penetration depth.

### Table 1. Simulation Parameters

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
<th>Units</th>
<th>Value</th>
<th>Units</th>
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<tr>
<td>Simulation time step</td>
<td>1</td>
<td>h</td>
<td></td>
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<td>Maximum time</td>
<td>10</td>
<td>years</td>
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<td>Total borehole length</td>
<td>350</td>
<td>m</td>
<td>1150</td>
<td>ft</td>
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<td>Buried depth</td>
<td>2 to 4</td>
<td>m</td>
<td>6.5 to 13.1</td>
<td>ft</td>
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<tr>
<td>Borehole spacing</td>
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<td>16.4</td>
<td>ft</td>
</tr>
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<td>Borehole radius</td>
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<td>in.</td>
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<td>Pipe outer radius</td>
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<td>in.</td>
</tr>
<tr>
<td>Pipe inner radius</td>
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<td>in.</td>
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<td>Shank spacing</td>
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<td>in.</td>
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<td>Undisturbed ground temperature</td>
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<td>°C</td>
<td>43.34</td>
<td>°F</td>
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<td>Ground thermal conductivity</td>
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<td>W/m-K</td>
<td>17.33</td>
<td>BTU-in./h-ft-°F</td>
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<td>Ground thermal diffusivity</td>
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<td>m²/s</td>
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<td>Grout thermal conductivity</td>
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<td>W/m-K</td>
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<td>Pipe thermal conductivity</td>
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<td>BTU-in./h-ft-°F</td>
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<td>Fluid flow rate</td>
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</table>

**Figure 3** Maximum absolute dimensionless outlet fluid temperature difference relative to assumption (1) for each of assumption (2) (left) and assumption (3) (right)

### CONCLUSION

A simulation model for geothermal bore fields was presented. The finite line source solution was used to calculate borehole wall temperatures, while a steady-state analytical solution of the interaction between the fluid in the U-tube pipes and the borehole wall was used to calculate the outlet fluid temperature. Different assumptions were considered for the calculation of ground temperature profiles to account for ambient air temperature variations: (1)
time-varying non-uniform ground temperature, (2) time-varying uniform ground temperature and (3) constant uniform ground temperature. The model was adapted to simulate fields of geothermal boreholes connected in series.

The model was used to simulate a residential geothermal heat pump system with boreholes connected in series. Predicted fluid temperatures were obtained for bore fields with numbers of boreholes ranging from 1 to 50 boreholes and corresponding borehole lengths from 350 m down to 7 m. The borehole buried depth also varied from 2 m to 4 m. It is shown that detailed time- and depth-variation modeling of ground variations is not needed to accurately predict fluid temperatures. A time-varying uniform ground temperature profiles results in predicted fluid temperatures within 1.5 % of those obtained with time-varying non-uniform ground temperatures.

While it was shown that ambient temperature variations may significantly affect fluid temperatures in short boreholes, some limitations of the present model need to be addressed to correctly assess the efficiency of systems using short boreholes. For instance, for short boreholes, ground thermal properties variations along the depth of the boreholes might also have an important impact on fluid, borehole and near-surface ground temperatures.

**NOMENCLATURE**

\(\alpha\) = Thermal diffusivity \((m^2/s)\)

\(A\) = Coefficient matrix of the system of differential equations for fluid temperatures

\(B\) = Coefficient matrix for connections between boreholes

\(c\) = Specific heat \((J/kg-K)\)

\(D\) = Depth (m)

\(\Delta h\) = Matrices of eigenvalues and eigenvectors of \(A\)

\(\Delta h\) = Matrix of segment-to-segment response factors from the finite line source solution

\(E, \Delta E\) = Coefficients in the borehole model for equal temperatures at \(z = H\) and energy balances over segments

\(k\) = Thermal conductivity \((W/m-K)\)

\(H\) = Borehole length (m)

\(m\) = Flow rate \((kg/s)\)

\(n_b, n_q\) = Number of boreholes and borehole segments

\(Q\) = Heat extraction rate \((W)\)

\(R\) = Matrix of thermal resistances \((m-K/W)\)

\(T\) = Temperature \((^\circ C)\)

\(t\) = Time \((s)\)

**Subscripts**

\(air\) = air

\(b\) = borehole

\(f\) = fluid

\(g, s\) = ground

\(i, j\) = borehole indexes

\(in\) = inlet

\(out\) = outlet

\(T\) = thermal penetration

\(u, v\) = borehole segment indexes

\(0\) = initial
REFERENCES


Alternate Approach to the Calculation of Thermal Response Factors for Vertical Borehole Ground Heat Exchanger Arrays Using an Incomplete Bessel Function

Andrew D. Chiasson  Rodwan Elhashmi

ABSTRACT
This article presents yet another methodology for the calculation of dimensionless thermal response factors for vertical borehole ground heat exchanger (GHX) arrays, which is a concept introduced by Eskilson (1987). The presented method is based on a well-known solution to an analogous problem in the field of well hydraulics. This solution method, known mathematically as an incomplete Bessel function, and known in the field of well hydraulics as the ‘leaky aquifer function’, describes the hydraulic head distribution in an aquifer with predominantly radial flow to a well combined with vertical ‘leakage’ from geologic layers above and below the pumped aquifer. The solution is adapted to model heat transfer from an array of arbitrarily-placed vertical boreholes of finite depth. With proper expression of parameters in the incomplete Bessel function, we show that g-functions of previous researchers can be approximated. The proposed method has been implemented into Matlab and Excel/VBA for g-function generation and monthly GHX simulation.

INTRODUCTION
The concept of thermal response factors for application to ground heat exchanger (GHX) dimensioning was originally introduced by Eskilson (1987). Those thermal response factors, known as g-functions, were developed using a combination of numerical and analytical techniques with the boundary condition of uniform borehole wall temperature along the length of the boreholes. More recently, based on desire to improve the computational speed of vertical GHX simulation and optimization calculations, several researchers have developed analytical solutions to generate thermal response factors. Such analytical solutions are also more desirable in applications with variably-spaced boreholes, as with underground thermal energy storage systems, for example.

Thermal response factors developed subsequently to those of Eskilson (1987) have been developed under differing boundary conditions at the borehole wall, and the literature now appears unsettled as to the most appropriate boundary condition. In reality, boreholes in a typical vertical GHX design have a common inlet fluid temperature, and both the temperature and heat transfer rate will vary along the length of the boreholes. Cimmino and Bernier (2014) succinctly summarize the various types of boundary conditions possible in development of thermal response factors: (i) uniform heat extraction rate along the length of the boreholes such that heat extraction rates are equal for all boreholes and the average temperatures along the length of all boreholes are unequal, (ii) uniform heat extraction rate along the length of the boreholes such that the average temperature along the length of the boreholes is equal for all boreholes...
boreholes, and (iii) uniform borehole wall temperature along the length of the boreholes such that the borehole wall
temperature is equal for all boreholes (Eskilson’s method).

The purpose of the present research is to approach the development of dimensionless temperature response
factors from a different perspective by adapting a solution to an analogous problem in the field of well hydraulics.
Results are compared to those of other published solutions.

BACKGROUND AND LITERATURE REVIEW

Eskilson (1987) introduced the concept of \( g \)-functions for heat extraction (or rejection) boreholes. The \( g \)-
functions provide a relation between the heat extracted from the ground at the borehole wall \( q' \) and the borehole wall
temperature \( T_b \) as:

\[
T_b = T_g - \frac{q'}{2 \pi k} \text{sgn} \left( \frac{t}{t_s} \right) \frac{t}{H} + \frac{b}{B} \left( \frac{r_b}{r_t} \right) \text{sgn} \left( \frac{r_b}{r_t} \right) \tag{1}
\]

where \( t_s \) is a characteristic time scale given by \( H^2/(9 \alpha) \). As described by Cimmino et al. (2013), Eskilson’s approach
introduced a fourth non-dimensional parameter, \( D/H \), but that parameter was not explicitly varied in his studies, as he
used a fixed value of \( D \) equal to 4 or 5 m.

Eskilson’s \( g \)-functions have been thoroughly described in the literature (eg. Cimmino et al. (2013), Cimmino and
Bernier (2014), and Spitler and Bernier (2016)), and are often taken as a benchmark for comparison with other
thermal response factors. Cimmino et al. (2013) describe the numerous analytical solution methods attempting to
duplicate Eskilson’s work, but an exact duplication has been elusive due to the method of the spatial superpositioning
of single borehole solutions used by Eskilson. He generated thermal response factors numerically using two-
dimensional (radial and axial) numerical simulations combined with spatial superposition to effectively obtain a three-
dimensional response of the borehole heat exchanger field; the temperature at the borehole walls was uniform along
the length of the boreholes and equal for all boreholes.

Cimmino et al. (2013) proposed a new method to approximate \( g \)-functions. The method, based on the analytical
finite line source, accounts for the variation of heat extraction rates among boreholes due to thermal interaction
among boreholes and for the \( D/H \) ratio. The heat extraction rates obtained with this method showed good agreement
with Eskilson’s numerical model, but there are some differences that were explained by the fact that the methods used
different boundary conditions at the borehole wall. However, for small simulation times, the differences are small and
the response factors are almost identical for up to 10 and 6 years for \( 3 \times 2 \) and \( 10 \times 10 \) borehole fields, respectively. For
large borehole fields, however, thermal interactions among boreholes become important and the observed differences
between the Cimmino et al. (2013) model and Eskilson’s \( g \)-function increase. For example, the \( g \)-function of a \( 10 \times 10 \)
borehole field obtained with the Cimmino et al. (2013) method overestimates the \( g \)-function by 32% at steady-state.

Cimmino and Bernier (2014) were able to closely match Eskilson’s \( g \)-functions by dividing boreholes into
segments to consider the variation of the heat extraction rates along the length of the boreholes, and then applying the
analytical finite line source (FLS) solution to calculate the temperature variations at the wall of each borehole segment
along the axial direction. Their method accounted for the time variation of the heat extraction rates among boreholes
and along the length of individual boreholes to obtain a uniform borehole wall temperature equal for all boreholes in
accordance with the original boundary conditions proposed by Eskilson. The difference between the Cimmino and
Bernier (2014) model was within 5% of Eskilson’s \( g \)-functions for all studied bore fields, except for fields of boreholes
located on a single row.
PROPOSED MODEL

Conceptual Model

A conceptual diagram of a borehole heat exchanger is shown in Figure 1. The term borehole heat exchanger (BHE) refers to a closed-loop pipe assembly installed in a vertical borehole with radius \( r_b \) over some active depth \( H \) for purposes of heat exchange with the subsurface. The top of the active BHE is buried at some depth \( (D) \) from the ground surface, and typical constructions consist of a single U-tube grouted in a borehole as shown in Figure 1, but other geometries exist (i.e., double U-tube, concentric tube, and groundwater-filled boreholes). These boreholes are designed to extract (or reject) a certain amount of thermal energy \( (q) \) per unit depth \( (H) \) by pumping a fluid through the heat exchanger. Thus, \( q' \) is the heat extraction or rejection rate divided by the active borehole depth. Heat transfer occurs from the fluid to the ground whose undisturbed “far-field” temperature remains at \( T_g \). \( T_s \) represents the average temperature at the borehole wall.

![Conceptual diagram of a borehole heat exchanger.](image)

Coupled to a building or other time-varying thermal load, multiple (i.e., two to tens) BHEs are connected together in a collective array to form what we refer here to as a ground heat exchanger (GHX). A common GHX design configuration is a line or grid pattern of BHEs with a fixed borehole-to-borehole field spacing, \( B \). Over time, BHEs become thermally-interacting.

Single Borehole

**Governing Equations:** The simple case of one-dimensional, radial heat conduction is given by the following partial differential equation:

\[
\frac{\partial^2 T}{\partial r^2} + \frac{1}{r} \frac{\partial T}{\partial r} = \frac{1}{\alpha} \frac{dT}{dt}
\]

(2)

In an analogy to heat conduction, Equation 2 is used in the field of well hydraulics to describe radial flow toward a well. However, in many situations, flow toward a well consists of vertical flow components caused by ‘leakage’ from
overlying and/or underlying geologic layers. The partial differential equation describing such cases is given by:

$$\frac{\partial^2 T}{\partial r^2} + \frac{1}{r} \frac{\partial T}{\partial r} + \xi = \frac{1}{\alpha} \frac{dT}{dt}$$

(3)

where $\xi$ expresses the ratio of vertical to horizontal thermal (or hydraulic) transmissivities or $\Delta T (\alpha t)^{-1}$.

**Initial and Boundary Conditions:** The initial condition is prescribed as $T(r,0) = T_g$. The boundary conditions are given as $T(r,\infty) = T_\infty$ and $\lim_{r \to 0} \left( r \frac{\partial T}{\partial r} \right) = q'_r / 2\pi \alpha$.

**Solution and Implementation:** The solution to Equation (2) for the temperature change ($\Delta T$) in a heat-conducting medium is that around an infinite line source, and is well known as:

$$\Delta T = \frac{q'}{4\pi k} \int_u^\infty e^{-u} \frac{q'}{4\pi k} \cdot Ei(u) = \frac{q'}{4\pi k} \cdot W(u)$$

(4)

where $Ei$ is known mathematically as the exponential integral function, $W$ is known in the field of *well hydraulics* as the *Theis well function* or more simply as the *well function*, and $u = r^2 / (\alpha t)$.

The solution to Equation 3, which models the temperature distribution around a finite line source, is:

$$\Delta T = \frac{q'}{4\pi k} \int_u^\infty e^{-u} \left( -u - \frac{r^2}{4\beta u} \right) du = \frac{q'}{4\pi k} \cdot W \left( u, \frac{r}{\beta} \right)$$

(5)

where $W(u, r/\beta)$ is known mathematically as an Incomplete Bessel Function, and in the field of *well hydraulics* as the *leaky well function* or the *Hantush-Jacob well function* after Hantush and Jacob (1955), and $\beta$ is a ‘leakage factor’. Further mathematical details of these functions are described by Harris (2008).

In an analogy to heat conduction, we define $\beta$ as $(\alpha t)^{1/2}$, where $t_c$ is the characteristic time scale defined by Eskilson (1987) as $H^2 / (9 \alpha)$. Thus, Equation 5 can be adapted to model the temperature distribution around a finite line source with length $H$ and burial depth $D$ as:

$$\Delta T = \frac{q'}{4\pi k} \cdot W(u, b)$$

(6)

where we define:

$$b = 3 \frac{r}{H} \left( 1 + \frac{D}{H} \right)^{-1}$$

(7)

The formulation for $g$ is now expressed with the leaky well function notation as:

$$g \left( \frac{t}{t_c}, \frac{r}{H}, \frac{B}{H}, \frac{D}{H} \right) = \frac{1}{2} W(u, b)$$

(8)
Multi-Borehole Arrays

The inherent three-dimensional temperature distribution in the ground is modeled by superpositioning of the single-borehole solution in two different ways corresponding to boundary condition (i) and boundary condition (iii) described above. That is: (i) uniform heat extraction rate along the length of the boreholes such that heat extraction rates are equal for all boreholes and the average temperatures along the length of all boreholes are unequal, and (iii) uniform borehole wall temperature along the length of the boreholes such that the borehole wall temperature is equal for all boreholes (Eskilson’s method).

By spatial superpositioning of finite line sources calculated with the proposed method, the response factors are determined for any arbitrary borehole field pattern and not necessarily limited to the prescribed geometries of Eskilson. Spatial superpositioning involves calculation of the dimensionless temperature response \( g \) in each borehole due to all the others at a particular time. Thus, the dimensionless temperature response of a particular borehole \( g_b \) is the sum of the influence from all other boreholes:

\[
g_{b_{x,y}} = \sum_{i=1}^{x_{bore}} \sum_{j=1}^{y_{bore}} g_{i,j} \tag{9}
\]

where \( i \) and \( j \) are indices, and \( x_{bore} \) and \( y_{bore} \) are the number of boreholes in the \( x \) and \( y \) direction, respectively. The \( g \)-function can be calculated for an entire borehole field at a particular time dividing the result of Equation 9 by the number of boreholes in the field.

Development of \( g \)-functions with the proposed method using boundary condition (i) is straightforward: an arbitrary constant heat rate was specified per borehole, and the temperature change at the borehole walls were calculated directly and then non-dimensionalized. Development of \( g \)-functions with the proposed method using boundary condition (iii) is less straightforward. Here, we specified a temperature change at each borehole and applied a multivariable optimization method to iteratively adjust the heat rates per length per borehole such that the borehole wall temperature remained unchanged. Calculation times using boundary condition (iii) were orders of magnitude greater than calculation times using boundary condition (i).

RESULTS AND DISCUSSION

To the best of our knowledge, Eskilson (1987) used a total borehole depth \((D+H)\) of 115 m in development of his \( g \)-functions. The active borehole length \((H)\) was 110 m and the buried depth \((D)\) was 5 m. In comparison to Eskilson’s work in what follows, we used a \( D/H \) ratio equal to 0.0454 and an \( r_b/H \) ratio of 0.0005.

Comparison to Eskilson’s \( g \)-functions – The Case of Constant Uniform Heat Rate Along the Borehole Lengths

\( g \)-functions determined with the proposed method using the boundary condition of uniform heat extraction rate along the length of the boreholes are compared to those of Eskilson (1987) in Figure 2. \( g \)-functions were generated for a variety of borehole field configurations (L-shaped, U-shaped, linear, open-square, rectangular, and square).

Observations of the comparisons of the \( g \)-functions shown in Figure 2 are consistent with those of other researchers (eg. Cimmino et al. (2013) and Malayappan and Spitler (2012)). That is, analytical solutions using a constant heat flux boundary condition at the borehole wall are essentially identical to Eskilson’s \( g \)-function for a single borehole, but tend to over-predict Eskilson’s \( g \)-functions for borehole fields with increasingly lower \( B/H \) ratio and increasingly greater number of boreholes. Here, we also observe that the proposed solution with increasing \( B/H \) ratio eventually begins to under-predict Eskilson’s \( g \)-functions. We further observe some differences in the \( g \)-functions obtained with differing boundary conditions as a function of \( \ln(t/t_0) \). These deviations imply different handling of end effects of the boreholes, which previous researchers attributed to the differing boundary conditions at the borehole wall.
Figure 2. Comparison of g-functions calculated by proposed model to those of Eskilson (1987) for various borehole field configurations using a boundary condition of uniform heat extraction rate along the length of the boreholes. Note $B/H = \infty$ is the g-function for a single borehole.
Comparison to Eskilson’s g-functions – The Case of Constant Borehole Wall Temperature

$g$-functions determined with the proposed method using the boundary condition of constant borehole wall temperature are compared to those of Eskilson (1987) in Figure 3. These $g$-functions were generated for comparison only for densely-packed borehole field configurations (i.e., $B/H$ ratio of 0.05). Here, we observe a much closer match between the proposed method and Eskilson’s $g$-functions for densely-packed borehole fields with low $B/H$ ratio. The steady-state values of the $g$-function determined by the proposed method are within 4% of Eskilson’s values.

![Figure 3](image)

**Figure 3.** Comparison of $g$-functions calculated by proposed model to those of Eskilson (1987) for various borehole field configurations using a boundary condition of constant temperature along the length of the boreholes. Note $B/H = \infty$ is the $g$-function for a single borehole.

Comparison to Other Solution Methods

$g$-functions generated with the proposed method using the boundary condition of constant heat flux at the borehole wall are compared to other solution methods graphically in Figure 4. These solution methods, described by Cimmino et al. (2013), are: (a) the analytical finite line source method and (b) the method developed by Cimmino et al. (2013). The former uses boundary condition (i) described above, while the latter uses boundary condition (ii) described above.

![Figure 4](image)

**Figure 4.** Comparison of $g$-functions calculated by proposed model to those presented by Cimmino et al. (2013) for boreholes in a (a) 3x2 rectangular pattern, (b) 6x4 rectangular pattern, and (c) 10x10 square pattern.

A review of Figure 4 reveals that the proposed solution matches the finite line source model up to a time of $\ln(t/t_s) = 0$, after which time the proposed solution more closely agrees with the method developed by Cimmino et al. (2013).
Discussion of Practical GHX Design Sizing

Here, we provide a comparison of borehole depth calculations for a reference building, where the GHX is designed using (i) the proposed method for the case of constant uniform heat rate along the borehole lengths, and (ii) the g-functions of Eskilson (1987). Loads for a reference medium-sized office building in Boulder, CO (U.S. Department of Energy Climate Zone 5B) were taken from datasets described by Field et al. (2010). The peak heating and cooling loads are 239 kW and 350 kW, respectively, and the annual cooling-to-heating loads ratio is about 1.6.

While GHX designs have many specific considerations, some generalizations are noted and made use of here. In most U.S. GHX designs, $H \approx 75$ m and $B \approx 6$ m, which results in $B/H \approx 0.08$. Further, given the general relationship between thermal conductivity to density and heat capacity, for common geologic materials, $\alpha \approx 1 \times 10^{-6}$ m$^2$/s. The foregoing generalities result in $\ln(t/t_s) \approx 0$ for a 20-year GHX design time scale.

In sizing a 10x10 borehole GHX with $B/H = 0.1$ for the reference medium-sized office building in Boulder, CO using the foregoing design assumptions, we find a difference of $\approx 2\%$ between the design length calculated with g-functions derived from the proposed PHWKRG $g$-functions. For an unrealistically close borehole spacing of $B/H = 0.05$, the difference between the two methods is $\approx 8\%$, which is significantly less than might be implied by observation of Figure 2(h). The main reasons for the similarity in the GHX design length determined by both methods is due to the inherent design considerations and building loads profile. Choice of $B$ and $H$ affect time scales of the problem, while the building loads dictate the short- and long-term ground thermal response. Thus, in a building with perfectly balanced annual loads, the long-term ground thermal response would be negligible, and both methods would yield nearly identical results.

CONCLUSION

This article has introduced yet another methodology for the calculation of dimensionless thermal response factors for vertical GHXs. The method, based on a well-known solution to an analogous problem in the field of well hydraulics, is known mathematically as an incomplete Bessel function, and known in the field of well hydraulics as the ‘leaky aquifer function’. The method has been implemented into Matlab and Excel/VBA for computationally-efficient $g$-function generation and subsequent GHX design sizing and/or simulation of thermally-interacting BHEs.

$g$-functions generated by the proposed method were compared to those of Eskilson (1987) for several borehole field configurations (L-shaped, U-shaped, linear, open-square, rectangular, and square) under two boundary conditions at the borehole wall (constant heat flux and constant temperature). As concluded by previous researchers, $g$-functions developed under the varying borehole wall boundary conditions have some similarities and some important differences. In reality, boreholes in a vertical borehole field have a common inlet fluid temperature, and both the temperature and heat transfer rate will vary along the length of the boreholes. Future work consisting of experimental validation of the various boundary condition assumptions would be quite helpful.

NOMENCLATURE

- $\alpha =$ Thermal diffusivity (m$^2$/s)
- $D =$ Depth of borehole below ground surface (m)
- $Ei =$ The exponential integral
- $H =$ Active borehole length (m)
- $k =$ thermal conductivity of underground materials (W/m-K)
- $q^\prime =$ heat rate per unit length (W/m)
- $r =$ Radial distance from the line source (m)
- $t =$ time (s)
- $T =$ Temperature ($^\circ$C)
- $W(\cdot) =$ Well function

Subscripts

- $b =$ borehole, $g =$ ground, $s =$ surface, characteristic time-scale
REFERENCES


A Model for Ground Temperature Estimations and Its Impact on Horizontal Ground Heat Exchanger Design

Lu Xing     Jeffrey D. Spitler     Liheng Li     Pingfang Hu

ABSTRACT

The ground-source heat pump systems are highly efficient and energy saving. Its main disadvantage is a significantly higher installation cost compared to conventional systems. The length of the ground heat exchanger (GHX) piping, consequently, the first cost, depends on several factors; one key factor is the undisturbed ground temperature estimations. Xing and Spitler model was developed which provides a new set of ground temperature results for GHX design. There are two common methods in United States to be used for ground temperature estimations - ASHRAE Handbook method and ASHRAE district heating manual method. This paper presents the impact of Xing and Spitler model development on the horizontal ground heat exchanger (HGHX) design. An analytical HGHX simulation tool is developed. 12 geographically diverse sites in United States are chosen for the case study. Three different HGHX configurations are investigated. For each site, HGHX design length using the Xing and Spitler model estimated ground temperatures as inputs are compared to design results based on measured ground temperatures; the calculated HGHX design length percentage error are within $\pm 18.9\%$. The calculated HGHX design length percentage error using the ASHRAE Handbooks results and ASHRAE district heating manual results are within $\pm 38.3\%$ and $\pm 57.7\%$ respectively.

INTRODUCTION

With the development of geothermal energy applications, the ground-source heat pump (GSHP) systems are used frequently in commercial, residential and industrial buildings as a type of sustainable heating and cooling systems. At the end of 2014, the installed geothermal heat pump power has been up to 50.2GW across 4.19 million units in buildings in a worldwide range. The annual energy use is 326,848 TJ/year, the energy savings for geothermal energy application equal 29.1 ton of equivalent oil (Lund and Boyd, 2016).

The GSHP system’s ground heat exchangers are usually placed in vertical boreholes or horizontal trenches. The drilling cost of the vertical boreholes is high and that is a barrier to system implementation. The excavation fees for horizontal ground heat exchangers (HGHX) installations are relatively lower. To lessen system initial cost, horizontal
piping is suggested for buildings located in large land areas and without ground space limitations. What’s more, it is required to develop an accurate procedure for sizing such systems. A system that is undersized may lead to equipment failure, while an oversized system is often inefficient and unnecessarily expensive.

Many models have been developed to simulate the GSHP systems or to calculate the required HGHX lengths. These models can be mainly classified as: numerical models and analytical models. Metz (1983) developed a 2-D numerical model to solve the underground heat flow of a buried tank. Mei and Emerson (1985) created a model to simulate double pipes, where the soil moisture freezing around a single pipe and interferences between pipes are considered. Piechowski (1996; 1999) presented a 3-D model which calculates the heat conduction problem of multiple pipes, and where moisture transport is considered. He assumed no thermal interference between pipes. Demir et al. (2009) developed a 2-D model considering the effect of snow cover rather than moisture transportation.

Numerical models consider several factors that affect the HGHX performance, they give accurate solutions and are good for theoretical analysis, but need extensive computational time (Florides, et al. 2013). The analytical method is commonly used for designing the HGHX. Ingersoll and Plass (1948) obtained the temperature field around an infinitely long line heat source/sink in an infinite soil domain. Hart and Couvillion (1986) obtained a time-dependent temperature distribution around multiple pipes by superimposing single line source. Persson and Claesson (2005) calculated the temperature distribution of multiple pipes buried in a semi-infinite soil domain, by using the multipoles method combined with the line source approach. Saastamoinen (2007) solves the unsteady state temperature fields due to several constant line sources in ground, by using integral transform method.

In order to design the HGHX using these analytical methods, knowledge of the undisturbed temperatures is required. Accurately determined values of undisturbed ground temperatures, at the depths and time of occurrence, are significantly beneficial for proper sizing of ground heat exchangers and ground source heat pump system as a whole (Kurevija, et al. 2011). This study mainly discusses the application of a simplified model developed by Xing and Spitler (Xing and Spitler 2016a, Xing and Spitler 2016b, Xing et al. 2016) for ground temperature estimations. The calculated ground temperatures are used as input data to the HGHX design model so as to study the simplified model impact on the HGHX piping design.

**METHODOLOGIES**

An analytical model for simulating GSHP systems using HGHX has been developed. A typical residential building is built in twelve locations in United States. Building heating and cooling loads, ground temperatures and ground thermal properties are used as inputs to the analytical HGHX model. For each site, the required HGHX lengths are calculated using four ground temperatures results: measured data, Xing and Spitler model results, ASHRAE Handbooks method results and ASHRAE district heating manual method results. The designed HGHX lengths using the three estimated ground temperature results are compared to the designed lengths using measured ground temperatures. The HGHX design length percentage error are summarized and analyzed so as to observe the relationship of ground temperature estimation errors and HGHX design length percentage error.

**Simulation of Horizontal Ground Heat Exchangers**

The HGHX simulation tool is developed based upon the foundation heat exchanger (FHX) simulation tool
(Xing et al. 2012). The FHX simulation tool uses analytical method modeling FHX pipes buried in a semi-infinite soil domain and are connected to indoor heat pumps for building heating and cooling purposes. Each FHX pipe is treated as a line source or sink, multiple ones are simulated based on superposition of a single one. The analytical model assumes that, in the soil, conduction heat transfer is important; moisture transport and freezing effects are neglected. It assumes that the effect of changing weather conditions can be accounted with inputs of undisturbed ground temperatures. With other inputs such as FHX configuration and properties, soil properties, heat pump performance parameters, etc., the simulation tool calculates the FHX pipe lengths required for the house. The simulation tool has been validated against one year hourly time step experimental data collected by Oak Ridge National Lab at a house in Oak Ridge, Tennessee (Xing et al. 2012).

The foundation heat exchanger is a relatively new type of ground heat exchanger that utilizes the excavation often made for basements and foundation in order to reduce the high cost of trench excavation. HGHX is similar to FHX in geometry, without the presence of a basement in close proximity to the heat exchanger tubing. Therefore, the foundation assumed in the foundation heat exchanger simulation tool is removed in order to simulate the horizontal ground heat exchangers.

### Heating and Cooling Loads for Prototype Houses

The HGHX simulation model requires monthly average and peak building loads as inputs. This study involved developing hourly building loads for a prototype house located at twelve different sites in United States. The monthly time step simulation model simplifies the hourly loads, which are treated as monthly constant loads applied over the whole month and monthly peak loads applied at the end of the month (Cullin and Spitler 2011).

**House Description:** The prototype house used in this study is a single-family residence and is modeled in the EnergyPlus Environment (Crawley et al. 2001). It has a floor area of 148m² (1590ft²) and an aspect ratio of 1.56. The house is maintained at set points of 24.5°C (76°F) in cooling and 21.7°C (71°F) in heating.

### Table 1. Twelve Parametric Study Sites

<table>
<thead>
<tr>
<th>States in the U.S</th>
<th>Sites Name (SCAN)</th>
<th>Köppen-Geiger climate classification</th>
<th>TMY weather files</th>
</tr>
</thead>
<tbody>
<tr>
<td>Arizona</td>
<td>Walnut Gulch</td>
<td>Bsk</td>
<td>Douglas-Bisbee Douglas International Airport</td>
</tr>
<tr>
<td>Colorado</td>
<td>Nunn</td>
<td>Bsk</td>
<td>Greeley-Weld County AWOS</td>
</tr>
<tr>
<td>New Mexico</td>
<td>Los Lunas PMC</td>
<td>Bsk</td>
<td>Albuquerque International Airport</td>
</tr>
<tr>
<td>Oregon</td>
<td>Lyn Hart Ranch</td>
<td>Csb</td>
<td>Klamath Falls International Airport</td>
</tr>
<tr>
<td>Alabama</td>
<td>WTARS</td>
<td>Cfa</td>
<td>Huntsville International Airport–Jones Field</td>
</tr>
<tr>
<td>Arkansas</td>
<td>UAPB Lonoke Farm</td>
<td>Cfa</td>
<td>Little Rock-Adams Field</td>
</tr>
<tr>
<td>Georgia</td>
<td>Watkinsville</td>
<td>Cfa</td>
<td>Athens-Ben Epps Airport</td>
</tr>
<tr>
<td>Kentucky</td>
<td>Mammoth Cave</td>
<td>Cfa</td>
<td>Bowling Green-Warren County</td>
</tr>
<tr>
<td>Maryland</td>
<td>Powder Mill</td>
<td>Cfa</td>
<td>Baltimore-Washington International Airport</td>
</tr>
<tr>
<td>Oklahoma</td>
<td>Fort Reno</td>
<td>Cfa</td>
<td>Oklahoma City-Will Rogers World Airport</td>
</tr>
<tr>
<td>South Carolina</td>
<td>Pee Dee</td>
<td>Cfa</td>
<td>Florence Regional Airport</td>
</tr>
<tr>
<td>Virginia</td>
<td>Tide Water AREC</td>
<td>Cfa</td>
<td>Franklin Municipal Airport</td>
</tr>
</tbody>
</table>

**Locations:** Twelve sites are chosen over a range of weather conditions. Site names and states for their location are presented in Table 1. These sites are classified to different climates based on the Köppen-Geiger climate classification system (Kottek et al. 2006). Measured Typical Meteorological Year (TMY) weather data are available.
for the sites. These are used as inputs to the Energy Plus house model.

**Ground Temperatures, Soil Properties and Others**

Ground temperatures are required as inputs to the HGHX simulation tool. Four sets of ground temperatures are used for calculating HGHX lengths; these are: measured ground temperatures, Xing and Spitler model results and two commonly used approaches calculation results.

**Measurements:** The Soil Climate Analysis Network (SCAN) provides ground temperature data at the twelve sites at four depths: 5cm, 20cm, 50cm and 100cm (2in, 8in, 20in and 40in) for 3-8 years inside United States (NRCS 2013). For each site, these results are averaged and compiled into a typical year ground temperature file. At these sites, ground temperature measurements are available to the 100cm (40in) depth. Ground temperatures varies at different depths and time of year, and they are not linearly related. To obtain the “measured” ground temperatures at the HGHX burial depths during certain time frame, the measured ground temperatures are represented into a two-harmonic model (Xing 2014). Lord Kelvin (Thomson 1862) presented a higher order harmonic model. At the order of two, it becomes a two-harmonic model with five parameters - annual average ground temperature, two annual temperature amplitude at the ground surface and the two phase lag. These measured results are the “best” ground temperatures can be achieved using a two-harmonic relationship even if the measurement data are available.

**Xing and Spitler Model:** Measured ground temperatures are limited, modeling can be a useful tool. Analytical model in an equation form requires much less computational time and is convenient for engineering applications. A simplified analytical model (Xing and Spitler 2016a, Xing and Spitler 2016b, Xing et al. 2016) for undisturbed ground temperature estimation has been developed. The model relies on five weather-related constants - annual average undisturbed ground temperature, two annual amplitudes of surface temperature variations and two phase angles to predict the ground temperatures. Automatic procedures have been developed for generating these constant values for 4112 sites or more in a world-wide range. The procedures have been validated using 3-8 years of measured results at nineteen SCAN sites in United States (Xing and Spitler 2016a, Xing and Spitler 2016b, Xing et al. 2016).

**ASHRAE Handbook Method:** In the U.S., a commonly used approach is the Fourier one-harmonic model (Narasimhan 2010). The model relies on three parameters - annual average ground temperature, annual temperature amplitude at the ground surface and the phase lag to estimate ground temperatures. The annual average ground temperature and annual temperature amplitude at the ground surface can be read from very small maps for the continental US as Figure 17 of Chapter 34 of the ASHRAE Handbook - HVAC Applications (2011) or North America as Figure 13 of Chapter 18 of the ASHRAE Handbook - Fundamentals (2013a). These maps can be traced back to research in the 1920s (Collins 1925) and 1950s (Chang 1958).

**ASHRAE District Heating Manual Method:** ASHRAE published a district heating guide (ASHRAE 2013b) which also uses one-harmonic model to estimate the undisturbed ground temperatures. The method is developed based on the assumption that the average monthly ground surface temperature equals the average monthly air temperature. This is done for all 5564 weather stations (U.S. and international) listed in Chapter 14 of 2009 ASHRAE Handbook - Fundamentals. These constants are publicly available (ASHRAE, 2013c).

For this study, typical ground thermal properties is assumed. The soil is 60% saturated clay loam, with thermal
conductivity of 1.08 W/m·K (0.624 Btu/ft·°F·hr) and volumetric heat capacity of 2.479 MJ/m³·K (36.96 Btu/ft³·°F). 3/4 inch diameter HDPE piping is used; two HGHXs are buried in a single trench at 1.5 m (4.9 ft) depths with a distance of 0.6 m (2.0 ft). The fluid flowing in the tubes is water mixed with 10% propylene glycol.

**Component Sizing**

The HGHX simulation tool allows users to perform HGHX simulation to determine the monthly average and peak fluid temperatures entering the heat pump using building heating and cooling loads, ground temperatures, soil properties and others as inputs. By changing the length of the HGHX pipes, the user can limit the fluid temperature to within heat pump constraints. The constraining temperature for the water to air heat pump is set to be minimum entering fluid temperature (EFT) of 0°C (32°F) and a maximum EFT of 35°C (95°F).

**RESULTS ANALYSIS**

For the twelve sites listed in Table 1, the required HGHX lengths are calculated and presented in Table 2. These sites are located in two climates: 4 sites are in arid or dry-summer climates and 8 sites are in warm climates. The HGHXs lengths are calculated using the four sets of ground temperatures previously described.

<table>
<thead>
<tr>
<th>Climate zone</th>
<th>States in the U.S.</th>
<th>SCAN site name</th>
<th>Köppen-Geiger climate type</th>
<th>Measured results</th>
<th>Xing and Spitler model</th>
<th>ASHRAE Handbooks method</th>
<th>ASHRAE district heating manual method</th>
</tr>
</thead>
<tbody>
<tr>
<td>Arid or dry summer climates</td>
<td>Arizona</td>
<td>Walnut Gulch</td>
<td>Bsk</td>
<td>87.9 (288.4)</td>
<td>82.6 (271.0)</td>
<td>82.9 (272.0)</td>
<td>64.8 (212.6)</td>
</tr>
<tr>
<td></td>
<td>Colorado</td>
<td>Nunn</td>
<td>Bsk</td>
<td>203.4 (667.3)</td>
<td>193.4 (643.5)</td>
<td>198.4 (650.9)</td>
<td>293 (961.3)</td>
</tr>
<tr>
<td></td>
<td>New Mexico</td>
<td>Los Lunas PMC</td>
<td>Bsk</td>
<td>65.9 (216.2)</td>
<td>63.1 (207.0)</td>
<td>56.4 (185.0)</td>
<td>77.1 (253.0)</td>
</tr>
<tr>
<td></td>
<td>Oregon</td>
<td>Lyn Hart Ranch</td>
<td>Csb</td>
<td>148.4 (486.9)</td>
<td>125.7 (412.4)</td>
<td>96.5 (316.6)</td>
<td>194.1 (636.8)</td>
</tr>
<tr>
<td>Warm climates</td>
<td>Alabama</td>
<td>WTARS</td>
<td>Cfa</td>
<td>79.2 (259.8)</td>
<td>83.2 (273.0)</td>
<td>82 (269.0)</td>
<td>84.7 (277.9)</td>
</tr>
<tr>
<td></td>
<td>Arkansas</td>
<td>UAPB Lonoke Farm</td>
<td>Cfa</td>
<td>77.1 (253.0)</td>
<td>73.3 (240.5)</td>
<td>73.3 (240.5)</td>
<td>70.1 (230.0)</td>
</tr>
<tr>
<td></td>
<td>Georgia</td>
<td>Watkinsville</td>
<td>Cfa</td>
<td>61.1 (200.5)</td>
<td>61.3 (201.1)</td>
<td>62.5 (205.1)</td>
<td>61.4 (201.4)</td>
</tr>
<tr>
<td></td>
<td>Kentucky</td>
<td>Mammoth Cave</td>
<td>Cfa</td>
<td>113.1 (371.1)</td>
<td>113 (370.7)</td>
<td>110.1 (361.2)</td>
<td>125.3 (411.1)</td>
</tr>
<tr>
<td></td>
<td>Maryland</td>
<td>Powder Mill</td>
<td>Cfa</td>
<td>103 (337.9)</td>
<td>99.2 (325.5)</td>
<td>88.5 (290.4)</td>
<td>110.8 (363.5)</td>
</tr>
<tr>
<td></td>
<td>Oklahoma</td>
<td>Fort Reno</td>
<td>Cfa</td>
<td>87 (285.4)</td>
<td>79.4 (260.5)</td>
<td>76 (249.3)</td>
<td>84.6 (277.6)</td>
</tr>
<tr>
<td></td>
<td>South Carolina</td>
<td>Pee Dee</td>
<td>Cfa</td>
<td>79.6 (261.2)</td>
<td>75.6 (248.0)</td>
<td>69.2 (227.0)</td>
<td>68.2 (223.8)</td>
</tr>
<tr>
<td></td>
<td>Virginia</td>
<td>Tide Water AREC</td>
<td>Cfa</td>
<td>65.1 (213.6)</td>
<td>63.2 (207.3)</td>
<td>59 (193.6)</td>
<td></td>
</tr>
</tbody>
</table>

For each site, the “Xing and Spitler model” result is compared to the “Measured results”; HGHX design length percentage error using Xing and Spitler model estimated ground temperatures is calculated and presented in Table 3. The HGHX design length percentage errors are within the range of ±15.3%. Table 3 also shows the HGHX design length percentage error using ASHRAE Handbooks method and ASHRAE district heating manual method, which are within the range of ±35.0% and ±44.1% respectively. It is found out that, using ASHRAE Handbooks methods estimated ground temperatures to design HGHX lengths, in most cases, the error introduced leads to designs with shorter boreholes compared to the reference. In other words, this methods will cause undersizing of HGHX pipes. The ASHRAE District heating manual method gives designed results with sometimes shorter and other times longer boreholes compared to the reference.
<table>
<thead>
<tr>
<th>Climate zone</th>
<th>States in the U.S.</th>
<th>SCAN site name</th>
<th>Köppen-Geiger climate type</th>
<th>Xing and Spitler model</th>
<th>ASHRAE Handbooks method</th>
<th>ASHRAE District heating manual method</th>
</tr>
</thead>
<tbody>
<tr>
<td>Arid or dry summer</td>
<td>Arizona</td>
<td>Walnut Gulch</td>
<td>Bsk</td>
<td>-6.0</td>
<td>-5.7</td>
<td>-26.3</td>
</tr>
<tr>
<td></td>
<td>Colorado</td>
<td>Nunn</td>
<td>Bsk</td>
<td>-4.9</td>
<td>-2.5</td>
<td>44.1</td>
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<tr>
<td></td>
<td>New Mexico</td>
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<td>Bsk</td>
<td>-4.2</td>
<td>-14.4</td>
<td>17.0</td>
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<td></td>
<td>Oregon</td>
<td>Lyn Hart Ranch</td>
<td>Csb</td>
<td>-15.3</td>
<td>-35.0</td>
<td>30.8</td>
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<td></td>
<td>Alabama</td>
<td>WTARS</td>
<td>Cfa</td>
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<td>3.5</td>
<td>6.9</td>
</tr>
<tr>
<td></td>
<td>Arkansas</td>
<td>UAPB Lonoke Farm</td>
<td>Cfa</td>
<td>-4.9</td>
<td>-4.9</td>
<td>-9.1</td>
</tr>
<tr>
<td></td>
<td>Georgia</td>
<td>Watkinsville</td>
<td>Cfa</td>
<td>0.3</td>
<td>2.3</td>
<td>0.5</td>
</tr>
<tr>
<td>Warm climates</td>
<td>Kentucky</td>
<td>Mammoth Cave</td>
<td>Cfa</td>
<td>-0.1</td>
<td>-2.7</td>
<td>10.8</td>
</tr>
<tr>
<td></td>
<td>Maryland</td>
<td>Powder Mill</td>
<td>Cfa</td>
<td>-3.7</td>
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<td></td>
<td>Oklahoma</td>
<td>Fort Reno</td>
<td>Cfa</td>
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</tr>
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<td></td>
<td>South Carolina</td>
<td>Pee Dee</td>
<td>Cfa</td>
<td>-5.0</td>
<td>-13.1</td>
<td>-14.3</td>
</tr>
<tr>
<td></td>
<td>Virginia</td>
<td>Tide Water AREC</td>
<td>Cfa</td>
<td>-2.9</td>
<td>-9.4</td>
<td></td>
</tr>
</tbody>
</table>

### Ground Temperature Estimations and HGHX Design length percentage error

It is found from Table 3, for the twelve sites, that the “Xing and Spitler model” result gives an average of 4.9% and 10.4% lower HGHX design length percentage error than the two common methods. At the two sites in Oregon and Colorado, the HGHX design length percentage error using Xing and Spitler model estimated ground temperatures is 19.7% and 39.2% lower than the two common used methods respectively. Why is that?

The HGHXs are designed by constraining the heat pump entering fluid temperatures (EFTs) within a range of 0°C - 35°C (32°F - 95°F). The heat pump EFTs are closely related to peak (annual maximum/minimum) ground temperatures. Therefore, the HGHX design length percentage error is supposed to be correlated to the peak ground temperature estimation error. Figures 1 plots the peak ground temperatures estimation errors versus corresponding HGHX design length percentage error, for the twelve sites. It is observed that the HGHX design length percentage error and peak ground temperature estimation errors are almost linearly correlated.

The Xing and Spitler model gives much lower peak ground temperature estimation errors, less than 1.6°C (2.9°F), circled in a square box in Figure 1. The corresponding HGHX design length percentage error are less than 15.3%. The ASHRAE Handbooks method and ASHRAE district heating manual method give higher peak ground temperature estimation errors 2.7°C (4.9°F) and 4.7°C (8.5°F). These lead to the higher HGHX design length percentage error, 35.0% and 44.1% respectively.

![Figure 1](image.png)
At two sites Oregon and Colorado (circled in a round box in Figure 1), data points deviate from the fitting line. These two sites are located in cold climates, where the HGHX lengths required for the system are relatively longer: 148.4m and 203.4m (486.9ft and 667.3ft). As the HGHX design length increases, the HGHX design length percentage error correspond faster to the increase of the ground temperature estimation error. This suggests that, in colder climates, the accuracy of predicting the ground temperatures is more influential for designing of HGHX.

**Pipe Configurations and HGHX Design length percentage error**

One typical type of HGHX configurations (Figure 2a) has been investigated and it is concluded that the ground temperature estimations error have an important effect on HGHX design accuracy. For the twelve parametric study sites, the Xing and Spiterl model gives lower ground temperature estimation errors than the two commonly used approaches do. Using the Xing and Spiterl model estimated ground temperatures as inputs to HGHX simulation model, the HGHX design length percentage error are greatly reduced. Would similar conclusions be drawn for different HGHX configurations?

![Figure 2](image)

Figure 2 presents three types of HGHX configurations. These HGHXs are buried at: (a) 1.5m (4.9ft) depth, 0.6m (2.0ft) horizontal distance between two pipes (b) 0.9m (3.0ft) and 1.5m (4.9ft) depths (c) 0.9m (3.0ft) and 1.5m (4.9ft) depths, 0.6m (2.0ft) horizontal distance between two pipes. Table 4 shows the HGHX design length percentage error using the Xing and Spiterl model results for these three configurations shown in Figure 2.

<table>
<thead>
<tr>
<th>Climate zone</th>
<th>States in the U.S.</th>
<th>SCAN site name</th>
<th>Köppen-Geiger climate type</th>
<th>Two pipes in one layer (Figure 2a)</th>
<th>Two pipes in two layers (Figure 2b)</th>
<th>Four pipes in two layers (Figure 2c)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Arid or dry summer</td>
<td>Arizona</td>
<td>Walnut Gulch</td>
<td>BSk</td>
<td>-6.5</td>
<td>-6.0</td>
<td>-6.5</td>
</tr>
<tr>
<td></td>
<td>Colorado</td>
<td>Nunn</td>
<td>BSk</td>
<td>-3.6</td>
<td>-4.9</td>
<td>-3.7</td>
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<tr>
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<td>New Mexico</td>
<td>Los Lunas PMC</td>
<td>BSk</td>
<td>-9.2</td>
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<td>-9.1</td>
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<tr>
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<td>Oregon</td>
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<td>Csb</td>
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<td>-18.9</td>
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<td>Warm climates</td>
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<td>WTARS</td>
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<td>5.1</td>
<td>6.0</td>
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<td>-4.9</td>
<td>-4.9</td>
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<td>Georgia</td>
<td>Watkinsville</td>
<td>Cfa</td>
<td>-1.2</td>
<td>0.3</td>
<td>-1.6</td>
</tr>
<tr>
<td></td>
<td>Kentucky</td>
<td>Mammoth Cave</td>
<td>Cfa</td>
<td>1.7</td>
<td>-0.1</td>
<td>1.8</td>
</tr>
<tr>
<td></td>
<td>Maryland</td>
<td>Powder Mill</td>
<td>Cfa</td>
<td>-5.1</td>
<td>-3.7</td>
<td>-5.5</td>
</tr>
<tr>
<td></td>
<td>Oklahoma</td>
<td>Fort Reno</td>
<td>Cfa</td>
<td>-8.9</td>
<td>-8.7</td>
<td>-9.1</td>
</tr>
<tr>
<td></td>
<td>South Carolina</td>
<td>Pee Dee</td>
<td>Cfa</td>
<td>-5.3</td>
<td>-5.0</td>
<td>-5.4</td>
</tr>
<tr>
<td></td>
<td>Virginia</td>
<td>Tide Water AREC</td>
<td>Cfa</td>
<td>-2.8</td>
<td>-2.9</td>
<td>-2.9</td>
</tr>
</tbody>
</table>

Table 4 results demonstrate that using different HGHX configurations, the HGHX design length percentage error using the Xing and Spiterl model results slightly change. The maximum variation is 5% and occurs in New Mexico. The Xing and Spiterl model is used for estimating ground temperature at the HGHX burial depths for a
certain time frame. When HGHX configurations change, peak ground temperature estimation errors vary. For the three HGHX configurations, the Xing and Spitler model peak ground temperature estimations errors are all less than 1.6°C (2.9°F). The ASHRAE Handbook and the ASHRAE district heating manual estimation errors are less than 2.9°C and 5.4°C (5.2°F and 9.7°F) respectively. Overall, the Xing and Spitler model gives lower ground temperature estimation errors at different depths and time of year compared to the two common methods. The HGHX design length percentage error is correlated to the peak ground temperature estimation error. Thus, using different HGHX configurations, it still gives relatively lower HGHX design length percentage error.

Figure 3 plots estimated HGHX pipe lengths using Xing and Spitler model results against the estimated HGHX lengths based on measured ground temperatures, for three HGHX configurations and twelve parametric study sites. Ideally, if designed HGHX lengths based on Xing and Spitler model results are identical to the designed value based on measured results, all these data points stay on a 45° line. Figure 3 shows that percentage errors for eleven sites are in the range of ±9.2%, with one site in the range of ±18.9%. Figure 4(a) and 4(b) plot the HGHX design length percentage error for ASHRAE Handbooks method and ASHRAE DHM method for three configurations: two pipes in one layer, two pipes in two layers and four pipes in two layers. Table 3 presents the designed length errors for two pipes in one layer configuration, the maximum errors are ±35% and ±44.1% respectively. For the other two configurations, errors are also calculated, maximum of which are ±38.3% and ±57.7% respectively. In Figure 4(a) and 4(b), at site Oregon and site Colorado, when the design HGHX lengths, measured results are at about 99.6m and 125.4m, the design HGHX lengths, ASHRAE Handbook are at about 61.5m and the design HGHX lengths, ASHRAE DHM are at about 197.8m. The green triangle dot showing the design percentage error are located on orange line which are ±38.3% and ±57.7% off the reference results.
CONCLUSIONS

This paper discusses the application of Xing and Spitler model 2016 and its impact on improving HGHX design accuracy. This study is performed for twelve locations in the U.S. and three different HGHX configurations are investigated. For each location and each HGHX configuration, the HGHX is designed with: measured ground temperatures, Xing and Spitler model results, two common methods results (ASHRAE Handbooks and ASHRAE district heating manual). The designed HGHX lengths using the three estimated ground temperatures are compared to the designed lengths using measured results respectively. The HGHX design length percentage error is found to be almost linearly correlated to the peak ground temperature estimation error. For all cases, the Xing and Spitler model estimations errors are less than 1.6°C (2.9°F). ASHRAE Handbook results errors are less than 2.9°C (5.2°F); ASHRAE district heating manual result errors are less than 5.4°C (9.7°F). Corresponding HGHX design length percentage error using the Xing and Spitler model results are within the range of ±18.9%. HGHX design length percentage error using the ASHRAE Handbook results and ASHRAE district heating manual results are within the range of ±38.3% and ±57.7% respectively. Application of the Xing and Spitler model helps improving HGHX design accuracy, and lead to a reduced capital cost of the installed GSHP system or promise a well performance of HSHP system.

REFERENCE

ASHRAE. 2013b. District Heating Guide. Atlanta: ASHRAE.


Xing, L. and J.D. Spitler. 2016b. Prediction of undisturbed ground temperature using analytical and numerical modeling. Part II: Methodology for developing a world-wide dataset. Science and Technology for the Built Environment. DOI: 10.1080/23744731.2016.1262705. ISSN: 2374-474X.


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ABSTRACT
The borehole thermal resistance is both an important design parameter and a key performance characteristic of a borehole heat exchanger. Another quantity that is particularly important for deep borehole heat exchangers is the internal thermal resistance between the upward-flowing and downward-flowing fluid channels in the borehole. The multipole method is a well-known and robust method to compute both these thermal resistances. However, it has a fairly intricate mathematical algorithm and is thus not trivial to implement. Consequently, there is considerable interest in developing explicit multipole formulas. So far zeroth-order and first-order multipole formulas have been derived for cases where the two legs of the borehole are placed symmetrically in a borehole. This paper presents new explicit second-order multipole formulas, which provide significant accuracy improvements over the previous formulas.

INTRODUCTION, SCOPE AND RATIONALE
Compared to ambient air, ground, in general, is a far superior source or sink of thermal energy because of its relatively stable temperature levels over the year. Since the turn of this century, the use of heating and cooling systems utilizing the ground as a heat source or a heat sink has grown at a remarkable rate (Lund and Boyd, 2016), stimulated by energy prices, technology advances and environmental concerns. A typical ground source heating or cooling system consists of a heat pump, a ground heat exchanger, and auxiliary systems for storage and distribution of thermal energy. The ground heat exchanger can be of open or closed type. In an open system groundwater is directly used as the heat carrier fluid, whereas in a closed system the heat carrier fluid is circulated in a closed loop, which can be horizontal or vertical. Various heat exchanger configurations can be used in closed-loop vertical systems, including single or double U-tubes, and simple or complex coaxial pipes. Among all types, a borehole heat exchanger with a single U-tube is by far the most commonly used ground heat exchanger in practice because of its low cost, small space requirements, and ease of installation. The scope of this paper is also limited to the application of single U-tubes in borehole heat exchangers.

Ground thermal conductivity ($\lambda$) and borehole thermal resistance ($R_b$) are the two principal parameters that govern the heat transfer mechanism of a borehole heat exchanger. The heat transfer outside the borehole boundary is dictated by the thermal conductivity of the ground, whereas the heat transfer inside the borehole is characterized by the borehole thermal resistance between the heat carrier fluid and the borehole wall. A high ground thermal conductivity is beneficial for the ground heat transfer. However, being an intrinsic property of ground, the ground thermal conductivity cannot be controlled in practice. On the other hand, a low borehole thermal resistance is desirable for better heat transfer inside the borehole. The borehole thermal resistance depends upon the physical arrangement and the thermal properties of borehole components including grouting, ground heat exchanger, and the heat carrier fluid. Its value can be engineered to a certain extent by optimizing the geometry and layout of the ground heat exchanger and by choosing appropriate materials for the borehole components.

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In physical terms, the thermal resistance of a borehole can be thought of as a ratio of the temperature difference between the heat carrier fluid and the borehole wall to the heat transfer rate per unit length of the borehole. This implies that reducing the borehole thermal resistance for a given heat transfer rate corresponds to minimizing the temperature difference between the heat carrier fluid and the borehole wall. There are basically two fundamental and well-established approaches for determining the thermal resistance of borehole heat exchangers: theoretical and experimental. In the theoretical approaches (Javed and Spitler, 2016), analytical or empirical formulas based on one- or two-dimensional steady-state conductive heat transfer are used to calculate the borehole thermal resistance. The borehole thermal resistance calculated by the theoretical approach is defined locally at a specific depth in the borehole. For a single U-tube heat exchanger, the heat carrier fluid temperature is the local average temperature of the fluid in the two legs of the U-tube at a specific depth in the borehole. The same is represented as Equation 1, where $T_{fl}$ is the local mean fluid temperature, $T_b$ is the mean borehole wall temperature and $q_b$ is the heat transfer rate per unit length of the borehole.

$$R_b = \frac{T_{fl} - T_b}{q_b}$$  \hspace{1cm} (1)

The thermal resistance of the borehole heat exchanger, together with the ground thermal conductivity, can be determined experimentally (Javed et al., 2012) through an in-situ thermal response test. Most thermal response test evaluation methods use the mean of the temperatures taken at the inlet and outlet of a borehole. The temperature measurements taken at the top of the borehole account for the effects of thermal short-circuiting between the upward and downward flow channels of the ground heat exchanger. The thermal short-circuiting between the counter flow channels negatively impacts the heat carrier fluid temperature, which consequently results in a higher borehole thermal resistance than calculated with Equation 1. This leads to the concept of effective borehole thermal resistance $R_b^*$, which can be defined as the effective resistance between the heat carrier fluid $T_f$, characterized by the simple mean of the inlet and outlet temperatures, and the mean borehole wall temperature $T_b$. The effective borehole thermal resistance is mathematically expressed as Equation 2.

$$R_b^* = \frac{T_f - T_b}{q_b}$$  \hspace{1cm} (2)

Unfortunately, the concepts of borehole thermal resistance and effective borehole thermal resistance have often been misunderstood both in research and practice (Javed and Spitler, 2016). This has led to indiscriminate and interchangeable use of these terms in research literature, causing confusion and discrepancies in calculating and analysing the borehole thermal resistance. The fundamental difference between the borehole thermal resistance defined by Equation 1 and the effective borehole thermal resistance defined by Equation 2 is that the former is defined locally at a specific borehole depth whereas the latter applies to the entire borehole. Depending on the depth of the borehole and the thermal capacitance of the heat carrier fluid, the effective borehole thermal resistance is higher than the local borehole thermal resistance by a few to several hundred percent (Javed and Spitler, 2016; Spitler et al., 2016a). For the most common borehole heat exchanger configurations, the effective borehole thermal resistance can be determined from the local borehole thermal resistance using the analytical expressions of Hellström (1991), Zeng et al. (2003) or Ma et al. (2015).

Hellström (1991) has derived Equations 3 and 4 for calculating the effective borehole thermal resistance of a single U-tube borehole heat exchanger. The two equations are respectively based on uniform borehole wall temperature and uniform heat flux boundary conditions along the borehole. These are both limiting boundary conditions and the real situation falls somewhere in between. Hence, the effective borehole thermal resistance is sometimes expressed as the mean value between the two equations. Calculation of effective borehole thermal resistance with Equations 3 and 4 requires knowledge of total internal thermal resistance $R_a$ and direct coupling resistance $R_{1,2}$ between the two U-tube legs, respectively, in addition to the borehole thermal resistance $R_b$. All these resistances can be calculated to a high degree of accuracy by means of the well-known Multipole method (Claesson and Hellström, 2011).
\begin{align*}
R_b^* &= R_b + \frac{1}{3} R_a \left( \frac{H}{\rho_l c_l V_l} \right)^2 \\
R_b^* &= R_b \eta \coth \eta, \quad \eta = \frac{H}{\rho_l c_l V_l} \frac{1}{2 R_b} \sqrt{1 + \frac{4 R_b}{R_{1-2}}} \tag{4}
\end{align*}

The multipole method is an analytical method based on two-dimensional steady-state conductive heat transfer in a borehole. It uses a combination of line heat sources and so-called multipoles to determine thermal resistances for any number of arbitrarily placed pipes in a composite region. The accuracy of the results increases with the number of multipoles used for the calculation. When implemented in a computer program, the order of the multipoles to be used for a calculation is typically prescribed to 10, which was the maximum possible order in the original (Bennet et al., 1987) implementation of the multipole method. Popular ground heat exchanger programs EED (Blocon, 2015) and GLHEPRO (Spitler, 2000) also use tenth-order multipoles when calculating the borehole thermal resistance. The tenth-order multipole calculations have an accuracy of over eight decimal digits (Claesson, 2012). However, on the adverse side, the multipole method has a quite rigorous mathematical formulation and a fairly complex algorithm. Its implementation in computer programs requires a considerable amount of coding — the original implementation in FORTRAN by Bennet et al. (1987) was nearly 600 lines in length. As a result, there has been considerable interest in simplifying the multipole method for typical borehole configurations. So far, closed-form multipole formulas for zeroth-order and first-order have been developed for the case of a single U-tube with symmetrical pipes.

This paper presents newly derived closed-form multipole formulas of second-order. The presented formulas include expressions for borehole thermal resistance \( R_b \), total internal thermal resistance \( R_a \) and direct coupling resistance \( R_{1-2} \). The formulas also allow the calculation of effective borehole thermal resistance from Equations 3 and 4. The accuracy of the presented formulas is established by comparing them to the original multipole method (i.e. the tenth order multipole calculation). The superiority of the explicit second-order multipole formulas over the existing zeroth-order and first-order formulas is also demonstrated.

**THERMAL Δ NETWORK FOR SINGLE U-TUBE**

The concept of thermal resistances in a borehole is best discussed with the help of a thermal resistance network. Several representations of the thermal resistance network are possible (Hellström, 1991; Liao et al., 2012; Spitler et al., 2016b), but any such representation is an approximation to reality under network-specific assumptions and restrictions. The simplest approach is to consider a \( \Delta \) thermal network as shown in Figure 1.

![Figure 1](image-url)

**Figure 1** Notations and definitions (left), and \( \Delta \) resistance network (right) for a borehole with a single U-tube.
The above network is based on heat flows $q_1$ and $q_2$, fluid temperatures $T_{f1}$ and $T_{f2}$, and thermal resistances $R_{1-b}$, $R_{2-b}$ and $R_{1-2}$ as defined by Equation 5. The resistance $R_{1-b}$ is between pipe 1 and borehole wall, resistance $R_{2-b}$ is between pipe 2 and borehole wall, and resistance $R_{1-2}$ is between pipe 1 and pipe 2. The thermal resistance network and Equation 5 both use *average* borehole wall temperature $T_{b,avg}$ instead of a uniform temperature. This is because the temperature distribution on the borehole wall is non-uniform. This is further discussed in Claesson and Hellström (2011).

$$q_1 = \frac{T_{f1} - T_{b,avg}}{R_{1-b}} + \frac{T_{f1} - T_{f2}}{R_{1-2}}$$

$$q_2 = \frac{T_{f2} - T_{b,avg}}{R_{2-b}} + \frac{T_{f2} - T_{f1}}{R_{1-2}}$$

(5)

In an actual installation, pipes 1 and 2 may be located anywhere in the borehole as long as they do not overlap each other. In reality, the position of pipes also varies along the depth of the borehole. In the absence of any a priori knowledge of the pipes position, it is customary to assume that two pipes are *symmetrically* placed about the center of the borehole. For two equal diameter pipes, this assumption leads to the conclusion that $R_{1-b} = R_{2-b}$. The problem can be further simplified by prescribing the heat fluxes $q_1$ and $q_2$ as even (i.e. $q_1 = q_2$) and odd (i.e. $q_1 = -q_2$).

Even Case

$$q_1 = q_2 \Rightarrow T_{f1}^+ - T_{b,avg} = R_f^+ \cdot q_1$$

(6)

Odd Case

$$q_1 = -q_2 \Rightarrow T_{f1}^- - T_{b,avg} = R_f^- \cdot q_1$$

(7)

Here $T_{f1}^+$ and $T_{f1}^-$ are fluid temperatures, and $R_f^+$ and $R_f^-$ are thermal resistances for even and odd cases as defined by Equations 6 and 7, respectively. The corresponding thermal networks for even and odd cases are shown in Figure 2. The subscript $J$ in Equations 6 and 7 refers to the number of multipoles considered at each pipe for the calculation. For $J = 0$, only line sources at the pipes are used. The accuracy increases with the number of multipoles used. Closed-form zeroth-order (i.e. $J=0$) and first-order (i.e. $J=1$) multipole formulas for calculating the borehole thermal resistance and the total internal thermal resistance are already available for the case of two symmetrical pipes (Hellström, 1991). In this paper newly-derived explicit second-order multipole formulas for calculating the borehole thermal resistance and the total internal thermal resistance resistances are presented.

![Resistance networks](image)

Figure 2  Δ resistance networks for (a) even and (b) odd cases.

The general expressions for borehole thermal resistance and total internal thermal resistance can be derived from the thermal networks of even and the odd cases, respectively. As can be inferred from Figure 2a, the borehole thermal resistance $R_b$ between the fluid in the pipes and the borehole wall consists of two equal resistances (each of value $R_{1-b}$) in parallel. On the other hand, as can be deduced from Figure 2b, the total internal thermal resistance $R_t$ between the two pipes consists of a pair of equal series resistances (each of value $0.5R_{1-2}$) connected in parallel to another pair of equal series resistances (each of value $R_{1-b}$).
\[ R_b = \frac{R_{1-b}}{2}, \quad R_a = \frac{2 R_{1-b} R_{1-2}}{2 R_{1-b} + R_{1-2}} \] (8)

The relationship between network resistances \( R_{1-b} \) and \( R_{1-2} \), and even and odd thermal resistances \( R^+ \) and \( R^- \) can be obtained from Figure 2 and Equations 6 and 7.

\[ R^+ = R_{1-b}, \quad \frac{1}{R^-} = \frac{1}{R_{1-b}} + \frac{2}{R_{1-2}} \] (9)

For second-order multipole (i.e. \( J = 2 \)), Equations 8 and 9 can be rearranged to give:

\[ R_{1-b} = R^+_2, \quad R_{1-2} = \frac{2 R^+_2 R^-_2}{R^+_2 - R^-_2} \] (10)

The borehole thermal resistance \( R_b \) and the total internal thermal resistance \( R_a \) from Equation 8 can now be expressed in terms of thermal resistances \( R^+_2 \) and \( R^-_2 \) using Equation 10.

\[ R_b = \frac{R^+_2}{2}, \quad R_a = 2 R^-_2 \] (11)

The explicit formulas for \( R^+_2 \) and \( R^-_2 \) are given by Equations 12 and 13. The derivation of these expressions will not be presented here due to space limitations. The full derivation and mathematical details appear in a technical report by Claesson (2016).

\[ R^+_2 = \frac{1}{2 \pi \lambda_b} \left[ \beta + \ln \left( \frac{r^2_b}{2 r_p x_p} \right) + \sigma \cdot \ln \left( \frac{r^2_b}{r^2_b - x^2_p} \right) \right] - B^+_2 \] (12)

\[ R^-_2 = \frac{1}{2 \pi \lambda_b} \left[ \beta + \ln \left( \frac{2 x_p}{r_p} \right) + \sigma \cdot \ln \left( \frac{r^2_b + x^2_p}{r^2_b - x^2_p} \right) \right] - B^-_2 \] (13)

\[ \beta = 2 \pi \lambda_b R_p, \quad \sigma = \frac{\lambda_b - \lambda}{\lambda_b + \lambda} \] (14)

The \( B^\pm_2 \) values are obtained from the following set of equations. In all following equations, the upper index ‘+’ corresponds to the even case and \( s = +1 \), and the upper index ‘−’ corresponds to the odd case and \( s = -1 \).

\[ B^+_2 = \frac{1}{2 \pi \lambda_b} \cdot b_1 \left( V^+_1 \right)^2 \left( 2 + b_2 A^+_{1,2} \right) - 2 b_1 b_2 V^+_1 V^+_2 A^+_{1,2} + b_2 \left( V^+_2 \right)^2 \left( 1 + b_1 A^+_{1,2} \right) \] (15)

\[ V^+_1 = -s \cdot p_0 + \sigma \cdot p_1 - s \cdot \sigma \cdot p_2, \quad V^+_2 = s \cdot p^+_0 + \sigma \cdot p^+_1 + s \cdot \sigma \cdot p^+_2, \quad s = \pm 1 \] (16)

\[ b_1 = \frac{1 - \beta}{1 + \beta}, \quad b_2 = \frac{1 - 2 \beta}{1 + 2 \beta}, \quad p_0 = \frac{r_p}{2 x_p}, \quad p_1 = \frac{r_p x_p}{r^2_b - x^2_p}, \quad p_2 = \frac{r_p x_p}{r^2_b + x^2_p} \] (17)

\[ A^+_{1,1} = p^+_0 \cdot s + \sigma \left[ (p_1 + 2 p_0) + p_2 (p_2 - 2 p_0) \right] \cdot s \] (18)

\[ A^+_{1,2} = -2 p^+_0 \cdot s + 2 \sigma \left[ p^+_1 (p_1 + 2 p_0) - p^+_2 (p_2 - 2 p_0) \right] \cdot s \] (19)

\[ A^+_{2,2} = 6 p^+_0 \cdot s + 2 \sigma \left[ p^+_1 (3 p^+_2 + 8 p_0 p_1 + 4 p^+_0) + p^+_2 (3 p^+_2 - 8 p_0 p_2 + 4 p^+_0) \right] \cdot s \] (20)
COMPARISON WITH EXISTING MULTIPOLe SOLUTIONS

In this section the second-order multipole formulas for borehole thermal resistance and total internal thermal resistance are compared with previously published results. The comparison is made using a reference dataset provided by Javed and Spitler (2017). The authors have compared and benchmarked the borehole thermal resistance estimations from 10 different analytical methods against the tenth-order multipole method for 216 different cases. The authors have showed that compared to other methods, the results of zeroth-order and first-order multipole formulas provide greater accuracies. In this paper, we will also benchmark the new second-order multipole formulas against the tenth-order multipole method. The second-order multipole formulas will also be compared to the zeroth-order and first-order multipole formulas to demonstrate improvements in the accuracy of the calculated results.

Table 1 provides the detailed summary of the 216 comparison cases provided by the reference dataset and used in this paper for the comparison of the second-order multipole formulas. The cases cover three different borehole diameters of 96 mm, 192 mm, and 288 mm. The U-tube outer pipe diameter value is held fixed at 32 mm for all cases. The total pipe resistance $R_p$ also remains constant at 0.05 m-K/W. For each borehole diameter, three shank spacing configurations, i.e. close, moderate and wide – corresponding, respectively, to Paul’s (1996) Configuration A, Configuration B and Configuration C – are considered. Four levels of ground thermal conductivity ranging from 1–4 W/m-K, and six levels of grout thermal conductivity ranging from 0.6–3.6 W/m-K are used. Given the existing and reasonably foreseeable values of design parameters, the 216 cases used for the comparison bracket almost all real-world single U-tube borehole heat exchangers.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Levels</th>
<th>No. of levels</th>
</tr>
</thead>
<tbody>
<tr>
<td>Ratio of the borehole radius to outer pipe diameter ($2r_b/2r_p$). Since pipe outer diameter ($2r_p$) is always fixed at 32 mm, borehole diameters ($2r_b$) are 96 mm, 192 mm, and 288 mm.</td>
<td>3, 6, 9</td>
<td>3</td>
</tr>
<tr>
<td>Shank spacing configuration; corresponds to Paul’s (1996) A, B, C configurations</td>
<td>Close, Moderate, Wide</td>
<td>3</td>
</tr>
<tr>
<td>For $n_p/2N_0 = 32$ mm, 43 mm, 64 mm</td>
<td>For $n_p/2N_0 = 32$ mm, 75 mm, 160 mm</td>
<td>For $n_p/2N_0 = 32$ mm, 107 mm, 256 mm</td>
</tr>
<tr>
<td>$\lambda$ – the ground thermal conductivity (W/m-K)</td>
<td>1, 2, 3, 4</td>
<td>4</td>
</tr>
<tr>
<td>$\lambda_g$ – the grout thermal conductivity (W/m-K)</td>
<td>0.6, 1.2, 1.8, 2.4, 3.0, 3.6</td>
<td>6</td>
</tr>
</tbody>
</table>

Figures 3–5 present a selection of the comparison results to demonstrate the efficacy of the second-order multipole formulas presented in this paper. The results shown in these figures are for a single ground thermal conductivity of 4.0 W/m-K. The left-side figures show the grout thermal resistance values, and the right-side ones show the total internal thermal resistance values, plotted against the grout thermal conductivity. Each figure presents three curves corresponding to close, moderate and wide shank spacing. The exact value of the shank spacing for each case is provided in Table 1. It must be pointed out that multipole formulas presented in the previous section, calculate the borehole thermal resistance and not the grout thermal resistance. However, in order to be consistent with the dataset provided by Javed and Spitler (2017), the values of grout thermal resistance have been calculated and presented in Figures 3–5. The grout thermal resistance values have been determined by subtracting the fixed pipe resistance of 0.05 m-K/W from the corresponding borehole thermal resistance values obtained from the multipole formulas. Computing the grout thermal resistance directly by disregarding the pipe resistance (i.e. setting $\beta = 0$) in Equations 12 and 13 gives erroneous results for all but zeroth-order multipole calculations.
Figure 3  Grout thermal resistance ($R_g$) and total internal resistance ($R_a$) for close ($2x_p = 32$ mm), moderate ($2x_p = 43$ mm) and wide ($2x_p = 64$ mm) configurations with $2n_b = 96$ mm and $\lambda = 4$ W/m-K.

Figure 4  Grout thermal resistance ($R_g$) and total internal resistance ($R_a$) for close ($2x_p = 32$ mm), moderate ($2x_p = 75$ mm) and wide ($2x_p = 160$ mm) configurations with $2n_b = 192$ mm and $\lambda = 4$ W/m-K.

Figure 5  Grout thermal resistance ($R_g$) and total internal resistance ($R_a$) for close ($2x_p = 32$ mm), moderate ($2x_p = 107$ mm) and wide ($2x_p = 256$ mm) configurations with $2n_b = 288$ mm and $\lambda = 4$ W/m-K.
Table 2. Mean and Maximum Absolute Percentage Errors in Calculation of the Grout Thermal Resistance for All 216 Cases.

<table>
<thead>
<tr>
<th>Method</th>
<th>Shank Spacing Configuration</th>
<th>Low (0.6 – 1.2 W/m-K)</th>
<th>Moderate (1.2 – 2.4 W/m-K)</th>
<th>High (2.4 – 3.6 W/m-K)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td>Mean</td>
<td>Max</td>
<td>Mean</td>
</tr>
<tr>
<td>Zeroth-order Multi pole</td>
<td>Close</td>
<td>6.1</td>
<td>12.4</td>
<td>2.9</td>
</tr>
<tr>
<td></td>
<td>Moderate</td>
<td>3.3</td>
<td>10.8</td>
<td>1.5</td>
</tr>
<tr>
<td></td>
<td>Wide</td>
<td>8.9</td>
<td>30.4</td>
<td>1.9</td>
</tr>
<tr>
<td>First-order Multi pole</td>
<td>Close</td>
<td>0.2</td>
<td>0.4</td>
<td>0.2</td>
</tr>
<tr>
<td></td>
<td>Moderate</td>
<td>0.0</td>
<td>0.2</td>
<td>0.0</td>
</tr>
<tr>
<td></td>
<td>Wide</td>
<td>0.5</td>
<td>2.2</td>
<td>0.0</td>
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<tr>
<td>Second-order Multi pole</td>
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<td>0.0</td>
<td>0.0</td>
<td>0.1</td>
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<td></td>
<td>Moderate</td>
<td>0.0</td>
<td>0.0</td>
<td>0.0</td>
</tr>
<tr>
<td></td>
<td>Wide</td>
<td>0.1</td>
<td>0.3</td>
<td>0.0</td>
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</tbody>
</table>

Results of the second-order multipole formulas for calculating the borehole thermal resistance and the total internal thermal resistance are summarized in Tables 2 and 3, respectively. It should be noted that although Figures 3–5 only showed results for ground thermal conductivity value of 4 W/m-K due to space limitations, the results presented in Tables 2 and 3 have been obtained considering all ground thermal conductivity values from 1–4 W/m-K. Each entry in these two tables represents the mean or maximum error in percentage for a sample containing two-three values of grout thermal conductivity, three values of borehole diameter, and four values of ground thermal conductivity. The errors have been determined by comparing the results of second-order multipole formulas to the tenth-order multipole method. For the ease of comparison, errors from zeroth-order and first-order multipole formulas, as reported by Javed and Spitler (2017), are included as well.

Table 2 shows that the grout thermal resistance values obtained from the second-order multipole formula are within 0.5 % of the tenth-order multipole method for all 216 cases. Also, the mean absolute percentage error of the results obtained from the second-order multipole formula is smaller than 0.2 %. In comparison, the mean and maximum absolute percentage errors for the zeroth-order multipole formula are as high as 9 % and 30 %, respectively. The first-order multipole formula has smaller errors than the zeroth-order formula. Nevertheless, compared to the second-order multipole formula, the errors from the first-order multipole formula are higher by several orders of magnitude.

Table 3. Mean and Maximum Absolute Percentage Errors in Calculation of the Total Internal Thermal Resistance for All 216 Cases.

<table>
<thead>
<tr>
<th>Method</th>
<th>Shank Spacing Configuration</th>
<th>Low (0.6 – 1.2 W/m-K)</th>
<th>Moderate (1.2 – 2.4 W/m-K)</th>
<th>High (2.4 – 3.6 W/m-K)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td>Mean</td>
<td>Max</td>
<td>Mean</td>
</tr>
<tr>
<td>Zeroth-order Multi pole</td>
<td>Close</td>
<td>23.3</td>
<td>37.6</td>
<td>6.9</td>
</tr>
<tr>
<td></td>
<td>Moderate</td>
<td>1.8</td>
<td>7.8</td>
<td>1.1</td>
</tr>
<tr>
<td></td>
<td>Wide</td>
<td>1.9</td>
<td>8.5</td>
<td>0.4</td>
</tr>
<tr>
<td>First-order Multi pole</td>
<td>Close</td>
<td>3.2</td>
<td>5.9</td>
<td>0.6</td>
</tr>
<tr>
<td></td>
<td>Moderate</td>
<td>0.2</td>
<td>0.8</td>
<td>0.0</td>
</tr>
<tr>
<td></td>
<td>Wide</td>
<td>0.3</td>
<td>1.2</td>
<td>0.0</td>
</tr>
<tr>
<td>Second-order Multi pole</td>
<td>Close</td>
<td>0.4</td>
<td>1.0</td>
<td>0.2</td>
</tr>
<tr>
<td></td>
<td>Moderate</td>
<td>0.0</td>
<td>0.0</td>
<td>0.0</td>
</tr>
<tr>
<td></td>
<td>Wide</td>
<td>0.0</td>
<td>0.2</td>
<td>0.0</td>
</tr>
</tbody>
</table>
Table 3 shows that the total internal thermal resistance values calculated from the second order multipole formulas are within 1% of the tenth-order multipole method for all 216 cases. The mean absolute percentage error of the results obtained from the second-order multipole formula never exceed 0.4%. In comparison, the zeroth-order and the first-order multipole formulas give maximum absolute percentage errors of approximately 38% and 6%, respectively. The mean absolute percentage errors of the zeroth-order and the first-order multipole expressions are as high as 23% and 3%, respectively.

Even though the second-order multipole formulas presented in this paper are more complicated than many other analytical expressions including the zeroth-order and first-order formulas, it is still simple enough to apply for computation purposes. The implementation of the second-order multipole formulas requires approximately 10 lines of coding of rather compact and simple algebraic expressions. This is a significant improvement over the original implementation of the Multipole method, which required about 600 lines of FORTRAN coding. Hence, due to their excellent accuracy and relative ease of implementation, the second-order multipole formulas are recommended for calculation of borehole thermal resistance and total internal thermal resistance for all cases where the two legs of the U-tube are placed symmetrically in the borehole.

CONCLUSION

Closed-form second-order multipole formulas for the calculation of borehole thermal resistance and total internal thermal resistance have been presented in this paper. The presented formulas can be used for all single U-tube applications where the two legs of the U-tube are symmetrically placed in the borehole. The newly-derived formulas have been compared with the original multipole method, as well as the previously-derived zeroth-order and first-order explicit multipole formulas. The second-order multipole formulas provide significant accuracy improvements over the zeroth-order and the first-order multipole formulations. The thermal resistance values calculated from the second-order multipole formulas are always within 1% of the original tenth-order multipole method. The presented formulas may also be used to estimate the effective borehole thermal resistance from the expressions defined by Equations 3 and 4.

NOMENCLATURE

- $c_f$ = Specific heat of the circulating fluid in the U-tube, J/kg-K
- $H$ = Depth of the borehole, m
- $J$ = Number of multipoles
- $p_0$ = Dimensionless parameter, dimensionless
- $p_1$ = Dimensionless parameter, dimensionless
- $p_2$ = Dimensionless parameter, dimensionless
- $q_{b}$ = Heat rejection rate per unit length of borehole, W/m
- $q_{1}$ = Heat rejection rate per unit length of pipe 1, W/m
- $q_{2}$ = Heat rejection rate per unit length of pipe 2, W/m
- $r_{b}$ = Radius of the borehole, m
- $r_{p}$ = Outer radius of the pipe making up the U-tube, m
- $R_a$ = Total internal borehole thermal resistance, m-K/W
- $R_b$ = Local or average borehole thermal resistance between fluid in U-tube(s) to borehole wall, m-K/W
- $R_b^*$ = Effective borehole thermal resistance, m-K/W
- $R_g$ = Grout thermal resistance; resistance between outer pipe wall of U-tube to borehole wall, m-K/W
- $R_{J_f}^{±}$ = Thermal resistance for even and odd cases for $J$ multipoles, m-K/W
- $R_{2}^{±}$ = Thermal resistance for even and odd case for second-order multipoles, m-K/W
\[ R_{1-2} = \text{Thermal resistance between U-tube legs 1 and 2, m-K/W} \]
\[ R_{1-b} = \text{Thermal resistance between U-tube leg 1 and borehole wall, m-K/W} \]
\[ R_{2-b} = \text{Thermal resistance between U-tube leg 2 and borehole wall, m-K/W} \]
\[ R_p = \text{Total fluid-to-pipe resistance for a single pipe – one leg of the U-tube, m-K/W} \]
\[ T_b = \text{Borehole wall temperature, °C} \]
\[ T_{b avg} = \text{Average temperature at the borehole wall, °C} \]
\[ T_f = \text{Mean fluid temperature inside the U-tube, °C} \]
\[ T_{f,\ell} = \text{Local mean fluid temperature, °C} \]
\[ T_{f1} = \text{Fluid temperature in U-tube leg 1, °C} \]
\[ T_{f2} = \text{Fluid temperature in U-tube leg 2, °C} \]
\[ T_{f1}^\pm = \text{Fluid temperature in U-tube leg 1 for even and odd cases, °C} \]
\[ V_f = \text{Volume flow rate of the circulating fluid in the U-tube, m}^3/\text{s} \]
\[ x_p = \text{Half shank spacing i.e. half of center-to-center distance between two legs of the U-tube, m.} \]
\[ \beta = \text{Dimensionless thermal resistance of one U-tube leg, dimensionless} \]
\[ \lambda = \text{Thermal conductivity of the ground, W/m-K} \]
\[ \lambda_b = \text{Thermal conductivity of the grout, W/m-K} \]
\[ \rho_f = \text{Density of the circulating fluid in the U-tube, kg/m}^3 \]
\[ \sigma = \text{Thermal conductivity ratio, dimensionless} \]

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Modelling of Alternative Borehole Configurations for Geo-Exchange

Ying Lam E. Law  Seth B. Dworkin

ABSTRACT

During the operation of a ground source heat pump (GSHP), the ground acts as a heat sink and heat source in cooling and heating modes, respectively. When the heating and cooling loads are extremely unbalanced, ground temperature can slowly migrate up or down in the long term, diminishing the GSHP system’s performance, and eventually causing the system to fail. This failure occurs when the ground can no longer accept or provide more heat for a building. Therefore, a method to mitigate thermal imbalance is needed.

Previous studies in the literature examine the effects of borehole configurations in geo-exchange. However, no study has been done to analyze the effects of varying borehole lengths in a bore field. The objective of this study is to examine the effects on thermal performance from changing the length of individual boreholes while retaining the same total borehole length. In this paper, the four centre boreholes in a 4x4 borehole system were shortened and the length of the remaining boreholes was recalculated to meet the total required ground loop length. A 20 year operation was simulated for a school building model with centre borehole lengths of 100 m, 80 m, and 50 m and separation distances of 3 m, 4 m, and 6 m, to study the benefits of shortening the centre boreholes. The results demonstrate that by adjusting the length of the centre boreholes, separation can be reduced.

INTRODUCTION

Ground-source heat pump (GSHP) systems use the ground as a stable heat transfer medium to provide heating and cooling for a building. Heat is extracted/released into the ground during heating and cooling modes, respectively. Because the ground, approximately 10 m beneath the surface, remains at approximately the same temperature throughout seasonal fluctuations, it is a stable medium for heat transfer. When the heating and cooling demands of a building are balanced, ground temperature remains steady over time. However, when heating and cooling demands of a building are greatly imbalanced, ground temperature can slowly climb or decline. An increase (or decrease) in ground temperatures can cause a degradation in system performance because of the inefficient heat transfer temperatures. Many systems in the past had to stop operation because of low system efficiencies, making them uneconomic to operate.

Borehole configurations can play a major role in ground temperature changes. In many cases, borefield sizing is determined based on peak heating and cooling demands. Compensation for thermal imbalance is more of an afterthought and much more complex in practice. The problem of ground temperature change is still a problem in many installations due to costly overdesigns or degradation in functionality. Borehole performance is immediately affected by neighbouring boreholes (Koohi-Fayegh and Rosen 2012). Borehole systems with small separation distances experience more thermal interference between neighbouring boreholes than systems with greater separation distances (Koohi-Fayegh and Rosen 2012). The increase in separation distance can also increase the percentage of temperature restoration due to the reduction in borehole thermal interference (Yuan, et al. 2016). In addition, the temperature distribution of the borehole field can also be affected by borehole separation distances. When studying the temperatures at the centre, side, and corner of the borehole field, the greatest change in temperature can be found in the centre of the configuration. However, when borehole separation distances are increased, the difference between

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the temperatures at these locations are reduced (Yuan, et al. 2016).

While borehole separation is important in the design of geo-exchange, studies have shown that borehole separation distance is not the only factor that contributes to thermal imbalance. The study described in Law and Dworkin (2016) showed that by varying the aspect ratio of borehole configurations, ground temperature changes can be slightly alleviated. The increase in borehole field perimeter was able to slightly lower the changes in ground temperature due to the larger area available for heat to dissipate to the surrounding soil. The decrease in ground temperature change can allow the system to operate efficiently for a longer period of time.

In an array of boreholes, the centre boreholes of the configuration are most affected by thermal imbalance due to neighbouring borehole interactions. These boreholes are the least effective because of the poor heat transfer temperatures in the surrounding soil. A method to alleviate ground temperature change is to remove inner boreholes (Bayer, et al. 2014). In this method, the least effective boreholes (inner boreholes) were removed from the borehole field to prevent thermal accumulation in the centre of the borehole fields. The results in Bayer, et al. (2014) showed that the cavity created by the removal of inner boreholes contributed to better heat transfer, resulting in smaller changes in ground temperature. This method moves the location of thermal accumulation away from the centre of the borehole field.

Another method to alleviate the effects of thermal imbalance is to alter borehole configurations in the axial direction. Examples of this method include the installation of inclined boreholes (Marcotte and Pasquier 2009). During the construction phase of a GSHP system, instead of drilling boreholes vertically into the soil, boreholes can be drilled on an incline, away from the centre of the borehole field. A dip angle was defined as the angle at which the borehole is oriented away from the vertical direction. A reduction in temperature changes was observed by changing the dip angle of the boreholes (Marcotte and Pasquier 2009). In addition, borehole length savings were also achieved (Marcotte and Pasquier 2009). The inclination of boreholes pose a beneficial design in geo-exchange because their installation does not incur a greater cost in construction.

While most simulation methods use the superposition of g-functions to determine the temperature distribution in a borehole field, these studies neglect axial effects. Axial effects are important in the simulation of boreholes especially in short borehole systems with unbalanced loads (Marcotte, et al. 2010). A study by Marcotte, et al. (2010) compared finite and infinite line-source models for the same borehole system. The infinite line-source model predicted a single temperature for the entire length of borehole, however, the finite line-source model predicted a gradient of temperatures approaching the infinite line-source temperature as the depth of the borehole increased. In the prediction of percentage borehole freezing, the discrepancy between the infinite and finite line-source models was 48% (Marcotte, et al. 2010). Due to the large discrepancies in the results of the two models, it is important to consider axial effects, especially in shallow boreholes.

Studies have indicated the importance of studying borehole separation distances, borehole configurations, and axial effects; there is a need to combine the effects of all three aspects in a borehole field. Borehole designs typically consist of various borehole field layouts of uniform length boreholes. However, no study has been done to assess the effect of operating a GSHP system with boreholes of varying length.

In this study, a 4x4 borehole configuration was studied. The four centre boreholes in the 16 borehole system were shortened and the length of the remaining boreholes was recalculated to maintain system capacity. A finite-element model was created to demonstrate the operation of a 16 borehole system with varying borehole lengths. The simulation was computed for a 20-year operating duration at hourly time-steps. Ten cases were simulated, each with different combinations of centre borehole length and borehole separation distance.

**METHODOLOGY**

This study consists of finite element modelling of alternative borehole configurations. The alternative borehole arrangement is illustrated in Figure 1. The centre four boreholes (depicted in grey) are shortened and the remaining boreholes (depicted in black) are extended relative to a base case. The borehole lengths used in this study are
summarized in Table 1. For example, in the ‘80’ configuration, the four centre boreholes are each 80 m in length and the remaining boreholes are each 106.67 m.

![Figure 1](image.png)

**Figure 1**  Alternative borehole arrangement layout

The test case is a high school building located in Toronto, Canada that had a GSHP considered. The GSHP was designed to meet 100% of the building’s heating and cooling capacities. The peak loads of the building are 170 kW and 145 kW for cooling and heating, respectively. The “normal” sizing process is to determine the required ground loop length using the method presented by Kavanaugh & Rafferty (1997) for equal borehole lengths.

<table>
<thead>
<tr>
<th>Borehole configuration</th>
<th>Inner borehole length (m)</th>
<th>Outer borehole length (m)</th>
</tr>
</thead>
<tbody>
<tr>
<td>100</td>
<td>100</td>
<td>100</td>
</tr>
<tr>
<td>80</td>
<td>80</td>
<td>106.67</td>
</tr>
<tr>
<td>50</td>
<td>50</td>
<td>116.67</td>
</tr>
</tbody>
</table>

First, hourly heating and cooling demands of a building were processed to obtain hourly borehole wall heat fluxes. Then, a finite element geometry as illustrated in Figure 1 was created in COMSOL Multiphysics (COMSOL Inc., 2016). Hourly heat fluxes were applied to the borehole walls to simulate a borehole providing heating and cooling for a building. The operation of the system was simulated for a 20 year period with hourly time steps. The results of the simulation are presented in the following sections.

**Model Set Up**

The method employed to determine the hourly heat flux is the same as the one used in (Law and Dworkin, 2016). The variation in COP during the operation of the GSHP was not considered in this study. Since the purpose of this study was to compare the various ground loop configurations, constant COPs were used since they would affect the test cases similarly. Although applying variable COPs may change the quantitative results, it is unlikely that they will greatly impact the qualitative comparisons presented in this study. Hourly heating and cooling demands were processed into hourly heat fluxes using MATLAB (MathWorks Inc., 2016). Positive heat flux represents the release of heat into the borehole field and negative heat flux represents the extraction of heat from the borehole field. The hourly heat flux was calculated by determining the hourly “ON” and “OFF” cycles of the heat pump. For every hour that the heat pump is turned “ON”, the maximum heating or cooling capacity is provided. For every hour that heating
or cooling is not required, the system turns “OFF”. Residual heating or cooling from the previous hour can be used to supply the building with heating or cooling when the system is “OFF”. For example, if 10 kWh of heating is the capacity of the heat pump and the heating demand of the building for the first two hours are 5 kWh each, the GSHP can turn “ON” for the first hour and be “OFF” for the second hour. The same approach as in (Law and Dworkin, 2016) was taken to calculate the hourly heat flux, upon determining the “ON” and “OFF” cycles of the heat pump.

Test case and simulation properties

The test case used in this analysis is a school building with slightly unbalanced heating and cooling loads. While a variation in borehole lengths can slightly reduce the effects of thermal imbalance, highly unbalanced loads are not recommended due to their large thermal impact on the soil. The net heating and cooling demands of this building can be found in Figure 2. The heating and cooling demands of the school building exhibit an interesting pattern in which the cooling demands drop significantly in the summer months when the occupancy of the school is low.

![Figure 2](image)

Figure 2  Net heating and cooling demands of a school building.

Using the same calculation method as outlined in (Law and Dworkin, 2016), the hourly heat fluxes were determined based on the hourly heating and cooling demands. The GSHP system was sized to its maximum heating and cooling capacities. The flow rate in all boreholes are considered to be constant and at system capacity. A simple on-off logic is used to maintain building comfort. Although variable speed pumps with varying system capacity are now available, in practice the majority of systems still operate in on-off scenarios using single speed pumps. Since the characteristic time of heat dissipation through the ground is much greater than that of on-off switching, the impact of this assumption is expected to be negligible. At each hour, it was determined whether the system was “ON” for heating, “ON” for cooling or “OFF” for both. It was assumed that turning the system “ON” would supply a building with its maximum capacity.

The heating and cooling systems' “ON” and “OFF” conditions were determined for each of the 8760 hours of the year and hourly heat fluxes are calculated based on these conditions. The hourly heat fluxes for the school building are presented in Figure 3. Lines in dark gray represent positive heat flux where heat is transferred into the ground. Lines in light gray represent negative heat fluxes where heat is extracted from the ground. “ON” and “OFF” cycles of the system can be observed in Figure 3 by the occasional spacing between hours.
The simulation was conducted in COMSOL Multiphysics for a 20 year period at hourly time steps (COMSOL Inc., 2016). At each hour, the boundary heat flux condition at the borehole walls are updated with the hourly heat fluxes calculated in the previous step. A quarter of the simulation domain was created to model 4 operating boreholes. Symmetry conditions were applied to two faces to mirror the domain. Using the symmetry conditions, a 16 borehole system can be simulated by only modelling 4 boreholes. This simplification of geometry allowed for a reduction in computation time. Open boundary conditions were applied to the remaining faces to model far-field conditions.

The simulation was repeated for 9 different cases of borehole configurations. The configurations are selected based on combinations of different borehole spacing and centre borehole length as presented in Table 2. For example, case 2 is a 16 borehole installation in a 4x4 configuration. The inner 4 boreholes in the system are 80 m in length and the remaining 12 are 106.67 m in length. The 3 m borehole spacing indicates that the boreholes are spaced 3 m apart from each other.

### Table 2. Borehole configurations

<table>
<thead>
<tr>
<th>Case</th>
<th>Borehole spacing (m)</th>
<th>Centre borehole length (m)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>3</td>
<td>100</td>
</tr>
<tr>
<td>2</td>
<td>3</td>
<td>80</td>
</tr>
<tr>
<td>3</td>
<td>3</td>
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</tr>
<tr>
<td>10</td>
<td>5</td>
<td>50</td>
</tr>
</tbody>
</table>

### RESULTS

**Borehole wall temperature**

In this simulation the temperature at a point near the borehole wall is studied. This temperature is extracted
from a point 1 cm away from the borehole wall (point A). The location of 1 cm was chosen because it is very close to the borehole wall, and therefore can represent borehole wall temperature. A parallel line 1 cm away from the borehole wall was considered along each borehole and the average temperatures of those lines were calculated. The yearly average borehole wall temperature were calculated and summarized in Figure 4.

In Figure 4, three cases are presented. The three cases have centre borehole lengths of 100 m and vary in separation distance (3 m, 4 m, or 6 m). Figure 4b and Figure 4c are similar for centre borehole lengths of 80 m and 50 m, respectively. From the figures, it is evident that the borehole wall temperatures are highest in the case with the 3 m separation distance and lowest in the case with the 6 m separation distance.

![Figure 4](image)

**Figure 4** Average temperature at point A for a) 100 m, b) 80 m, and c) 50 m centre borehole length configurations

In Figure 4c, it is interesting to note that the temperature difference between the 3 m and 4 m separation distance is significantly greater than the temperature difference between the 5 m and 6 m cases. The rate of increase in ground temperature is higher when separation distance is small and is lower when the separation distance is greater. The rate of increase in ground temperature is higher when separation distance is small because there is less soil volume available between boreholes for heat to dissipate to.

The effects of varying borehole lengths were also studied. The results are presented in Figure 5 for the cases with 4 m borehole separation distances. The plots for 3 m and 6 m separation distances are not presented in this
analysis because of their similarities with Figure 5. In Figure 5, the average temperatures at point A were studied for three borehole configurations. It can be observed from the figure that the “50” configuration has a smaller increase in temperature compared to the “80” and “100” configurations over the 20 year study period. Thermal benefits in using the “50” configuration can be observed. Using the same amount of drilling and piping, over 20 years, the “50” configuration on average has a 0.5°C lower temperature than the “100” configuration. Due to the lower increase in borehole wall temperature, more effective heat transfer and better COP management can occur.

![Figure 5](average_temperature.png)

**Figure 5**  Average temperature at point A at 4 m separation distance

**Alternative designs**

In the hopes of developing ways to alleviate the effects of thermal imbalance, the ground temperature changes in all 10 configurations indicated in Table 2 were compared. Annual average temperatures were calculated for 20 years for each system and summarized in Figure 6. Based on the results in this figure, it can be noticed that ground temperature increases with decreasing separation distance and increasing centre borehole length. As borehole wall temperature increases, ground temperature also increases due to the heat dissipated from the borehole wall into the surrounding soil. Although in all cases, ground temperature increases as time increases, the overlapping/shadowing lines may lead to potential alternative designs.

Several potential designs can be considered based on the results presented in Figure 6. If a GSHP system was originally designed with a uniform borehole field (100 m boreholes) with separation distance of 6 m (100, 6 in figure), alternative configurations can be determined using lines near the proposed design line in Figure 6. For example, a “50, 6” design can be used to reduce the ground temperature increase.

Alternatively, based on the simulation results, a “50, 5” design can be proposed. It can be observed that the behaviour of his line is very similar to the original “100, 6” design. This new design would decrease borehole spacing, allowing the system to be installed in a smaller space. This finding can provide potential designs for buildings that have property sizes that are determined to be insufficient for installation of GSHP systems.
CONCLUSIONS

Although studies have previously been done to characterize the effects of borehole geometry in terms of borehole field arrangements and borehole separation distance, little research had been done to explore the effects of alternating the length of boreholes. To fill the knowledge gap in the study of borehole lengths, a series of simulations were performed in this work.

The operation of a 4x4 configuration borehole field with varying borehole lengths and borehole separation distance was simulated. Although variable speed pumps are available, an on-off scenario with single speed pumps was modeled in this study as many systems still operate using this technology. Ten cases were studied and an increase in ground temperature near the borehole wall can be observed with the decrease of borehole separation distance. In addition, an increase in ground temperature near the borehole wall can also be observed with the increase of inner borehole depth. It was observed from the present results that alternative designs can be used to achieve the same ground temperature change with uniform borefield lengths. Specifically, it was found that the configuration with inner borehole length of 50 m at a separation distance of 5 m behaved similarly as the configuration with inner borehole length of 100 m at a separation distance of 6 m. This result indicates that borehole separation distance can be reduced by alternating the borehole lengths of a system. This reduction can allow GSHP systems to be installed in smaller spaces, which is beneficial to locations where land prices are high, or properties are dense.

This study demonstrated the importance of the study of alternative borehole configurations. Alternative borehole configurations can be used to reduce the effects of ground thermal imbalance. To fill the knowledge gap present in this study, further study should be performed on the effects of alternating borehole lengths in other borehole field arrangements. The study of varying borehole lengths should be extended to incorporate different borehole field layouts. In addition, refined details should be added to the current model to develop models that are more representative of operating GSHP systems. Such details should include but not limited to fluid flow inside pipes, additional soil properties and varying COPs.

ACKNOWLEDGEMENTS

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The Effect of an Off-Peak Ground Pre-Cool Control Strategy on Hybrid Ground-Source Heat Pump Systems

Adam A. Alaica  Dr. Seth B. Dworkin

ABSTRACT

Hybrid Ground-Source Heat Pump (HGSHP) systems have been introduced as an alternate system configuration to remedy the current financial hurdles associated to the installation of geo-exchange technology. However, there still remains potential for increased economic feasibility with the addition of improved system control. This study introduces an operational strategy referred to as an ‘Off-Peak Ground Pre-Cool’, employing time-of-use conscious operating logic to facilitate artificial bore-field pre-conditioning. Artificially pre-cooling a system’s bore-field during an off-peak operating bracket allows for improved thermal characteristics for the following peak period. With improved bore-field thermal characteristics during peak periods, cooling mode operation can be exploited more efficiently, resulting in a reduction in peak power consumption and operating costs. This study presents a preliminary evaluation of the impact the proposed off-peak ground pre-cool strategy has on the operation of a HGSHP system, simulated for a mid-rise multi-residential facility located in Toronto, Canada. Two analyses are presented simulating the strategy’s impact as a function of pre-cool duration and hybrid system proportions. This study explores the potential benefit that a proactive bore-field pre-condition poses for the operation of a HGSHP system, intending to concurrently address improving system economics and aid in the balancing of the electrical grid.

INTRODUCTION

There is a prominent need for the development and integration of sustainable energy alternatives to alleviate our current dependence on energy sources that produce substantial carbon footprints. When assessing a nation’s annual energy consumption, the building sector accounts for a significant portion. According to the U.S Energy Information Administration, the building sector consumed 47.6% of total energy used in the United States, as of 2012 (EIA 2012). With space heating/cooling requirements typically accounting for 50% of a building’s annual energy usage, the integration of high efficiency alternatives present the potential for a significant reduction in energy consumption, operating costs, and green-house gas emissions (NRCan 2013).

Hybrid Ground-Source Heat Pump (HGSHP) systems provide a sustainable means of space heating/cooling, pairing a geo-exchange system with auxiliary heating/cooling units, or in the case of a retrofitted installation, the existing conventional heating/cooling systems are utilized. Hybridization is used as a technique to provide flexibility in the design process, allowing for economic optimization techniques to be utilized as suggested by Alavy et al., (2013). Hybrid design procedures size the geo-exchange component to meet a percentage of buildings base load requirements. During peak hours of operation, the auxiliary system provides supplementary assistance to ensure the building’s demand is met (Hackel and Pertzborn 2011). Solar assisted ground-source heat pump (SAGSHP) systems have been proposed as alternative hybrid configuration, allowing for improved heating mode efficiency by the means of a bore-field thermal

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storage strategy. This technique employs a solar thermal collector to heat the ground-loop working fluid, which is circulated through a bore-field to pre-heat the surrounding soil prior to a dwelling requiring heat. The literature suggests significant improvements in solar assistant geo-exchange system efficiency, resulting in a 28.1% increase in bore-field heat exchange rates and 8.74% to 9.3% increase in the coefficient of performance (COP) (Nam et al. 2015; Verma and Murugesan 2014).

A performance assessment study (Jassen et al. 2015), conducted by the Toronto and Region Conservation Authority of urban geo-exchange projects in the Greater Toronto Area, presented valuable insight on various research areas to further the potential of geo-exchange technology. In this study, time-of-use (TOU) based control strategies were highlighted as an area for further research and development, indicating a potential to reduce electricity costs from 20 to 25%. Aside from the cost savings associated with TOU control, additional benefits are provided to utilities in the form of electrical load leveling, which helps alleviate the pressure placed on the grid during peak hours (Jassen et al. 2015). The study presented by Carvalho et al., (2015) proposed a TOU-conscious demand side management strategy; using a GSHP as a flexible load to artificially consume energy in off-peak periods to pre-heat a service building. The building pre-heat allowed for a portion of the GSHP’s operating cycles to be isolated within off-peak operating periods; resulting in a 34% reduction in electricity costs, due to reduced electricity rates.

The scope of this paper is to provide a preliminary evaluation of a proactive control strategy for cooling mode operation of HGSHP systems. Referred to as the ‘off-peak ground pre-cool’ (OGPC) control strategy, this methodology utilizes the auxiliary cooling system as a flexible load to artificially consume electricity during off-peak TOU brackets to remove heat from the ground, when energy costs are most economical. This study aims to demonstrate that a HGSHP system operated with an OGPC strategy can exploit the bore-field’s/ground’s thermal mass with a pre-cool, creating improved thermal characteristics during the following mid-peak/peak periods. With a TOU-conscious operating strategy, the proposed scheme has the potential to address improving system economics through an increase in operating efficiency and concurrently aiding in the balancing of the electrical grid.

The analysis in the present study was conducted by the use of a newly developed numerical model, characterizing the operation of a HGSHP system. The model was used to simulate the response of a HGSHP system when exposed to two operating scenarios as: (1) base case set-point control and (2) an off-peak ground pre-cool operating strategy. Simulations were conducted to predict the impact the proposed strategy has on energy consumption, operating cost, and peak power reduction for three suggested pre-cool schedules; shoulder, peak, and full season. The details of these proposed schemes will be further explained in the following section.

**METHODOLOGY**

The methodology applied in the presented study involves a three-part procedure. First, building energy simulations (BES) were conducted to generate estimates of the mid-rise building’s annual hourly heating/cooling loads. Second, the annual hourly heating/cooling loads are utilized as input variables to design an economically optimized HGSHP system, following the rigorous computerized design methodology outlined in Alavy et al., (2013). The results of the BES and the optimized HGSHP are used as input parameters to the numerical performance prediction model. Further detail of the methodology applied in this study is provided in the following subsections.

**Building Energy Simulation and Hybrid System Design**

In this study, annual hourly thermal loads were generated for a mid-rise multi-residential building, located in Toronto, Canada using eQuest software. The simulated results initially allow for both heating and cooling to be supplied in a simultaneous fashion during each time interval. However, the heating and cooling loads are corrected under the assumption the building’s demand can be satisfied with an internal mechanism before relying on the compensation from the geo-exchange system. In this analysis, a common water loop distribution system has been assumed, by neglecting the power consumption of the internal mechanism, the net demand will be provided by the GSHP to/from the common water loop distribution system.
The computerized design methodology presented by Alavy et al., (2013) is used in this study to size an economically optimized HGSHP system. This design algorithm is based on the governing equations outlined in ASHRAE’s design strategy (Kavanaugh and Rafferty 1997), automatically sizing the geo-exchange system and auxiliary equipment to meet peak building demands. Introducing a shave factor (α), defined as the portion of peak demand met by the geo-exchange system, the variable hybrid designs can be determined by fluctuating this factor between zero and one. The capital and operating costs for each shave factor value are calculated and discounted into a net present value. The optimal hybrid system design is selected for the shave factor associated to the lowest cost in net present value. The following Table 1 provides a summary of the modified hybrid design parameters selected in this study, all other variables were held constant, presented in Alavy et al. (2013).

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Heat Pump</th>
<th>Entering Sink Temperature</th>
<th>EER</th>
</tr>
</thead>
<tbody>
<tr>
<td>Cooling design specifications</td>
<td>ClimateMaster Tranquility TT</td>
<td>25.1 °C</td>
<td>13.6</td>
</tr>
</tbody>
</table>

For the HGSHP system design the ClimateMaster Tranquility TT heat pump is considered, with a dual stage heat pump unit capable of full load and 67% part load operation. The auxiliary cooling system selected in this study is a non-reversible air-source heat pump (water-to-air), with an assumed average COP of 3.2 (Esen et al. 2007).

**Numerical Model**

In this study, in order to simulate and analyze the off-peak ground pre-cool control strategy, a numerical model was developed using MATLAB to characterize the operation of a HGSHP system for an annual duration. The model is deterministic in nature and simulates a load-based analysis determining which mechanical system has operational authority in each simulated time step. The model runs a yearly simulation with a time-step of 20 minutes for a total of 26,280 iterations. The generated model consisted of three primary inputs; annual hourly heating/cooling loads (eQuest), optimal HGSHP system specifications (Alavy et al., 2013), and TOU electrical cost structure (off-peak: 8.0 ¢/kWh, mid-peak: 12.2 ¢/kWh, peak: 16.1 ¢/kWh). The input data is then processed to determine important characteristics of the building’s thermal requirements, such as total heating/cooling demand required, peak heating/cooling loads, and whether the building under consideration is heating or cooling dominant. The capacities of the GSHP and auxiliary systems are then determined based on the optimal shave factor (α), defined as the percentage of peak cooling demand met by the geo-exchange system, with the remaining load (1 - α) being supplied by the auxiliary systems. The numerical simulation is initiated as a set-point control algorithm fitted to the model is used to determine the operational authority of the sub-systems, depending on the building’s thermal requirement in the current time-step. For every time-step of the numerical simulation, the portions of demand supplied by the geo-exchange system and the auxiliary systems are determined; the variation in the GSHP COP, and the bore-field average temperature response is predicted simultaneously.

The functionality of the variable GSHP is determined from a combination of experimental and numerically simulated results. The GSHP system transient COP response as a function of cycle time is estimated for experimental data presented by Alzahrani (2013). Utilizing the numerically generated data presented by Nam et al. (2015), a set of linear correlations were generated and used to predict the bore-field’s response to pre-cooling and resulting impact on GSHP cooling output/COP. The developed numerical model simulates two physical responses to the presence of a pre-cooled bore-field, an increase in the rate of heat rejection to the ground during cooling operation (increase in cooling output) and an increase in the GSHP system COP (reduced compressor compensation). The increase in cooling output is due to the assumed reduction in ground temperature, resulting in an average increase in heat injection per meter length of the ground-loop. The increase in heat pump COP in cooling mode is due to an assumed reduction in compressor compensation, resulting from lower entering fluid temperatures.
Off-Peak Ground Pre-Cool Control Strategy

The following Figure 1 is the proposed system orientation required to implement a ground pre-cool. The assumed hybrid configuration uses a series connection of the auxiliary cooling system with the ground-loop. The introduction of an additional operating loop is presented in Figure 1, accomplished by actuating V1 – V4 into their flow diverting state, resulting in the decoupling of the GSHP from the bore-field heat exchanger (BHE) circuit. With the auxiliary cooling system acting as the only active load, bore-field pre-conditioning would be made feasible. Figure 1 illustrates a simplified conceptual schematic of how pre-cool operation would be made feasible. The hybrid system model simulates the GSHP and ASHP in a series configuration on the distribution system (common water loop); during peak demand this configuration produces a net capacity capable of meeting the buildings requirements. Pre-cooling is accomplished through a secondary circulation loop coupling the ASHP to the ground-loop.

![Figure 1 HGSHP Functional Schematic](image)

The primary control technique used in the operation of the HGSHP system is a conventional set-point control scheme. This strategy is used in this study as a base case to reveal the potential benefits that an OGPC has on the operation of a HGSHP system. The OGPC algorithm’s operational authority is restricted to the control of only the auxiliary cooling system, under the condition that specific logic indicators are satisfied. The preliminary stage of the OGPC strategy utilizes two control variables to determine the beginning of an OGPC cycle, being: time of the year and TOU operating bracket. The time of year was utilized as a control variable in the OGPC algorithm to permit flexibility in pre-cool operation, allowing for time spans of various cooling load densities to be targeted by the algorithm. In this study, three operating schedules were analyzed; ‘shoulder season’ (SS) operation (April 1st – May 31st), ‘peak season’ (PS) operation (July 1st – August 31st), and ‘full season’ (FS) operation (April 1st – September 30th). The TOU control variable is a critical component in the OGPC algorithm where a pre-cool operation is strictly restricted to off-peak TOU brackets (19:00 – 7:00), taking full advantage of low electrical energy prices. For the suggested schedules, pre-cooling is designated to weekday operation, excluding weekends and holidays which are fixed to an off-peak utility rate.

RESULTS

The following sections present the simulated results for the mid-rise multi-residential building fitted with the three previously suggested off-peak pre-cool schedules. For this analysis, a single year of cooling season operation was evaluated, as the proposed operating strategy’s scope is limited to improving system performance in cooling mode. The base case strategy utilizes the GSHP as the primary mechanical system, meeting base load demand, and is only assisted by the auxiliary cooling system when the demand exceeds the GSHP system’s capacity. Table 2 presents a summary of the mid-rise’s building energy simulation and optimal hybrid system design results. The results indicated an optimal
A shave factor of 0.23 for a peak cooling load of 274.5 kW, with a resulting ground-loop length of 2115.5 meters. The building is characterized as cooling dominant, with a total heating-to-cooling ratio of 0.25.

### Table 2. Summary of building/hybrid system characteristics for the mid-rise

<table>
<thead>
<tr>
<th>Peak Cooling Load</th>
<th>Shave Factor</th>
<th>Ground-Loop Length</th>
<th>Total Cooling Demand</th>
<th>Heating/Cooling Ratio</th>
</tr>
</thead>
<tbody>
<tr>
<td>274.5 kW</td>
<td>23%</td>
<td>2115.5 m</td>
<td>573.7 MW</td>
<td>0.25</td>
</tr>
</tbody>
</table>

### The Effect of Pre-Cool Duration on HGSHP System Performance

To simulate the effect of the OGPC control strategy on the presented hybrid geo-exchange system, a primary analysis was conducted in which 36 unique annual simulations were carried out. In these analyses, only the pre-cool duration were varied; the simulations were conducted with 20 minute increments of pre-cool time, up to a maximum of 12 hours (full off-peak operating bracket). Upon completion of this study, the OGPC strategy was analyzed to determine the peak power reduction potential resulting from the introduced thermal benefit of the pre-conditioned bore-field. The simulations were conducted on the mid-rise building, for an optimally sized HGSHP system corresponding to a shave factor of 23%.

Figure 2 presents the simulated results for the mid-rise building fitted with the proposed off-peak ground pre-cool schedules, with the base case line acting as a datum reference for visual comparison. In Figure 2a, the mid-rise’s AECC is presented as a function of pre-cool duration. It can be seen that all three pre-cool schedules exhibit similar trends, where longer pre-cool periods are required to realize the potential benefit for the proposed bore-field pre-conditioning. Both the shoulder season (SS) and full season (FS) schedules begin to illustrate energy savings potential after pre-cool periods greater than 8 hours. In Figure 2b, the mid-rise’s ACC is presented for varying pre-cool durations. Figure 2b indicates the shoulder season pre-cool schedule shows the greatest potential for operating cost savings for pre-cool periods greater than 9 hours; with negligible and no benefit illustrated with the FS and PS schedules, respectively. From the simulated results, it can be seen that the pre-cool strategy shows its greatest potential when implemented in a seasonal period of lower cooling load density (shoulder season). This benefit is realized because greater pre-conditioning can be accomplished due to a lower frequency of bore-field heat rejection.

![Figure 2](image)

**Figure 2** (a) AECC and (b) ACC versus pre-cool duration

Table 3 presents the simulated results for the mid-peak/peak power reduction potential for the mid-rise building. It can be seen that the three suggested pre-cool schedules indicate the potential for significant reduction in mid-peak/peak power consumption. When considering the reduction potential associated to the peak season operating schedule, it is clear that there is less of an incentive for bore-field pre-conditioning during seasonal periods of typically high cooling load requirements; indicated by the lower mid-peak/peak power percent reduction.
Table 3.  **Mid-peak/peak power reduction potential for proposed pre-cool schedules**

<table>
<thead>
<tr>
<th>Operating Bracket</th>
<th>Shoulder Season (SS)</th>
<th>Peak Season (PS)</th>
<th>Full Season (FS)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Mid-Peak</td>
<td>16.3%</td>
<td>10.4%</td>
<td>26.6%</td>
</tr>
<tr>
<td>Peak</td>
<td>16.6%</td>
<td>8.8%</td>
<td>25.3%</td>
</tr>
</tbody>
</table>

**The Effect of HGSHP System Proportions on Pre-Cool Benefit**

To determine the proposed operating strategy’s sensitivity to the hybrid system proportions, 99 unique simulations were conducted. In these simulations only hybrid system specifications were varied, corresponding to a shave factor of 1% - 99%, for a maximum pre-cool duration of 12 hours (full off-peak operating bracket). The scope of this analysis is to characterize the impact on the following outputs: annual electricity consumption for space cooling (AECC), annual cost for space cooling (ACC), and mid-peak/peak power consumption. For the following plots, each simulated data point represents an annual operating period for the respective hybrid shave factor with the corresponding pre-cool schedule; for which the meaning of a negative savings nullifies to a percent increase.

Figure 3a and Figure 3b present the simulated results for the variation in annual electricity consumption and annual operating cost, respectively. For the mid-rise (Figure 3a) the slope of AECC savings has a positive trajectory for small shave factor values (under 20%). In all three building cases the trajectory of the AECC savings profiles become negative for a shave factor range of approximately 20% to 60%. After transitioning into a shave factor range of approximately 60% to 99%, the slope of the AECC savings profiles becomes positive, converging at a shave factor of 99% with a predicted AECC savings percentage of zero.

It is illustrated that the proposed pre-cool control strategy has the smallest impact on system energy consumption for a shave factor range of approximately 5% to 20%; where there is a negligible increase in total energy consumption for space cooling. When considering the potential economic benefit associated to the suggested pre-cooling strategy, similar trends as those present in the AECC savings profiles (Figure 3a) occur. In Figure 3b, the scheduled ground pre-conditioning shows potential for improved system economics, occurring for undersized hybrid systems (low shave factor values). The ACC savings potential occurs for a shave factor range of approximately 5% - 40%; with the maximum potential for ACC savings occurring on the lower end of the range and gradually degrading as the upper limit is reached. For the mid-rise building a shoulder season (SS) pre-conducting schedule shows the least sensitivity to variation in hybrid system proportions.

To conceptualize the physical meaning of trends addressed in Figure 3a, the significance of the hybrid system shave factor must be understood. The shave factor parameter represent the percentage of a buildings peak cooling demand met by the GSHP, with the difference being supplied by the auxiliary cooling system. For example, if a building requires 100 kW of peak cooling and a shave factor design of 10% is selected, the resulting cooling capacity of the GSHP and auxiliary cooling system are 10 kW and 90 kW, respectively. When the shave factor is varied, three hybrid system parameters change: ground-loop length, GSHP system cooling capacity, and auxiliary system cooling capacity. From the simulation results, undersized hybrid systems are most promising for this strategy due to smaller ground-loop lengths (meaning smaller bore-field/soil volumes) and large auxiliary cooling capacity, which would translate to faster ground temperature response. This concept is further illustrated for shave factor values of 1% and 99%; 1% representing the smallest hybrid system configuration (large auxiliary cooling capacity and small ground-loop length) resulting in excessive energy consumption from pre-cooling with insufficient GSHP operating benefit, due to an undersized bore-field heat exchanger. Alternatively, a shave factor of 99% represents the largest hybrid system configuration (small auxiliary cooling capacity and large ground-loop length) resulting in negligible energy consumption increase from pre-cooling and negligible thermal impact on the bore-field, due to large bore-field/soil volumes.
The predicted annual mid-peak and peak power reduction sensitivity to hybrid system proportions are presented in Figure 4a and Figure 4b, respectively. The mid-rise building simulations depict consistent trends with regard to mid-peak/peak power reduction potential. In this case, maximum peak power reduction occurs within a shave factor range of 5% to 40%. As the hybrid system size increases past a shave factor of 40%, mid-peak/peak power reduction degrades until a negligible impact on the results, at a shave factor of approximately 60%.

Figure 3 (a) AECC and (b) ACC, savings potential versus hybrid system shave factor (α)

Figure 4 (a) Mid-peak and (b) peak, power reduction potential versus hybrid system shave factor (α)
CONCLUSION

The analyses in this paper present a preliminary evaluation demonstrating the potential benefit that an off-peak ground pre-cool control strategy has on a HGSHP system’s performance. A three-step evaluation procedure was implemented, consisting of: an eQuest building energy simulation, a computerized HGSHP design procedure (Alavy et al. 2013), and numerical simulations of a HGSHP operating over a cooling season (conducted with the aid of a newly developed numerical performance prediction model). This study was led for a mid-rise multi-residential building, located in Toronto, Canada. Two analyses were carried out, evaluating the bore-field pre-conditioning strategy’s sensitivity to various simulation parameters. First, the impact of pre-cool duration was evaluated for an optimally sized HGSHP, predicting the response of the proposed shoulder, peak, and full season schedules for varying pre-cool duration. Secondly, the impact of HGSHP proportions was studied for system sizes in the shave factor range of 1% to 99%. The following is a summary of the significant findings and conclusions drawn from the analyses conducted on the mid-rise building:

- The potential benefit proposed by the OGPC strategy is sensitive to a building’s cooling load characteristics. No economic benefit was indicated for pre-cool schedules during operating periods with greater frequency/amplitude of cooling requirements (due to increased heat rejection to the bore-field). However, positive trends were generated with regard to mid-peak/peak power reduction potential.

- A shoulder season (SS) operating schedule indicated the greatest potential for a reduction in operating cost. The shoulder season simulations suggested the least sensitivity to variations in hybrid system proportions, indicating the greatest potential for an economic benefit with a larger range of system sizes.

- Undersized hybrid systems are most promising for the integration of an OGPC strategy due to smaller ground-loop lengths (meaning smaller bore-field/soil volumes) and large auxiliary cooling capacity, translating to faster ground temperature response.

The proposed control methodology suggests an alternative perspective for the operation of HGSHP systems, targeting the improvement of system economics through a reduction in peak power consumption. The analyses performed in this paper shed light on the potential trends that would occur for the physical implementation of an OGPC control strategy. Due to the complexity of the proposed operating concept, further research and development is required to provide a detailed characterization of the benefit that an OGPC control poses for the operation of HGSHP systems.

ACKNOWLEDGMENTS

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REFERENCES


Numerical investigation on the thermo-mechanical behavior of a quadratic cross section pile heat exchanger

Maria Alberdi-Pagola, Søren Madsen, Rasmus Lund Jensen, Søren Erbs Poulsen

ABSTRACT

Pile heat exchangers are traditional foundation piles with built-in heat exchangers. As such, the footing of the building both serves as a structural component and a heating/cooling supply element. The existing geotechnical design standards do not consider the nature of thermo-active foundations and, therefore, there is a need to develop guidelines to design them properly. This paper contributes by studying the thermo-mechanical behavior of the precast piles which are 15-meter long and have a quadratic cross section and a W-shape pipe heat exchanger. This article aims to numerically assess the additional changes in the pile load transfer generated by its heating and cooling. In addressing this objective, a preliminary multi-physical finite element analysis is conducted which serves as a tool for exploring: i) the thermally induced mechanical stresses within the concrete and on the pile-soil axial and shaft resistances; ii) the maximum upward/downward displacements. A one-year time span is considered under operational and extreme thermal boundary conditions. The results show that a typical geothermal utilization of the energy foundation does not generate significant structural implications on the geotechnical capacity of a single energy pile. However, ground thermal loads need to be considered in the design phase to account for potential extreme temperature changes, which could generate thermal stresses that equalize the mechanically generated ones.

INTRODUCTION

The Danish government aims to reduce 40% the greenhouse gas emissions by 2020 relative to 1990 and to cover the total domestic energy requirements by renewable resources by 2050 (Danish Energy Agency, 2012). In this matter, ground source heat pump systems cooperate in the transition towards sustainable energy sources.

Pile heat exchangers, also known as energy piles, are thermo-active ground structures that utilize reinforced concrete foundation piles as vertical closed-loop heat exchangers, developed as an alternative to borehole heat exchangers (Brandl, 2006). As such, the foundation of the building both serves as a structural and a heating and/or cooling component and as a result, their dimensioning becomes a coupled thermo-hydro-mechanical challenge. Pile heat exchangers can vary in length from 10 to 50 meters and in width from 0.3 to 1.5 meters. Besides, the geothermal pipes can be placed in the central part or closer to the pile edge (Brandl, 2006). Due to this variety, several experimental and numerical studies attempt to develop novel approaches that...
characterize the heat transfer in and around such structures (Cecinato and Loveridge, 2015, Park et al., 2013, Loveridge and Powrie, 2013, Bandos et al., 2014).

Pile design approaches in Europe are based on the determination of the ultimate and serviceability limit states according to Eurocode 7 (DS/EN 1997-1/AC, 2010). Yet the regulations do not consider the geothermal use in the foundation design process with regards to structural requirements. Thermal piles can be subject to a net change of the temperature relative to the initial condition over time, which causes thermal stresses and head displacements. Under thermo-elastic conditions, if the pile is a free body, i.e. it has no restraints, it will expand while heating and contract during cooling to yield a thermal free strain. In reality, a pile will not expand or contract freely as it will be confined by the structure on top and the surrounding soil, at different levels of degrees of freedom (Figure 1). As a result, the measured strain change due to temperature changes will be less than the free axial thermal strain and the constrained strains will develop thermal stresses (GSHP Association, 2012).

![Figure 1](image)

Response mechanism of a pile heat exchanger to thermal loading, a) for heating and b) for cooling.

The structural implications of the thermal loading in the service life of energy piles is still uncertain (Pahud and Hubbuch, 2007). The study of the effect of the mechanical loads in the long term is still an issue for practitioners and, therefore, energy foundations carry the same difficulties exacerbated by the cyclic (seasonal) thermal load effects in the soil and pile-soil interface. It has not been investigated whether the long-term bearing capacity of thermo-active piles is affected by the thermal cycles even though no operational failures have been reported to date. To ensure that the geotechnical performance of the pile is not negatively affected, conservative safety procedures are applied: the fluid temperature in the ground loop is not allowed to go below 2°C and there is a tendency to place more energy piles than required (VDI, 2001, VDI, 2010, SIA, 2005, NHBC Foundation, 2010, GSHP Association, 2012, Loveridge, 2012, Mimouni and Lalouï, 2014).

The temperature range imposed by the geothermal exploitation of the foundations falls between 2°C to 30°C or higher and their nature depends on the needs of the building (Lalouï and Di Donna, 2013). These temperature changes can affect the stress state at the pile-soil interface and the shear strength of the soil that affects the tip resistance (Olgun et al., 2014). Recent studies on the impact of thermal loading at pile-soil interface indicate that the bearing capacity of the pile is not significantly affected (Suguang et al., 2014, Di Donna, 2014, GSHP Association, 2012, Mimouni, 2014, Olgun et al., 2014). Xiao et al. (2014) and Di Donna (2014) have analyzed monotonic temperature variations in the range from 6°C to 50°C-60°C and have concluded that higher temperatures increase the strength of the clay-concrete contact and that the sand-concrete interface is not affected by the monotonic temperature changes.
Two main full-scale performance studies of energy piles lead the state of the art in the field: the Lambeth College, London (Bourne-Webb et al., 2009) and the École Polytechnique Fédérale de Lausanne EPFL (Laloui et al., 2006, Mimouni, 2014). Both studies conclude: 1) short-term plastic response of soils has not been observed due to the geothermal use since effective stresses typically are within yield surfaces, i.e., within the thermo-elastic domain; 2) the additional stresses produced in the energy pile due to thermal loads depend on the degrees of freedom of the pile. Therefore, the pile-soil interaction under working mechanical and thermal loads provokes systems that depend on soil conditions, level of pile confinement and magnitude of thermal loads, making hard to establish general rules. Fortunately, simple descriptive mechanistic frameworks have been established from observed behaviors (Bourne-Webb et al., 2009, Amatya et al., 2012, Knellwolf et al., 2011, Laloui and Di Donna, 2011).

Numerical tools are used to analyze not just experimental conditions but also potential scenarios, supporting the understanding of the physics behind the problem and assisting the development of behavior rules. Several numerical studies explore the thermo-mechanical phenomena of energy piles in different soil conditions. Regarding load transfer mechanisms, Suryatriyastuti et al. (2012), Hassani Nezhad Gashti et al. (2014) and Laloui et al. (2006) encompass good examples validated against experimental data.

The understanding of the behavior of the thermo-active foundations is still fundamental for their optimization during the design phase and under operational conditions. This paper presents a preliminary attempt to describe the thermo-mechanical implications, additional to those due to static axial loading, disturbing the thermally active version of a single precast quadratic cross section pile under operational and extreme situations for a specific case study, described in Alberdi-Pagola et al. (2016).

This paper is organized as follows: firstly, three ground thermal demand scenarios are defined based on measured data. Secondly, a three dimensional finite element model is described where the thermal loads are used as boundary conditions. Then, the Results and Discussion section analyzes the structural implications under the different thermal circumstances on a 1-year time span and, finally, conclusions are drawn.

**ANALYZED DATA**

The equivalent energy wave technique has been developed by Abdelaziz et al. (2015) to analyze the long-term performance of ground coupled heat exchangers. It generates a realistic annual sine curve based on measured operational ground thermal loads. The ground load is defined as the power measured through the ground loop of the ground source heat pump installation, divided by the number of energy piles and the length of the piles [W/m].

![Figure 2](image.png)  
**Figure 2** a) Measured ground thermal load and its equivalent wave; b) Generated ground thermal load and its equivalent wave, valid for extreme cooling of the pile (when heat extraction) and extreme heating of the pile (when heat injection).
An equivalent wave has been generated for a year of operational data available from Rosborg Gymnasium reported in Alberdi-Pagola et al. (2016) (Figure 2a). The treated ground source heat pump installation is mainly used for heating yet some free-cooling partially recharges the ground in summer (Figure 2a). This situation imitates the current working conditions and it will be referred as “Measured thermal loads”. Two more scenarios account for extreme thermal load conditions (Figure 2b), making the heat pump work in peak capacity conditions during 8 hours per day, 5 days a week. The extreme heat extraction from the ground represents an extreme heating need of the building during winter with no recharge in summer. The extreme heat injection to the ground mimics an extreme cooling need of the building. The extreme cases equal in magnitude but own opposite sign.

**METHODOLOGY**

Three dimensional finite element modelling has been used to analyze the coupled thermo-mechanical problem of a single quadratic cross section thermal pile. The model aims to reflect the geotechnical and operational conditions at the mentioned case study.

**Finite element model characteristics**

The finite element software COMSOL Multiphysics 5.2 (COMSOL Multiphysics, 2015) has been used to calculate the subsurface temperature response and the stress and strain domains in and near the pile heat exchanger under heating and cooling loads. A linear-elastic behavior was imposed to all the model domain, based on the observed thermo-elastic structural behavior of an energy pile under normal working in-situ conditions reported in Bourne-Webb et al. (2009) and Laloui et al. (2006). The thermal interaction of the pile heat exchanger with the surrounding soil is modelled by pure conduction. The presence of groundwater flow is ignored in the calculations. The soil is assumed to be isotropic and homogeneous.

The 3D model contains two domains (Figure 3a): the soil and the concrete pile, which contains a line heat source mimicking the PE-X pipe (Figure 3c). The model dimensions were established following the suggestions by Suryatriyastuti et al. (2012): lateral extension 25B, being B = 0.3 m (pile side), and vertical height for the soil volume...
2L, being L = 15 m (pile length). A quarter of the domain has been simulated taking advantage of symmetries (Figure 3b). No interface elements between the pile and the soil have been considered, allowing a perfect contact between them. This has been considered a conservative scenario since Suryatriyastuti et al. (2012) and Hassani Nezhad Gashti et al. (2014) reported that simulated thermal stresses are larger in perfect contact. The finite element model uses a mesh with 34,505 tetrahedral elements, more refined around the pile and it gets coarser with distance (Figure 3a).

**Material properties**

Table 1 summarizes the parameters for the pile and the soil (sand) used in the model. These values are taken from performed measurements and literature. The steel reinforcement has not been considered as it means less than 5% of the total weight of the pile.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Young modulus pile</td>
<td>41,900.00 MPa</td>
</tr>
<tr>
<td>Young modulus rigid sand (Geotechdata.info, 2013)</td>
<td>30.00 MPa*</td>
</tr>
<tr>
<td>Poisson ratio pile</td>
<td>0.30</td>
</tr>
<tr>
<td>Poisson ratio soil</td>
<td>0.30</td>
</tr>
<tr>
<td>Thermal expansion coefficient pile</td>
<td>3.00E-05 1/K</td>
</tr>
<tr>
<td>Thermal expansion coefficient soil</td>
<td>1.50E-05 1/K</td>
</tr>
<tr>
<td>Density concrete</td>
<td>2370.00 kg/m³</td>
</tr>
<tr>
<td>Density soil</td>
<td>1900.00 kg/m³</td>
</tr>
<tr>
<td>Thermal conductivity concrete</td>
<td>1.80 W/m/K</td>
</tr>
<tr>
<td>Thermal conductivity soil</td>
<td>2.30 W/m/K</td>
</tr>
<tr>
<td>Volumetric heat capacity concrete</td>
<td>1.98 MJ/m³/K</td>
</tr>
<tr>
<td>Volumetric heat capacity soil</td>
<td>2.60 MJ/m³/K</td>
</tr>
</tbody>
</table>

*Considered constant with depth.

**Initial- and boundary conditions**

Rolled displacement boundary conditions fix the horizontal movement on the side borders while pinned conditions restrict both the horizontal and vertical movement on the bottom boundary of the soil domain. To account for the gravity effect and the mechanical load, a two-step stationary run has been performed. These steps provoke stresses and strains, both in the concrete and in the soil that should be added to the thermally-induced ones. The initial soil vertical effective stress is 0.25 MPa at the bottom of the pile and the horizontal stresses of the soil are neglected for the hereon analysis. The bearing capacity of the 15 meter pile is 1510 kN in compression (non factored), estimated from data from Dansk Geoteknik A/S (1973). The pile head displacement under geostatic conditions, meaning no mechanical load added, is 16 mm (1.1 mm/m of pile length).

The pile is restrained at the toe allowing the free movement during pile cooling and prohibiting the expansion while heating, mimicking an end-bearing pile. The mechanical load applied at the top of the pile is 600 kN (factoring the bearing capacity 2.5 times), which gives a ratio of 0.1 between the applied load and the compressive strength of concrete (68 MPa). The application of this load resulted in a pile top settlement of 1 mm (0.06 mm/m of pile length).

Regarding the thermal boundary conditions, an initial temperature of 10°C, similar to the observed average undisturbed ground temperature at the represented case, is assumed in the whole domain. A constant temperature boundary of 10°C is also applied to the ground level and the side boundaries of the soil block are selected as open boundaries to mimic the infinite soil (Hassani Nezhad Gashti et al., 2014).

The fluid circulating inside the pipes has been represented as a line heat source (Figure 3c) subjected to a
transient and uniform heat rate [W/m] over its length (described in section “Analyzed data”). There was no external mechanical load applied during the transient models to ease the estimation of the uncoupled thermal stresses and strains.

RESULTS AND DISCUSSION

The pile-soil system has been modelled, subjected to transient thermal loads over a year. The time span is limited to the available data. The analysis aims to quantify the maximum thermal stresses and the maximum thermal displacements generated in the pile-soil system due to the geothermal use. Therefore, two types of mechanical boundary conditions have been applied to the pile extremes, as described in Table 2. Besides, the provided naming code will assist the identification of each simulated case in the analysis hereby.

<table>
<thead>
<tr>
<th>Table 2. Applied boundary conditions and identification of simulations.</th>
</tr>
</thead>
<tbody>
<tr>
<td>Boundary conditions</td>
</tr>
<tr>
<td>---------------------</td>
</tr>
<tr>
<td>Measured thermal loads</td>
</tr>
<tr>
<td>Extreme heat extraction</td>
</tr>
<tr>
<td>Extreme heat injection</td>
</tr>
</tbody>
</table>

The axial thermal stresses generated in the center of the pile for the three scenarios are shown in Figure 4a, uncoupled from the stresses generated by the mechanical load. Table 3 summarizes the main results of the transient simulations. The contour of the profiles in Figure 4 are comparable to previous literature (Hassani Nezhad Gashiti et al., 2014), with a stabilized zone along the middle and two transient zones at the extremes indicating the uniform nature of the thermal effects.

For heat injection cases (simulations 1A in summer and 3A), the upper zones with higher compression stresses, occur due to the influence of the constant temperature boundary condition imposed at the soil surface and the restrained movement of the pile head. The second region at the bottom might be influenced by the thermal gradients generated below the end of the heat line source and the restrained toe movement. The maximum compression stress as a result of the thermal loads reaches 6.5 MPa (simulation 3A, Table 3). For this case, the magnitude of the temperature-induced load is very close to the purely mechanical load, increasing the solicitation of the toe, resulting on a combined load of 1185 kN, almost 80% of the pile capacity, corresponding to failure. This would not be allowed for design, as it is above the design bearing capacity (including safety factors) and it indicates that thermal loads need to be considered in the design of the energy foundations. For operational circumstances, on the other hand, the combined load in summer hardly increases 2%.

A temperature increment of 1°C results in an additional temperature-induced vertical force of 100 kN approximately, very similar to the values reported by Laloui et al. (2006) and Amatya et al. (2012). However, in terms of stresses, the studied pile suffers considerable increases, in the order of 1000 kPa/°C, due to the small cross section of the pile.

The cooling of the pile provokes a constrained contraction, generating a maximum tensile stress of -2.2 MPa and -0.7 MPa for simulations 2A and 1A in winter, respectively (Figure 4a). The tensile stresses dissipate with depth as the upwards movement of the pile toe is permitted.

Regarding the mechanical properties of concrete, the maximum compressive stresses developed means 10%
of the ultimate compression strength. Hence, the combined thermal and mechanical load would reach 20% of the compression resistance. The tensile strength of the concrete, without reinforcement, has been estimated as 5 MPa from Neville (1995). It is still twice higher than the computed maximum tensile stress.

The maximum shaft shear stress estimated is 160 kPa at the pile toe for the extreme heat injection case (simulation 3A) even though over the pile length the shear stresses were negligible. The constrained pile expansion at the pile toe induce an increase of shear stresses concentrated in the bottom region of the pile (Figure 4b). On the other hand, during pile cooling, the perfect contact in the interface increases the shaft stress component, creating a higher mobilized shear stress over the pile length for the extreme heat extraction (simulation 2A). As highlighted by Hassani Nezhad Gashti et al. (2014), the mobilization of the shaft loads when the bottom movement is allowed rises concern on the behavior of floating thermal piles.

Figure 4  a) Thermally induced axial stresses over the pile length at its center and the axial stresses mobilized by the static mechanical load. Negative sign states for tensile stresses. b) Mobilized thermally-induced shear stresses.

According to DS/EN 1997-1/AC (2010), the pile fails when it settles 10% of the pile base diameter (i.e., 0.023 mm/m). For the considered cases, the displacements are insignificant (Table 3). The maximum predicted deformation belongs to the combined effects of a temperature decrease that generates the pile contraction (simulation 2B) and the mechanical load, providing a head settlement of 0.17 mm/m, i.e., 7.5% of the allowed settlement. This agrees with the conclusions from Xiao et al. (2016) who reported that the cooling cycle can dominate the serviceability limit state design of pile heat exchangers. It is concluded that the deformations resulted from the geothermal exploitation are very low, probably due to the high rigidity of the pile.
Table 3. Maximum thermal stresses and displacements predicted for a one-year study.

<table>
<thead>
<tr>
<th>Measured thermal loads</th>
<th>Maximum temperature change from initial condition</th>
<th>Maximum thermal-induced axial stress</th>
<th>Maximum thermal-induced displacement of pile head relative to pile length</th>
</tr>
</thead>
<tbody>
<tr>
<td>Heat injection in summer</td>
<td>0.5°C</td>
<td>0.3 MPa [1A]</td>
<td>-0.01 mm/m [1B]</td>
</tr>
<tr>
<td>Heat extraction in winter</td>
<td>2.0°C</td>
<td>-0.7 MPa [1A]</td>
<td>0.03 mm/m [1B]</td>
</tr>
<tr>
<td>Extreme heat extraction</td>
<td>-6.0°C</td>
<td>-2.2 MPa [2A]</td>
<td>0.11 mm/m [2B]</td>
</tr>
<tr>
<td>Extreme heat injection</td>
<td>6.0°C</td>
<td>6.5 MPa [3A]</td>
<td>-0.11 mm/m [3B]</td>
</tr>
</tbody>
</table>

Sign criteria: for displacements, positive means downwards and for stresses, positive means compression.

This paper investigates the additional thermal stresses and displacements generated in a pile heat exchanger due to its geothermal use over a year. The time span was limited by the available data. This rough study indicates i) orders of magnitude of the thermally induced stresses and displacements and potential temperature changes under certain thermal conditions and ii) relevant aspects that require more refining in coming studies.

The analyzed thermal loads do not produce the thermal quasi steady state situation required to dimension the long-term performance of the ground source heat pump installations. This condition would allow to quantify the total temperature change that the pile-soil system would be subjected to in the long term. Therefore, the present study underestimates the net temperature change. i.e., for heat extraction cases, higher tensile stresses and higher pile head settlements than the simulated ones would be predicted in the long term, while compression stresses would increase when heat injection is required. On the other hand, the mechanically restrained toe and head boundary conditions and the perfect contact between pile and soil are conservative measures that give rise to overestimated stress states.

CONCLUSIONS

This paper presents a preliminary approach to quantify the additional thermal stresses and displacements generated in a pile heat exchanger due to its geothermal use. For that, measured and extreme thermal loads have been applied to a linear thermo-elastic 3D finite element model of a single precast energy pile and the surrounding soil.

Transient simulations over a year show that a typical utilization of the energy foundation does not generate significant structural implications on a single thermal pile in terms of axial and mobilized shaft stresses and generated displacements. However, for extreme heating conditions, meaning a temperature increase of 6 °C, the combined thermal and mechanical loads can reach 80% of the bearing capacity of the pile in compression, which would not be acceptable in design. Besides, the stress state conditions could worsen in the long term, highlighting the importance of proper structural analysis in the design phase of the pile heat exchangers.

Future attempts should account for more complex phenomena (pile-soil friction, thermo-mechanical constitutive laws of soils, pile group effects, etc.). Besides, the thermal influence between neighboring piles, the effect of the natural temperature variations of the soil and the impact of the building on top of the geothermal reservoir domain should be emphasized as they will affect the amount of usable thermal energy in the long-term.

ACKNOWLEDGMENTS

We kindly thank the following financial partners: Centrum Pæle A/S, INSERO Horsens and Innovationsfonden Denmark. We express our deep gratitude to Rosborg Gymnasium & HF for facilitating access to their installations, to Víctor Marcos Mesón for his advice and the two anonymous reviewers for their comments.
REFERENCES


Influence of Ground Heat Exchanger Zoning Operation on the GSHP System Long-Term Operation Performance

Mingzhi Yu
Kai Zhang

Hongmei Rang
Zhaohong Fang
Jinbao Zhao

ABSTRACT

To alleviate the ground heat accumulation after long term running of ground source heat pumps (GSHP), ground heat exchanger (GHE) zoning operation can be adopted. Two GHE operation modes—zoning operation and full running—are compared in a case where heat release to the ground in summer is larger than the heat extraction from the ground in winter. In this study, the soil thermal conductivity, volumetric specific heat capacity, borehole depth and spacing are 2.0 W/(mK)^1, 5.0×10^6 J/(m^3K)^1, 100 m and 5 m respectively with the boreholes arranged in a square 20×20 array. Under the given conditions, the simulation results show that GHE zoning operation depresses the increase in amplitude of GHE outlet water temperature and so that the GSHP systems operate normally throughout the whole service life. By adopting GHE zoning operation, the energy consumption of the GSHP system is found to be reduced compared with that of a GHE operated without zoning. Operation without zoning shows that the GHE summer outlet water temperature increases faster than that with zoning operation and power demands are increased for the given GSHP load. Furthermore, in this case, the GSHP would not be able to run normally in the last several years due to the condensing temperature exceeding its upper limit.

INTRODUCTION

In the severe cold or the relatively warm regions, the summer and winter thermal loads imposed on the ground heat exchangers (GHE) of ground source heat pumps (GSHP) are usually unbalanced, i.e. the heat that the GHE rejects to the ground in summer is unequal to that absorbed from the ground in winter. For short term operation, unbalanced GHE thermal load will not induce excessive cold or heat accumulation in the ground as the ground heat dissipation capacity. However, for long term operation, summer and winter load imbalance would cause ground temperatures to successively increase or decrease (Cao, et.al, 2012) and the GSHP system operating efficiency would correspondingly decline. In extreme conditions the GHE may not be able to satisfy the thermal load on the GSHP system when the GHE outlet temperature approaches or exceeds the condensing temperature/pressure limit.

Nowadays, the main effective methods to alleviate the ground thermal accumulation include adopting hybrid
ground coupled heat pump systems (HGCHP)(Xi, et.al, 2012; Wang, et.al, 2011; Liu, 2011; Man, et.al, 2011), enlarging the space between boreholes(Yu et.al, 2010; Kurevija et.al, 2012) and GSHP operating intermittently(Gao, et.al, 2006; Shang, et.al, 2012). The HGCHP systems are often used in cases with unbalanced thermal loads. For HGCHP systems, a cooling tower is typically adopted when the thermal load in summer is greater than that in winter and supplement heating equipment such as solar energy collector would be employed when the winter load is bigger than the summer load. Sometimes, for bigger winter load cases, some measures are used to inject heat into the ground in the other seasons so that the stored heat can be used in winter. Although the thermal accumulation can be inhibited effectively by the hybrid heat pump heating systems, the energy efficiency of supplementary loop or equipment is usually lower than that of GSHPs. Moreover, the operation and adjustment of a hybrid system is much more complex than that of a pure GSHP. In fact, enlarging the space between boreholes increases the ground volume occupied by the GHE and the ground temperature variation will maintain relatively even due to the increase of the total heat capacity of the GHE field. Enlarging borehole-spacing is a simple and practicable method, but it need more ground surface and is restricted in the places where is short of land. Intermittent operation provides time for the ground thermal recovery, but the power consumption of the compressor is increased. Moreover, the operation mode of a HVAC system is usually set according to the building demand, and hardly guarantee the operation time needed for ground thermal recovery.

This paper intends to use the GHE zoning operation method (Yu, et.al, 2016) to solve the problem of ground thermal accumulation.

**ZONING OPERATION STRATEGY**

During the long-term operation of the GSHP systems, the heat/cold accumulated in the central GHE field induced by the unbalanced seasonal thermal load hardly transfers to the field outside the GHE area even after many years. For the case with heat injection in summer is greater than heat extraction in winter, it is obvious that the thermal anomalies would be gradually aggravated with time extension if all buried pipes of the GHEs are utilized in winter and summer. Therefore with running time growth, the heat in the central GHE part would continue to accumulate and the soil temperature gradually increase, the heat exchange capacity of buried pipes in summer gradually decrease, and even cause the heat exchange of some buried pipe failures. For this case the zoning operation mode can be used, in which the whole GHE operates in summer and only the relatively central part of the GHE works in winter. For the systems only extract heat from the relatively central GHE area, the workload imbalance of the relatively central part of the GHE between summer and winter will drop significantly. Although this mode would make the soil temperature in the central region obviously lower than that with no zoning operation, which is not conducive to draw heat from the soil in winter but is helpful to release heat into the ground in summer. The zoning operation method can help eliminate underground heat accumulation and consequently, it is possible to improve the annual energy efficiency of the heat pump units and the long-term operational efficiency of the system. Furthermore, water pump power demands will be reduced as overall flow rates are lower during zoning operation.

**ANALYSIS AND DISCUSSION**

**Mathematic Model of Heat Transfer between GHE and Ground**

Assuming that the ground is a semi infinite media with uniform initial temperature and constant thermal properties, and the GHE is regarded as a group of finite line heat source, hence the finite line heat source model can be used to describe the heat transfer between the GHE and the ground. The soil temperature around a borehole can be obtained by the following equation (Zeng, et.al, 2002; Louis, et.al, 2007),

\[ \text{Soil Temperature} = \text{Heat Source Model} \]
\[ T = T_0 + \frac{q}{4\pi \lambda} \int_0^\infty \left[ \frac{\text{erfc}(r^2 + (z-h)^2 / 2\sqrt{at})}{\sqrt{r^2 + (z-h)^2}} \right] dh \]

Where, \( q \) is the heat release of a borehole per unit depth, W/m; \( \lambda \) is the ground thermal conductivity, W/m\( \cdot \)K; \( a \) is the ground thermal diffusivity, m\(^2\)/s; \( r \) is the distance from the center of the buried pipe, m; \( z \) is the axial coordinate of the pipe, m; and \( H \) is the borehole depth, m.

The temperature change of any point in the GHE area is the superposition of temperature rises induced by each borehole according to the superposition principle. For cases with inconstant load, the excess temperature can be obtained by considering the varying load as the integration of a series of step loads (Fang, et.al, 2002),

\[ \theta(x, y, z) = \sum_{i=1}^{n} \sum_{j=1}^{m} \frac{q_{i,j} - q_{i,j-1}}{4\pi \lambda} \int_0^\infty \left[ \frac{\text{erfc}(r_{i,j}^2 + (z-h)^2)}{2\sqrt{a(t_n - t_{j-1})}} \right] dh \]

Where, \( t \) is time, s, and

\[ r_{i,j} = \sqrt{(x-x_i)^2 + (y-y_i)^2} \]

Here, \((x_i, y_i)\) is the coordinate of the \( i \)th borehole.

**Model of the Heat Pump Unit**

A chiller model in the simulation program DOE-2 is used to analyse the heat pump unit (Hydemam, et.al, 2002). To simplify calculation, the circulating water flow rate is assumed constant. The power of the heat pump unit can be calculated by Equation 4.

\[ P = C_1 C_2 P_r \]

Where, \( P_r \) is the input power that the heat pump unit needed at the rated state, KW; \( C_1 \), the input power correction coefficient; \( C_2 \) the input power correction coefficient of the heat pump unit at part load.

\[ C_1 = \alpha_1 + \alpha_2 T_{eo} + \alpha_3 T_{eo}^2 + \alpha_4 T_{ci} + \alpha_5 T_{ci}^2 + \alpha_6 T_{eo} T_{ci} \]

\[ C_2 = \beta_1 + \beta_2 PLR + \beta_3 PLR^2 \]

Here, \( \alpha \) and \( \beta \) are regression coefficients; \( PLR \), the part load rate of the heat pump unit; \( T_{eo} \) the outlet temperature of the evaporator, °C ; and \( T_{ci} \) the inlet temperature of the condenser, °C.

**Table 1. Main Rated Parameters of the Heat Pump of EXLSR530.1**

<table>
<thead>
<tr>
<th>Cooling Capacity / kJ/h</th>
<th>COP</th>
<th>Flow Rate at the Chilled Water Side of the Evaporator / kg/h</th>
<th>Flow Rate at the Chilled Water Side of the Condenser / kg/h</th>
<th>Outlet Water Temperature of the Evaporator / °C</th>
<th>Return Water Temperature of the Condenser / °C</th>
</tr>
</thead>
<tbody>
<tr>
<td>1576800</td>
<td>5</td>
<td>75265</td>
<td>88738</td>
<td>7</td>
<td>26.67</td>
</tr>
</tbody>
</table>

The heat pump unit parameters used are according to the heat pump unit of EXLSR530.1 (A product of Neimenggu Yike Air Conditioning Equipment Limited Company). The main rated parameters are listed in Table 1.
Coefficients of Equation 5 and 6 are obtained by nonlinear regression and then Equations (5) and (6) can be rewritten as,

\[ C_1 = 0.7360 + 0.0189 T_{eo} - 0.0015 T_{eo}^2 + 0.2167 T_{ci} + 0.2055 T_{ci}^2 \]  

\[ C_2 = 0.2726 - 0.08413 PLR + 0.8102 PLR^2 \]

**Model of the Pump**

The effective power of the pump is,

\[ N_e = \gamma QH / 1000 \]  

Where, \( N_e \) is the energy water gained by the pump, KW; \( \gamma \), the water volume weight, N/m\(^3\); \( Q \), water flow rate, m\(^3\)/s and \( H \) the pump delivery head, m.

The shaft power of the pump is

\[ N = \frac{N_e}{\eta} = \frac{\gamma QH}{1000\eta} \]

Here, \( \eta \) is the total efficiency of the pump, which is the product of volumetric efficiency, hydraulic efficiency and mechanic efficiency.

**Case Study**

A 20 × 20 matrix arrangement multi-borehole ground source heat pump system is adopted to analyze the zoning operation conditions which heating load in summer is significantly greater than the cooling load in winter. To cut down the calculation workload and simplify analysis, the GSHP system is assumed keeping on running and maintains constant all the time during summers and winters. The GHE heat injection in summer is 666 KW and heat extraction in winter is 411 KW, the summer/winter load ratio is 1.6:1. All buried pipes of the GHE are utilized in summer and the outermost two rows of pipes stop running in winter, as shown in Figure 1. The reason to stop the outermost two rows of pipes running in winter is that, for running boreholes, the heat injection in summer approximately equals to the heat extraction in winter, which means loads maintain balanced in summer and winter. As a result, the thermal accumulation in the ground will be less serious than that of all boreholes running in both summer and winter. The main parameters of the boreholes and the ground are presented in Table 2. A buried pipe is assumed failure when its outlet water temperature reaches 40°C. The load that formerly undertook by the failure pipes are then shared by other pipes which outlet water temperature less than 40°C.

<table>
<thead>
<tr>
<th>Table 2. Main Parameters of Borehole and Soil Thermal Properties</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Borehole Spacing /m</strong></td>
</tr>
<tr>
<td>5</td>
</tr>
</tbody>
</table>
Figure 1  GHE zoning operation mode

Figure 2 is the monthly average outlet temperature of the GHE in the air-conditioning season which is running normally. The running time are 3 months in summer and winter respectively every year and the GSHP system stop running in the other 6 months, so the total running time in summer or winter are both 60 months when the ground source heat pump system was operating normally 20 years.

Figure 2  GHE outlet temperature vs. time
For the two kinds of running modes have the same initial conditions in the first summer, so the summer average GHE outlet water temperature is 20.27°C, and the water temperature rises to 21.70°C at the end of this summer for both running modes. At the beginning of the next summer, the outlet water temperature of the two modes are all decreased due to the heat extraction in winter and thermal recovery in spring. From the second year, the outlet water temperature at zoning running mode is always lower than that at the full running mode, which enables the GSHP system running in a relative high efficiency way. With running time increase, the outlet water temperature of the zoning running becomes higher and higher because heat continues to be accumulated in the relative central area. As mentioned above, a buried pipe is considered failure when its outlet water temperature reaches 40°C. After 15 years, i.e. 45 summer months, some buried pipes at the full running mode become failure for their outlet temperature exceed 40°C and cannot release heat normally in summer, therefore the outlet water temperature increases more rapidly for other pipes should bear more loads from the 15th year. The outlet water temperature of the whole buried pipe in the case of full running mode are all over 40°C in the 17th year, which leads to the GSHP system cannot running normally anymore. At the same time, the outlet water temperature of every month in air-conditioning season is 30.89°C, 31.71°C and 32.39°C respectively in the zoning operation case, which is far below than that at the full running mode. The above results show that the zoning operation mode can effectively reduce the thermal accumulation in the GHE field, which will guarantee the GSHP system runs normally in 20 years and even longer.

The GHE outlet water temperature in winter decreases month by month with both operation modes. The outlet water temperature at zoning running mode maintains lower than that at the full running mode from the first winter, and the temperature difference increases with operation time increases. Although the zoning operation reduces the outlet water temperature in winter which result in lower winter operation efficiency, the global annual efficiency is likely higher than that of full running operation due to bigger GHE outlet temperature in summer. As shown in Figure 2, the GSHP system operates normally in 20 years at zoning operation mode, while it can only work 15 years at full running mode.

Figure 3 is the heat pump unit power needed in summers and winters. During the summer operation in the first year, the two operation modes have the same GHE outlet temperature and the heat pump unit requires same power (104.5kW). At zoning operation mode which only put the inner 18×18 of buried pipes into use in winter, the GHE outlet temperature is lower than that at full running mode, and in the first winter the power needed of zoning running is 88.8kW which is higher than that at full operation mode. From the second year, the two running modes are affected by thermal accumulation in the last summer and the outlet temperature and power needed are all increased. The power increase amplitude at zoning operation mode is lower than that at the full operation mode, therefore
zoning running saves energy. Meanwhile, in winter, GHE outlet temperature at both modes increase and power needed decrease due to ground thermal accumulation. Power needed at zoning operation mode reduces less, which results in higher power needed comparing with that at full running mode. However, Figure 3 shows that, at full running mode, the power cut down in summers is much more than the power more consumed in winters. Therefore, comprehensively, the zoning running mode is more energy-saving comparing with the full running mode.

**CONCLUSION**

This paper has analyzed the GHE outlet temperature and the heat pump unit power needed at zoning operation mode and full running mode by the case of the building load in summer is greater than that in winter, the results show that:

Zoning operation can significantly depress the GHE outlet water temperature increase due to ground thermal accumulation caused by unbalanced thermal load in winter and summer. This apparently improves the GSHP system operation efficiency.

Although power needed at zoning operation mode is higher than that at the full running mode in winter, it is far lower than that at full running mode in summer. Comprehensively, comparing with the full running mode, the zoning operation mode is energy-saving.

**ACKNOWLEDGMENTS**

This work is supported by the Natural Science Foundation of China (NSFC) No.51176104 and the State's Key Project of Research and Development Plan No. 2016YFC0700803-01.

**NOMENCLATURE**

- \(a\) = Soil thermal diffusivity (m²/s)
- \(\lambda\) = Soil thermal conductivity (W/mK)
- \(H\) = Borehole depth (m)
- \(q\) = Heat release of a borehole per unit depth (W/m)
- \(r\) = Radial distance from the borehole axis (m)
- \(\theta\) = Excess temperature (°C)
- \(T\) = Temperature (°C)
- \(T_o\) = Initial temperature of the ground (°C)
- \(t\) = Time (s)
- \(P\) = Power (KW)
- \(C_1\) = Input power correction coefficient
- \(C_2\) = Input power correction coefficient of the heat pump unit at part load
- \(P_r\) = Water volume weight (N/m³)
- \(PLR\) = Part load rate
- \(\alpha\) = Regression coefficient
- \(\beta\) = Regression coefficient
- \(\gamma\) = Water volume weight (N/m³)

**Subscripts**

- \(eo\) = outlet of the evaporator
- \(ci\) = inlet of the condenser
REFERENCES


ABSTRACT

Pile foundations provide a low-intrusive mechanism for installing heat exchangers into the ground, especially if the piles are being installed to support the building. The ground temperature is relatively constant at depths >10 m, providing consistent performance of the heat exchanger to the operator. However, as these piles are also being used to support the building above, a greater understanding of clay’s response to temperature must be further identified to allow confidence in their installation. A review of case studies, field tests and laboratory tests illustrates the need for further work in this area. This paper reports initial results from a series of specialized tests on two types of clay subjected to five thermal cycles, representing the long-term operation of pile foundation heat exchangers, and shows that distinct shrink/swell cycles occur leading to ~5% reduction in volume over the five cycles for a kaolin clay and 0.5% for a glacial till.

INTRODUCTION

Pile foundation heat exchangers (PFHX) are a fast growing technology across the world, providing low carbon heating and cooling across the non-domestic sector. As non-domestic building emissions targets are tightened PFHX can help to significantly reduce the carbon emissions associated with air-conditioning in large commercial and public buildings. At present, only 1% of the current UK heating demand is met by renewable sources. The UK Government predicts that the decarbonisation of the heating sector will be required in order to meet their 2020 and 2050 greenhouse gas targets. By 2020 it is expected that around 40% of UK commercial floor spaces will be air-conditioned, and for the UK climate this is predicted to be a 60/40 cooling split over a year causing seasonal temperature variation in the ground surrounding a pile foundation heat exchanger (PFHX) (Evidence Directorate, 2009). Present levels of uptake are hindered by the uncertainty surrounding their effect on ground strength and pile performance.

PFHX impose a cyclic hot and cold temperature profile onto the ground surrounding the pile in order to meet the heating and cooling demands of a building, as illustrated by Figure 1. PFHX utilise the ground surrounding the pile to source heat during the winter months and sink heat during the summer months. Busby et al. (2009) found that the variation in the near surface ground temperature throughout the year at a depth exceeding 10 m showed less than a 5°C variation in ground temperature. This makes pile foundations ideal to use as heat exchangers due to the more predictable temperature profile at greater depths compared to ground heat exchangers installed closer to the ground. The effect may also allow the ground to be used to store heat between seasons in regions of low groundwater flow.
CURRENT LITERATURE

Current literature presents very little information on the influence of cyclic temperature variation in PFHX with respect to the structural performance of the in-situ clay in which the piles are installed. Literature is split between case studies of in-situ PFHX and laboratory testing on discrete clay samples.

Case studies

In the current literature there has been a distinct focus on the performance factors of ground source heating and cooling through energy piles and the structural performance of the concrete piles during the heating. Studies tend to focus on a specific clay type, or a specific set of conditions. This study aims to understand the performance of the clay in which PFHX are installed, therefore those case studies that provide potential evidence of the clay’s response to thermal cycles are reviewed here.

Bourne-Webb et al. (2011) discussed the hypothesis that a thermal load on an in-situ pile foundation creates expansion and contraction within the pile itself causing axial forces to develop. During a field test on a PFHX, an increase in shaft resistance was observed, which increased by 25 to 75 kPa along the length of the pile during the heating phase, representing an increase in shaft resistance of three times that without heating. This caused a general reduction in axial load along the pile, and the axial load was also observed to become tensile in the lower third of the pile. This theory and framework was applied to the authors’ case studies at Lambeth College and that of Laloui et al. (2006) at École Polytechnique Fédérale de Lausanne (EPFL). It was found that the hypothesis proposed by Bourne-Webb et al. (2011) supported the field results from the Lambeth College and EPFL case studies, giving evidence that cyclic temperature variation does impact on the skin friction of a pile foundation. However, this study was primarily focused on the effects arising due to thermal expansion of the pile itself.

There is conflicting evidence from different case studies about the mobilisation of skin friction due to thermal cycles. Another case study by Brandl (2006) found during efficient PFHX operation, the shaft resistance was not affected by temperature. This was thought to be due to the pile base load remaining constant, independent of the total load to the pile and temperature variation in and around the pile, however no further explanation was provided. Proper operation of a PFHX is defined by Brandl (2006) as reducing a local ground temperature of 10-15 °C to approximately 5-10 °C, preventing the formation of ice lenses.

Bourne Webb et al. (2009) and Laloui et al. (2006) also observed pile head displacements of 5 mm during their in-situ short term cyclic temperature testing. This provides clear evidence that cyclic temperature variation has the potential
to affect the settlement and serviceability limit of pile foundations. Particularly concerning from this result is the likelihood of differential settlement across a building’s foundation for the case where not all the piles are used as PFHX. The propensity for settlement of the few piles undergoing heating and cooling may unequally share load onto the static piles and risk damaging the building as a result.

**Laboratory testing**

Current literature on laboratory testing provides a greater insight to clay response undergoing thermal cycling than that discussed in the case studies. However, the major focus until recent times has been the increase in temperature of clays to a greater magnitude than that used in PFHX (as the application under investigation has been the storage of nuclear waste). Recent case studies have provided little insight on the influence of cyclic temperature variation on the ground surrounding a pile foundation, largely focusing on individual clay types and temperature profiles as found in PFHX case studies. Design guides (Ground Source Heat Pump Association 2012) take into consideration only the temperature effect of PFHX rather than the long term effect of heating and cooling of the ground surrounding the pile as the building is heated and cooled. A brief overview of recent laboratory studies is presented below.

**Engineering properties**

In early studies, the response of clay to temperature was considered with regard to the shear strength (Mitchell, 1993) volume change and pore pressure variation (Campanella & Mitchell, 1968). A recent study by Abuel-Naga et al. (2008) has considered the impact of temperature on a range of general engineering properties for geosynthetic bentonite clay layers. These include permeability, compressibility and shear strength through a study on a soft low plasticity clay. Other studies have addressed individual engineering properties including the effect on overconsolidation ratio (Cekerevac & Laloui, 2004) and plasticity (Towhata et al., 1993). However, these studies have been carried out for a variety of individual clay types and discrete temperature increases. Results from these studies have yet to be unified into a single framework for pile design.

Although not evidenced, Campanella & Mitchell (1968), Abuel-Naga et al. (2008) and Cekerevac & Laloui (2004) propose that a change in sample temperature produces a change in the fabric of the clay with interparticle forces and the viscous shear resistance of the absorbed water being affected. The understanding of these changes will significantly aid the future interpretation of thermal cycles.

With regard to the effect of temperature on the stiffness and shear strength of clay, Abuel-Naga et al. (2008) identified that higher temperatures increased the shear strength and stiffness of Bangkok clay, through both drained and undrained testing. However, evidence of this in current literature relating to other clay types has not been identified.

**Volume change**

The general effects of temperature, reported by Abuel-Naga et al. (2008) and Cekerevac & Laloui (2004), on the volume change and compressibility of soil which have been identified are:

- A decrease in soil volume with increase in temperature for normally consolidated and lightly overconsolidated clay, provided temperature is the only variable in testing
- An increase in soil volume with increase in temperature for heavily overconsolidated clay followed by a decrease, provided temperature is the only variable in testing
- Any temperature increase causes an immediate volume change with the magnitude of change dependent on the magnitude of the temperature change
- An increase in compressibility with an increase in temperature

Both authors carried out testing on clay samples of varying overconsolidation ratio and both concluded that the
stress history of a sample influences the degree of thermal volumetric expansion or contraction produced through heating. Campanella & Mitchell (1968) first introduced the idea of volume change, identifying that a reduction in sample height occurred during drained heating of a normally consolidated Boom clay sample. What remains unknown from the investigation is whether this change in sample height was attributed to a change in diameter of the sample (i.e. barreling) or isotropic contraction. Three temperature cycles were carried out on the sample with a temperature change from 4°C to 60°C. Following the heating and cooling cycles, only 47% of the water was reabsorbed back into the sample following the 3 cycles, indicating an irreversible change occurred within the sample. In a second drained test on Boom clay, the sample height was measured through the same 3 temperature cycles. During this test, the original sample height of 89mm increased by 3.4mm during heating phases. It is possible that radial expansion of the sample may occur due to the isotropic expansion of the sample during heating followed by anisotropic contraction of the sample during cooling, with axial contraction dominating.

Towhata et al. (1993) conducted tests on kaolin and bentonite samples within a temperature controlled oedometer. Samples were incrementally heated whilst their void ratio was monitored. Testing showed that during heating a reduction in void ratio of 3% for kaolin and 12% for bentonite was observed. As testing was only for heating, the effect of cyclic void ratio change during the cooling stages could not be determined. It is possible that during the temperature cycles, an increase in pore water pressure within the clay leads to the production of a pseudo-consolidation effect. Similarities are thought to exist between temperature induced and pressure induced consolidation effects.

Drainage

It could be assumed that the clay in which the PFHX is situated will function in an undrained condition during a change in temperature, assuming low permeability clayey ground conditions. This assumption may be too simplistic given the longevity of PFHX. For the case of using a PFHX to heat and cool a building, the temperature cycles being input to the ground have a 12 month period which will allow sufficient time for a proportion of the excess pore water pressure arising from the thermal expansion of water to drain. To achieve this, water must either drain radially from the pile into the surrounding ground or along the pile/soil interface: potentially causing a weakened interface and reducing the shaft resistance of the foundation. Alternatively the drainage may cause clay consolidation, increasing the clay strength immediately surrounding the pile and increasing the foundation strength. This is an area yet to be fully understood.

Overview

Bringing together the laboratory studies identified, the application of temperature provides a similar response in the clay to a consolidation effect for normally consolidationed samples. In normally consolidated samples, under constant effective pressure, an increase in temperature leads to a reduction in volume as thermal expansion of the water leads to compression the clay and a pressure gradient and flow of water from the sample. Over consolidated samples have been shown to expand initially during heating before following the same trend as normally consolidated samples. The initial expansion may be due to the increase in temperature providing a thermal expansion that is not initially offset by compression of the (stiffer) clay and drainage. However once conditions bring the sample back to normally consolidated behaviour, the sample’s volume reduces. The effect of repeated temperatures cycles is yet to be fully understood with regards to how this would influence the structural performance of a PFHX.

UNDERSTANDING CYCLIC EFFECTS
To further the understanding of the long-term influence of PFHX operation on clays, a series of clay tests has been carried out using a modified triaxial system similar to that developed by Laloui et al. (2006). The system developed at the University of Sheffield incorporates thermocouples alongside a copper heating coil in order to heat and monitor the sample during heating and cooling cycles (Figure 2). This system simulates infield conditions through simultaneous control of the stress and temperature of the sample in the cell.

![Figure 2 University of Sheffield triaxial system with temperature regulating modifications](image)

An expansion ring at the base of the cell was commissioned for the installation of thermocouples within the cell. These thermocouples are used in the calibration of the test system and monitoring of the temperatures during testing. To regulate the temperature of the cell a bespoke heating system was manufactured. This included a regulated heated water bath and central heating pump with a maximum pressure of 6 bar and a maximum flow rate of 3.5m³/h, in order to circulate water at a high enough rate to enable efficient heating within the cell. Staged heating and cooling was carried out using a program developed in National Instruments Labview. This operates as a thermostat, with the desired temperatures being input.

In order to further identify the changes that occur during thermal cycles, local linear variable differential transformers (LVDT) were installed on the outside of the sample. The pressure and volumes within the cell, sample and loading ram were controlled using GDS Instruments pressure transducers. These allowed for an automatic maintenance of pressures independent of volume change.

**Test sample**

This study has tested two types of clay of varying plasticity (Table 1), kaolin clay and a glacial till. The samples were produced by consolidating clay slurry (mixed at a ratio by mass of 1:1 with water for kaolin and 1:0.5 for glacial till, approximately twice the liquid limit of the clay) to an effective stress of 500kPa within a 250mm diameter Rowe Cell, maintained at 26 °C. From the large Rowe Cell sample, samples of 50mm diameter and 100mm in height were cut for testing. By consolidating samples to 500kPa, the samples are significantly stiff, allowing for minimal impact of the local LVDT; on the sample deformation whilst setting up the apparatus.
Table 1 Atterburg limits and plasticity index for (a) kaolin (as tested by author) (b) glacial till (Glendinning & Hughes, 2014)

<table>
<thead>
<tr>
<th>Material</th>
<th>Liquid Limit (%)</th>
<th>Plastic Limit (%)</th>
<th>Plasticity Index</th>
</tr>
</thead>
<tbody>
<tr>
<td>Kaolin</td>
<td>63</td>
<td>32.3</td>
<td>30.7</td>
</tr>
<tr>
<td>Glacial till</td>
<td>45</td>
<td>24</td>
<td>21</td>
</tr>
</tbody>
</table>

Following saturation samples were isotropically consolidated to an effective stress of 500kPa within the triaxial system, at 26 °C (which was the laboratory temperature). The samples should therefore be normally consolidated. During the heating phase the sample was allowed to drain, in order to identify the impact of heating on the volume change of the sample through the amount of water expelled. The temperature profile is applied to the sample by heating the water bath from 27 °C to 80 °C over a period of 6 hours. This resulted in the clay temperature increasing from ~26 °C to 52 °C (Figure 3). The sample was then allowed to naturally cool over a period of 6 hours due to the ambient laboratory temperature to 32 °C before the temperature cycle began again. Subsequent cycles ran from 32 °C to 52 °C. The sample was allowed to drain freely during these cycles to allow for excess pore water pressure to dissipate. It is recognized that the temperatures profile used within this study is beyond that of normal PFHX operation (typically a maximum temperature of approximately 35 °C), however the greater temperature profile allows for a more complete understanding of the impact of temperature on clay.

Figure 3 Temperature profile with volume of water drained from the sample

Results

Figure 4 shows the relative variation in water volume within the samples over the course of the heating and cooling cycles. The initial volume of water within the sample was determined using the samples’ moisture content at triaxial set up, with the net volume or ‘back volume’ measured by the GDS following saturation and consolidation to an
effective stress of 500kPa. This back-volume figure was measured at a constant temperature at the GDS. The differing y-axis scale between the two graphs highlights the difference in the magnitude of water drained from the sample during the cycles.

Both samples showed a permanent loss of water over each thermal cycle, with the initial temperature increase producing the greatest volume change. The drainage and reduction in volume provides further evidence to that discussed previously that temperature induces thermal consolidation effects, mimicking that of stress induced consolidation. The glacial till, with a lower plasticity, appears to be less affected by the temperature variation, displacing only 0.5% of its initial volume of water over 5 cycles in comparison to kaolin displacing approximately 4.5% of its initial volume of water.

The kaolin appears to complete its drainage at the end of each cycle indicating that the permeability is high enough to allow this. However for the testing carried out on glacial till, it can be seen that water continues to drain from the sample during the first cycle of cooling. Although not determined for this specific sample, this effect is possibly due to a lower permeability/coefficient of consolidation in the glacial till, leading to the thermal cycling outpacing the drainage.

Over the course of 5 cycles, the net volume of water drained from both samples decreases, with the volume of water drained tending to reduce cycle after cycle. Although not evident from only 5 cycles, it could be thought that over more cycles this difference would continue to reduce, tending to zero additional water being drained during subsequent cycles. Future testing is currently being conducted to identify the change in volume drained over 10 cycles.

**DISCUSSION**

This study has found through testing of two different plasticity clays in a modified thermal triaxial cell, that thermal cycles produce a consolidation effect. Such effects need to be extrapolated to the timescales and dimensions of a PFHX system. Considering the core behavior identified with respect to the effect on a PFHX, further consolidation of the clay surrounding the pile could provide additional strength. However this increase in strength could be compromised by the volume reduction in the clay effectively shrinking away from the foundation, decreasing the shaft resistance whilst increasing the axial load, and also potentially reducing thermal contact between pile and ground, as well as the effect of a preferential drainage path at the pile/soil interface. In addition the volume change has the potential to cause settlement of the ground surrounding the pile and affect the stability of the whole foundation.
Further testing on a greater number of thermal cycles will provide an insight to these long-term effects.

CONCLUSIONS

1. The results of cyclic heating/cooling tests on two different plasticity, normally consolidated clays have been reported. 5 cycles of heating/cooling led to a ~5% volume change in kaolin and a ~0.5% volume change in a glacial till.
2. The heating cycles generally lead to contraction while the cooling leads to expansion, however the volume is not fully recovered on the cooling phase, leading to an overall ‘ratcheting’ of volume decrease.
3. It is tentatively postulated that continued cycles will lead to a steady state where expansion/contraction balances. However the overall contraction is likely to lead to settlement around the pile at least during initial operation of the thermal pile.

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Optimal control and operation of a GSHP system for heating and cooling in an office building

ABSTRACT

Ground source heat pump (GSHP) systems are widely considered as being among the most efficient heating and cooling renewable technologies currently available. Nevertheless, both an optimal design of components and an optimal operation of the system as a whole become crucial in order to contribute to an attenuation of the global warming potential of this kind of installations.

The overall objective of this research work is to perform the control and energy optimization of an experimental GSHP system installed at the Universitat Politècnica de València, located in València (Spain), making the control system adaptive to the thermal demand of the building and to the climate conditions.

The optimization of any system requires a comprehensive study of its behaviour, by means of a thorough analysis of all the variables and parameters implied on its performance. Therefore, as a first step, both the short-term and long-term performance based on the experimental data collected at the installation will be analyzed. Finally, the different control strategies that have been developed, implemented and experimentally tested in the system will be described. These strategies’ focus is not in optimizing the performance of each individual component, but in optimizing the energy performance of the system working as a whole, while keeping the user comfort. Experimental results for a one-year operation period demonstrate important energy savings, up to 35% in the summer season and 53% in the winter season.

INTRODUCTION

The use of ground source heat pump (GSHP) systems can lead up to a 40% energy savings for a one-year operation period compared to conventional air-to-water heat pumps (Urchueguía, et al. 2008). However, when considering the system as a whole including the auxiliaries’ consumption and not only the heat pump, an integrated optimal operation is required. In this context, the adaptation of the heat pump capacity and the auxiliaries’ operation to the thermal energy demand of the building is crucial. Some examples focused on capacity control by means of variable speed compressors were presented by Fahlén and Karlsson, (2003; 2005). In the case of GSHP systems, Edwards and Finn (2015) developed a control strategy and implemented it in a simulation model to predict optimal GSHP water flow rates for single speed and tandem speed heat pumps under part load operation. Del Col, et al. (2014) analyzed the energy performance of an existing GSHP installation with variable speed compressor, variable speed water circulation pumps and variable speed fans in the coils. Marmaras, et al. (2016) developed a control algorithm to optimize the water
circulation pumps’ operation in a de-coupled primary-secondary existing GSHP system. Montagud, et al. (2014) developed an experimental in situ optimization methodology for the water circulation pumps frequency of single stage ON/OFF GSHP systems, which was also adapted to multi-stage (Cervera-Vázquez, et al. 2015a) systems. Results led to the development of an energy optimization algorithm combined with an outdoor temperature reset strategy. Its implementation in the control board of the system allowed keeping both the setpoint of the heat pump and the flow rates of the water circulation pumps at an optimal combination (Cervera-Vázquez, et al. 2014). After running the system during a whole year, experimental results were analyzed and the optimized control was compared to a standard one. Energy savings around 30% were obtained depending on the analyzed period (Cervera-Vázquez, et al. (2014; 2015a)). However, after several years of operation, some complaints from the users were found. After a careful analysis of the experimental data, it was detected that the user comfort was not met in extreme summer conditions, being the values of indoor temperature and relative humidity 27°C and 70% respectively. To overcome this issue, a new upgraded optimized control was developed to keep the system working at its optimal point and also meeting the user comfort. This new algorithm was implemented in the control board of the system and tested during one whole cooling season (May - October 2015) and one heating season (November 2015 - February 2016). This research work describes the upgrade of the former integrated optimization strategy and presents the results for both heating and cooling mode. A previous analysis was done during the cooling season (Cervera-Vázquez, et al. 2015b).

GEOTHERMAL EXPERIMENTAL PLANT

The geothermal installation studied in this paper was built in year 2005 in the framework a European project called GEOCOOL (Geothermal Heat Pump for Cooling and Heating along European Areas, contract number NNE5-2001-00847). Since then, the installation has been completely monitored. In the framework of another European project, GROUND-MED (Advanced ground source heat pump systems for heating and cooling in Mediterranean climate, contract number TREN/FP7EN/218895), different energy optimization studies were carried out during years 2009-2013. The experimental plant air-conditions a set of spaces in the Department of Applied Thermodynamics at the Universitat Politècnica de València, Spain, covering a total floor area of approximately 250 m², and includes a corridor, nine offices (located on the east façade of the building), a computer room and a service room with office equipment and other internal loads. Each office, along with the service room, is equipped with one fan coil, except for the computer room which has two fan coils installed. The corridor is not air conditioned. Each fan coil can be individually regulated by means of a thermostat; and comfort temperature and fan speed can be manually selected by the user. The control for each fan coil is governed by a three-way valve that allows the heating/cooling water to be modulated through the fan coil. The valve is controlled by the thermostat of the room. Each room has a fan coil unit supplied by the GSHP system. The GSHP system mainly consists of a reversible water-to-water GSHP, a ground source heat exchanger (GSHX) and two hydraulic loops (one for the internal loop coupled to the building, and another one for the external loop coupled to the ground). A diagram of the installation can be found in (Cervera-Vázquez, et al. (2015b)) . The GSHP currently installed, is a prototype unit developed and manufactured in the framework of the GROUND-MED project. It consists of a water-to-water reversible GSHP, which uses R410A as refrigerant. The heat pump is reversible both on the refrigerant and on the secondary fluid circuits thanks to the use of water reversing valves, in such a way that it always works in counter current conditions resulting in a higher efficiency. The nominal heating and cooling capacities are 18 kW (35°C building return/17°C GSHX return) and 15.4 kW (14°C building return/25°C GSHX return) respectively. In order to better adapt the GSHP capacity to the building thermal load, the GSHP has two scroll compressors of the same capacity working in tandem. The evaporator and condenser are two brazed plate heat exchangers of the same geometry with 42 plates each. The GSHX consists of 6 vertical borehole heat exchangers (grid of 2x3 boreholes, 50 m deep, 3 m separated from each other); each borehole contains a single polyethylene U-tube of 25.4 mm internal diameter, with a 70 mm separation between the upward and downward tubes.

The system is divided into two hydraulic circuits. The internal one consists of 12 parallel-connected fan coils, an internal hydraulic loop and a water storage tank. The external one consists of the GSHX which is coupled to the heat
pump by an external hydraulic loop. The system operates from 7 a.m. to 8 p.m. during 5 days per week. Both circuits have circulation pumps, in order to pump the water to the fan coil units (internal circulation pump (ICP); nominal values at 50 Hz: 3180 kg/h, 0.63 kW) and the GSHX (external circulation pump (ECP); nominal values at 50 Hz: 2650 kg/h, 0.36 kW). The ICP operates continuously along the day, whereas the ECP only works when at least one of the compressors of the GSHP is running. The water flow rate in both circuits can be varied by two frequency inverters. A network of sensors was set up so that the installation is completely monitored: water temperatures (four-wire PT100 with accuracy ±0.1°C), mass flow rates (Danfoss flow meter model massflo MASS 6000 with accuracy <0.1%) and power consumptions (Gossen Metrawatt power meter model A2000 with accuracy ± 0.5%). A more detailed description of the system can be found in previous publications (Ruiz-Calvo and Montagud 2014; Cervera-Vázquez, et al. 2015a).

**OPTIMIZATION METHODOLOGY FOR THE WATER CIRCULATION PUMPS FREQUENCY AND TEMPERATURE COMPENSATION**

The authors previously developed an experimental in situ optimization methodology for the water circulation pumps frequency (Cervera-Vázquez, et al. 2015a). Its main advantage is that it is based on experimental measurements and it can be carried out in situ at any installation. Therefore, it is able to take into account the real characteristics of the installation as well as real operating conditions. A summary of this methodology, which consists of three steps, is presented in this subsection.

The first step consists of several experimental tests of pseudo-random sequence of frequency steps, sweeping both circulation pumps (one corresponding to the building hydraulic loop to supply the hot/cold water to the terminal units, and another one on the GSHX hydraulic loop) frequencies in the range of operating frequencies (from 20 Hz to 55 Hz). Through this step it is intended to characterize the performance of the system during the ON time operation of the compressor. Since the heat pump consists of two compressors working in tandem, the tests must be repeated when there is only one compressor cycling ON/OFF or one compressor continuously running and the other one cycling ON/OFF.

The second step consists of analyzing the results obtained from step 1, estimating the system COP under quasi-steady state conditions (only the ON cycle) and finally representing the values of the COP in a contour map as a function of both circulation pumps frequencies.

The third step allows taking into account the influence of the internal circulation pump and the parasitic losses during the OFF cycle calculating, from the quasi-steady state performance maps obtained in step 2 of the methodology, the optimal frequencies as a function of the thermal demand of the building. The thermal demand of the building is represented by the partial load ratio of the system ($\alpha$), which can be defined as the relation between the instantaneous thermal load of the building and the heat pump capacity. This step finally provides the system performance factor as a function of the partial load ratio of the system for different values of both circulation pumps frequencies. A sample of the resulting contour maps is shown in (Cervera-Vázquez, et al. 2015b). From these maps, expressions which provide the optimal values of both circulation pumps’ frequencies as a function of the partial load ratio can be obtained and implemented in the control board of the system. It should be noted that, in order to provide a stable control, the values of the partial load ratio considered in the control algorithm are not instantaneous values. They are calculated for each working cycle of the heat pump according to the following equation:

$$\alpha = \frac{\sum_{n=0}^{N} n \cdot t_{ON,cn}}{N \cdot (t_c)}$$  \hspace{1cm} (1)

According to Eq.(1) the partial load ratio can be defined as a weighted average of the number of compressors switched on ($n$), depending on the amount of time each compressor works during the cycle ($t_{ON,cn}$), and taking into account the total number of compressors of the heat pump ($N$) and the total working time of the cycle ($t_c$). Further details about this calculation can be found in Ruiz-Calvo, et al. (2016). These parameters are measured thanks to the
heat pump control board which provides a signal that indicates the state of each compressor during the GSHP operation.

In order to complete the optimized control, the setpoint of the heat pump, which uses the building supply temperature to control the starts and stops of the compressors, is also varied according to a simple outdoor temperature reset strategy. The controlled variables are the temperature setpoint of the heat pump (\( T_{SB} \)) and the frequency of both the internal (\( f_{ICP} \)) and the external (\( f_{ECP} \)) circulation pumps. The measured variables are the ambient temperature (\( T_{amb} \)) and the partial load ratio of the system (\( \alpha \)). The temperature setpoint of the heat pump was determined as a function of the ambient temperature solely; so, the higher is the ambient temperature, the lower the temperature setpoint is set at the heat pump controller. On the other hand, the circulation pumps’ frequencies were calculated as a function of the partial load ratio solely, as it was determined from the three-step methodology. The higher the load ratio, the higher frequency values were set at the frequency inverters, and hence the water flow rates at both hydraulic circuits.

The objective of the control algorithm is to optimize the energy performance of the system, represented by the system performance factor (\( PF_{sys} \)) defined as the relation between the useful thermal energy supplied to the building (\( Q_{useful,IC} \)) and the system energy consumption including both the heat pump and the circulation pumps’ consumption as expressed in Eq. (2).

\[
P_F_{sys} = \frac{\int_0^t Q_{useful,IC} \cdot dt}{\int_0^t (W_{HP} + W_{ECP} + W_{ICP}) \cdot dt}
\]

Finally, the value of the ICP frequency which optimizes the performance factor of the system was calculated, by means of the experimental methodology developed, in terms of the partial load ratio of the system (\( \alpha \)), and the building supply temperature was determined as a function of the ambient temperature (\( T_{amb} \)) as a first control attempt.

As presented in (Cervera-Vázquez, et al. 2015a), this optimized control was implemented in the control board of the system and tested on the geothermal experimental plant for a period of several months. Results showed that energy savings of \((31.70 \pm 0.07)\%\) were obtained by applying the optimized control, when comparing to a standard control consisting of a fixed frequency value of 50 Hz for both circulation pumps and a setpoint of 10°C in cooling mode and 40°C in heating mode. However, it was found that user comfort was not met in extreme weather conditions during the summer. This is mainly because the influence of decreasing the internal water flow rate was not taken into account in the fan coil effectiveness, which resulted in the comfort of the user not being met. This happened because both control strategies were uncoupled, that is to say, they worked independently of what each other did. In order to find the optimal values of the controlled variables, namely the internal circulation pump frequency (\( f_{ICP} \)) and the building supply temperature (\( T_{SB} \)), that optimize the performance factor of the system while being able to meet the user comfort, it is necessary to couple both strategies. In the former control, the supply building temperature was calculated as a function of the ambient temperature solely (\( T_{SB} = f(T_{amb}) \)), whereas the internal frequency was determined considering only the partial load ratio (\( f_{ICP} = f(\alpha) \)). Now both control parameters need to be determined as a function of both variables, the ambient temperature and the partial load ratio of the system (\( f_{ICP} = f(T_{amb}, \alpha) \)) and \( T_{SB} = f(T_{amb}, \alpha) \). For a start, the performance factor of the system will depend on the controlled variables: internal circulation pump frequency (\( f_{ICP} \)) and the supply building temperature (\( T_{SB} \)). Its variation can be expressed as the sum of the partial derivatives of the system performance factor with each one of the variables.

\[
\frac{\partial PF_{sys}}{\partial f_{ICP}} \bigg|_{T_{SB}} + \frac{\partial PF_{sys}}{\partial T_{SB}} \bigg|_{f_{ICP}} \frac{dT_{SB}}{df_{ICP}} = 0
\]

The supply building temperature (\( T_{SB} \)) is calculated according to Eq. (4), as a function of the room temperature (\( T_{room} \)), the ambient temperature (\( T_{amb} \)), the parameter \( \beta \) (Eq. (5)), the maximum effectiveness of the fan coil (\( \varepsilon_{FC,max} \)).
and the correlation of the fan coil effectiveness in terms of the internal frequency and the building supply temperature ($\varepsilon_{FC}(f_{ICP}, T_{SB})$). This expression is very similar to the one obtained for cooling mode (Cervera-Vázquez, et al. 2015b). A more detailed explanation can be found in (Cervera-Vázquez 2016).

$$T_{SB} = T_{room} + \frac{\beta \cdot \varepsilon_{FC,\text{max}} \cdot (T_{room} - T_{amb})}{\varepsilon_{FC}(f_{ICP}, T_{SB})}$$

$$\beta = \frac{T_{room} - T_{SB,\text{min}}}{T_{amb,\text{max}} - T_{room}}$$

In order to calculate the building supply temperature, the value of the internal frequency resulting from the control surfaces (Cervera-Vázquez 2016) is considered and Eq. (4) is solved for $T_{SB}$ (2nd order equation, the negative root is the right solution, as in cooling mode). However, there are both an upper and a lower limit in the setpoint that should not be overpassed. The upper limit is 45°C. There is no need to supply the water to the building at a temperature higher than 45°C, as it would not mean any improvement in the user comfort but it would mean higher energy consumption. The lower limit is 35°C. It does not make sense to supply the water at a temperature lower than 35°C, as it would be better to turn off the heat pump instead. As far as the lower limit is concerned, there is no problem in setting a temperature higher than the one calculated. The energy savings will not be so important, but the user comfort will still be met. However, if the algorithm calculates a value of $T_{SB}$ higher than 45°C (which is unlikely to happen considered the mild winters in Valencia), but the setpoint is set to 45°C, the user comfort might not be met. That is the reason why, in case that this happens, the internal frequency should be recalculated by means of Eq. (4). Solving Eq. (4) for $f_{ICP}$ and considering a $T_{SB} = 45$°C, will provide the corrected value of the frequency (higher than the previous) that will allow the system to meet the user comfort. Figure 1 shows a flow diagram of the optimized control implemented in the control board of the system for the case of heating mode. The flow diagram of the new optimized control for heating mode presents the same structure than for cooling mode (Cervera-Vázquez, et al. 2015b). In fact, they have the same structure. The main difference lies in the equation used for the calculation of the building supply temperature. The other difference is the temperature limit for $T_{SB}$, which is 45°C, instead of 7°C.

![Flow diagram of the new optimized control (heating mode).](image)

Figure 1  Flow diagram of the new optimized control (heating mode).
DISCUSSION OF RESULTS

Results for one typical heating day are presented in Figure 2. The structure of the figure is the same as for cooling mode (Cervera-Vázquez, et al. 2015b). Figure 2a, depicts the power consumption of the compressor and the system temperatures, whereas Figure 2b shows the partial load ratio and the circulation pumps’ frequencies.

As it can be observed in Figure 2a, the higher the ambient temperature, the lower the temperature of the water supplied to the building. When the calculated $T_{SB}$ is lower than 35°C, the limit applies and the actual supply temperature ($T_{tank}$) cycles around 35°C. It can be observed that the building supply temperature does not get the limit value of 45°C at any time. The reason for this is that the last winter has been very mild in Valencia and there are no days in which that limit value has been reached.

Regarding the frequencies, the set values are directly proportional to the value of the partial load ratio. This can be observed in Figure 2b. On the other hand, a positive users’ feedback was received, and the temperature and relative humidity in two selected rooms was observed to be inside the acceptable comfort range according to Spanish
regulations: [23°C,25°C] and [45%,60%], for cooling mode; [21°C,23°C] and [40%,50%], for heating mode. So, the main target of this new optimized control which was to meet user comfort while saving as much energy as possible, was finally reached.

Regarding the energy savings, in order to carry out a proper energy optimization analysis, the optimized control has been compared to a standard control, as it was done for the first approach to the integrated control, consisting of a temperature setpoint of 7°C (45°C for heating mode) and 50 Hz of frequency for both circulation pumps, according to the GSHP manufacturer recommendations. This standard control is slightly different from the one employed for the first approach, due to the specifications of the GROUND-MED project, where the minimum setpoint allowed was 10°C for cooling mode and 40°C for heating mode, according to the targets of the project.

The same automatic control in the control board of the system has been employed in such a way that the standard control is applied on odd days and the optimized control is used on even days, in order to finally obtain 50% of the days working with each type of control algorithm.

Figure 3 shows the values of the daily performance factors DPFs (integration period equal to 24hours) from $D_{PF1}$ to $D_{PF3}$, for the period analysed, that is to say, from November 2015 to February 2016. $D_{PF1}$ considers only the heat pump consumption, $D_{PF2}$ includes also the consumption of the external circulation pump and finally $D_{PF3}$ adds the consumption of the internal circulation pump, being this factor the daily system performance factor ($D_{PFsys}$).

The alternation of standard and optimized days can only be seen from January onwards. The explanation to this is that the implementation and tuning up of the upgraded algorithm in the control board of the system came with a delay of one month since the beginning of the season (November). Therefore, in an attempt to equal the number of standard days obtained in November, the optimized control was applied on a daily basis during the month of December. Still, in the case of heating mode a similar or even higher improvement in $D_{PF3}$ can be observed.

Figure 3 shows DPFs calculated for each day in the analysed period. Getting together the standard days on one hand, and the optimized days on the other hand, a kind of Seasonal Performance Factor (SPF) can be obtained as the relation between the useful energy transferred to the building and the energy consumption for each of the different types of control over the period where each one was applied. This way, the improvement achieved by the optimized control when compared to the standard control can be analysed for both seasons. Figure 4 presents this comparison: Figure 4a, for cooling mode and Figure 4b for heating mode. When compared to the results of the first approach, it can be observed that the improvement on $SPF_1$ (heat pump seasonal performance factor) is more representative with the upgrade. This is because a better control of the heat pump setpoint is achieved with the new optimized control.

![Figure 3](image_url) Daily performance factor (upgrade). Heating mode campaign.
However, the interest should be on $SPF_3$ which is the system seasonal performance factor as it considers the energy consumption of all the system components except the terminal units, it is to say, it includes the energy consumption of the heat pump and both circulation pumps. Therefore, it is the performance factor that expresses the energy savings in a more reliable way. In the case of cooling mode, this improvement is 35%, which is slightly worse than with the first approach. In the case of heating mode, a 53% improvement is obtained in $SPF_3$. It should be noted that the difference in the percentage of improvement when compared to the first approach, above all in heating mode (which presented an improvement of 32%), is mainly due to two reasons. First, the standard control considered in the first approach was more restrictive, due to the requirements of GROUND-MED project, whereas the one considered when the upgraded algorithm was tested was a more realistic standard control suggested by the heat pump manufacturer. And second, this winter season has been especially mild in Valencia, which means lower thermal energy loads and therefore lower partial load ratios, what has led to a higher improvement of the energy performance. Finally, it should be noted that the impact of the outside temperature and the building occupation in the energy savings cannot be isolated and the energy savings might change from year to year. This is why they need to be measured and checked year by year.

![Figure 4](image)

**Figure 4** System seasonal performance factor (upgrade): (a) cooling mode test campaign; (b) heating mode test campaign.

**CONCLUSIONS**

A new upgraded control algorithm to keep the system working at its optimal point and also meeting the user comfort was developed, implemented and tested in heating and cooling mode in a real GSHP installation located at the Universitat Politècnica de València in Valencia, Spain. The new control approach is based on coupling the estimation of the optimal values of the internal circulation pump frequency and the building supply temperature so that the system is able to work under the minimum energy consumption, provided that the fan coils have a capacity such as to cover the thermal demand and meet the user comfort even in extreme weather conditions. The values of both controlled variables (internal circulation pump frequency and building supply temperature) are calculated as a function of both the ambient temperature (outdoor conditions) and the partial load ratio of the system (thermal load conditions), making the geothermal system smart and adaptive to actual operating conditions. The energy optimization methodology developed in this work is perfectly applicable to any other ground source heat pump installations of the same type. Results show an improvement in the seasonal performance factor of the system up to 35% in cooling mode and 53% in heating mode.
REFERENCES


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Design and Analytical Analysis of Foundation Pile Ground Heat Exchanger with Spiral Coils

Yi Man, Yunxia Qu, Zejiang Wang, Zhaohong Fang

ABSTRACT
Recently, utilization of building foundation piles as the ground heat exchanger (GHE) received more and more attention since it can reduce the initial cost and land area requirement compared with the borehole GHE. This study designs a foundation pile GHE with spiral coil (FPGHE) by intertwining the circulating coil pipe tightly in spiral shape against the reinforcing steel of a pile. The distinct advantage of this proposed FPGHE is that it can offer higher heat transfer efficiency, reduce pipe connection complexity, prevent air blocking and decrease the thermal “short-circuit” between the feed and return pipes compared with other existing configurations. In order to analyze its heat transfer characteristic, analytical models are established for the proposed FPGHE. Analytical thermal analysis is carried out to simulate temperature responses of the coil pipe wall and the circulating water entering/effusing the FPGHE to the short time step heat transfer loads based on the established analytical model. Furthermore, the operation performance and heat exchange capacity of the FPGHE is investigated.

INTRODUCTION
The ground heat exchangers (GHE) with vertical boreholes (Bose, et al. 1985) have been the mainstream technology for the ground coupled heat pump systems, but the high initial cost and land area requirement to install the borehole GHE remain the major obstacles of this technology. Since the foundation pile is commonly used in high rise buildings, combining the heat exchanger and building foundation pile can eliminate the drilling expense and land area requirement of borehole GHE. Therefore, utilization of building foundation piles as the GHE has received more and more attention (Mehrizi, et al. 2016; Luo, et al. 2016; Huerta and Krarti 2015; Loveridge and Powrie 2014). In existing studies, pipes are usually buried in foundation piles in configurations of U-tubes, W-tubes or spiral coils. For the first two configurations, the heat transfer area inside foundation pile is small and the air blocking may occur in pipes. In order to overcome these drawbacks, this study focus on the third type and designs a foundation pile GHE with spiral coil (FPGHE). The circulation coil pipe is intertwined tightly in spiral shape against the reinforcing steel of a pile, and is disposed within about 0.1m of the pile’s outer surface. The distinct advantage of FPGHE is that it can offer higher heat transfer efficiency, reduce pipe connection complexity, prevent air blocking and decrease the thermal “short-circuit” between the feed and return pipes compared with other existing configurations. The schematic diagrams of a conventional single U-tube vertical borehole GHE and the FPGHE with spiral coils are compared in Figure 1.

Yi Man (manyilaura@163.com) is associate professor, Yunxia Qu is professor, Zhaohong Fang is professor, and Zejiang Wang is master degree candidate of Thermal Engineering School at Shandong Jianzhu University in China.
Modeling the FPGHE with spiral coils is complex and existing studies concentrated on the numerical or experimental methods. Suryatriyastuti carried out a 3-D numerical heat transfer analysis on the energy pile under the constant heat load case (Suryatriyastuti, et al. 2012). Xiang built a numerical model includes a 1-D transient convection–diffusion submodel for the fluid domain and a 1-D transient diffusion submodel for the solid domain (Xiang, et al. 2015). Luo conducted the thermal performance test to analyze the operation performance of energy pile under an intermittent condition (Luo, et al. 2016). In the present study, an analytical method is explored as it can provide a more practical and convenient tool for engineering design, as well as thermal analysis of the FPGHE, compared with existing numerical and experimental methods.

For the proposed FPGHE, its diameter is much thicker and depth is usually shorter compared with the borehole GHE. Obviously, classical heat transfer models for the borehole GHE fail for the FPGHE. By analyzing the heat transfer process of proposed FPGHE, the analytical finite spiral heat source model is established in this study based on the Green’s function theory, the virtual heat source theory, and the superposition method. The temperature responses of the spiral heat source, the coil pipe wall, and the circulating water entering/effusing the FPGHE to the short time step heat transfer loads are deduced based on the established analytical model. Then the operation performance and the heat exchange capacity of the FPGHE is investigated.

DESIGN OF FOUNDATION PILE GHE WITH SPIRAL COIL

The high-density polyethylene (HDPE) pipe with exterior and interior diameters of 25mm and 20mm, respectively, are selected as the circulation pipe of proposed FPGHE. First hydrostatic test with pressure of 0.8Mpa and last for 15min are required in order to prevent the leakage before the pipe are installed. Then the pipe is intertwined tightly in spiral shape against the reinforcing steel cage of a pile, as shown in Figure 2(a). The second hydrostatic test with pressure of 0.8Mpa and last for 15min should be carried out followed. Then the combination of reinforcing steel cage and coil pipe are put into the hole of pile. After the third hydrostatic test with pressure of 0.8Mpa and last for 2 hours, the last step is the concrete pouring of the pile foundation, as shown in Figure 2(b).

Extreme caution should be paid during the concrete pouring, and coarse aggregate in concrete must be smooth and non-angular particles. The conduit are utilized to lead the concrete into the bottom of pile hole, and it should be extracted gradually from the bottom to top according to grouting speed. During the placement and extraction process of conduit, it is important to keep the vertical and center for preventing the hanging cage of concrete, ensuring the compaction of pouring and decreasing the heat transfer resistance. The concrete pouring process is finished when the density of return slurry is identical with which of pouring concrete.
SPIRAL HEAT SOURCE MODEL ESTABLISHMENT

As shown in Figure 3, the buried coil inside FPGHE is simplified into heat source in spiral shape with finite-length. Then the finite spiral heat source model is developed to take the three-dimensional geometrical characteristic of the spiral pile as well as the effects of heat flow through the top and bottom ends of pile into account.

Figure 3 Proposed FPGHE, the finite spiral heat source model and the virtual heat source model

Green’s Function and Temperature Response to a Point Heat Source

According to the Green’s function theory, the Green’s function is the temperature response to an instantaneous point source of heat. For an instantaneous point heat source with intensity of $\rho c$, located at \((r',\varphi',z')\) and activated at the instant \(\tau'\), its Green’s function in the cylindrical coordinates at point \((r,\varphi,z)\) can be expressed as:

\[
G(r,\varphi,z;\tau',r',\varphi',z') = \frac{1}{8\pi\alpha(\tau'-\tau')} \cdot \exp\left[-\frac{(r \cos \varphi - r' \cos \varphi')^2 + (r \sin \varphi - r' \sin \varphi')^2 + (z - z')^2}{4\alpha(\tau - \tau')}\right]
\]

(1)

Then the temperature response in the medium resulted from the step heating of the spiral heat source can be expressed as the integration of all the point sources acted successively from the starting instant \(\tau' = 0\).

Temperature Response to the Finite Spiral Heat Source
As in most analytical models, assumptions are made for finite spiral source model as follows:

(1) The ground is regarded as a homogeneous semi-infinite medium, and its thermophysical properties do not change with temperature.

(2) The boundary of the medium, i.e. the ground surface, remains constant throughout the period concerned.

(3) The medium has a uniform initial temperature, \( t_0 \).

(4) The spiral heat source with radius \( r_0 \) and pitch \( b \) stretches from the \( h_1 \) to \( h_2 \) below the ground surface, and each circle of the spiral sources emits constant heat transfer rate at the intensity of \( q_l b \) from time zero, i.e. \( \tau = 0 \).

With the virtual heat source theory, a virtual spiral heat sink with negative heating rate \(- q_l\) and of identical physical dimensions is set on symmetry to the boundary in order to keep the constant temperature of the ground surface, as shown in Figure 3. The finite spiral heat source and heat sink can then be approximated as the sum of numerous point heat sources and heat sinks. Then, the Green’s function theory and the superposition method is employed to obtain the temperature response of the medium:

\[
\theta_{r, \text{spiral}} = \frac{q_l b}{2 \pi k} \int_{2 \pi a}^{2 \pi b} \left[ G(z' = b \phi'/2\pi) dq' - G(z' = -b \phi'/2\pi) dq'\right]
\]

\[
= \frac{q_l b}{16 \pi k} \int_{2 \pi a}^{2 \pi b} \exp \left[ -\frac{r^2 + r_0^2}{4 a (\tau - \tau')} \right] \exp \left[ \frac{(z - b \phi'/2\pi)}{4 a (\tau - \tau')} \right] \exp \left[ \frac{(z + b \phi'/2\pi)}{4 a (\tau - \tau')} \right] dq'
\]

By introducing the following dimensionless variables:

\[
\Theta_{r, \text{spiral}} = \frac{k \theta_{\text{spiral}}}{q_l}, \; \phi_0 = \frac{\pi r_0}{2 b}, \; \phi_0' = \frac{\pi r_0}{2 b}, \; R = \frac{r}{r_0}, \; Z = \frac{z}{r_0}, \; Z' = \frac{z'}{r_0}, \; B = \frac{b}{r_0},
\]

\(H_1 = \frac{h_1}{r_0}\) and \(H_2 = \frac{h_2}{r_0}\), the dimensionless temperature excess around the finite spiral source can be expressed as:

\[
\Theta_{r, \text{spiral}} = \frac{B}{16 \pi k} \int_{2 \pi a}^{2 \pi b} \exp \left[ \frac{1}{4 (\phi_0 - \phi_0')} \right] \exp \left[ \frac{R^2 + 1}{4 (\phi_0 - \phi_0')} \right] \exp \left[ \frac{(Z - b \phi'/2\pi)}{4 (\phi_0 - \phi_0')} \right] \exp \left[ \frac{(Z + b \phi'/2\pi)}{4 (\phi_0 - \phi_0')} \right] dq' d\phi_0'
\]

**ANALYTICAL SOLUTIONS AND DISCUSSION**

**Temperature Field Solutions of Finite Spiral Source Model**

The temperature responses of the finite spiral source model can be calculated according to Equation (3). Take an example of finite spiral source with \( B = 1 \), \( H_1 = 2.0 \), and \( H_2 = 12.0 \), the temperature fields covering the pile and its surrounding soil at different dimensionless times of \( \phi_0 = 0.2 \), \( \phi_0 = 1.0 \) and \( \phi_0 = 5.0 \) are described in Figure 4. As shown, the temperature rise fluctuates considerably in the vicinity to the spiral source and shows a periodic variation in the Z direction. Besides, the influence of the axial heat conduction caused by the boundary and finite length of the heat source is limited to the two ends of the pile in relatively short periods of time, and the impact of heat dissipation through the ends penetrate deeper into the whole depth of the pile for longer term operation.
Temperature Response of Pipe Wall

When the dimension of the coil pipe is considered, the heat source can be approximated as located at the centre of coil pipe and the temperature response of the coil pipe wall located at \( r_p \) away from spiral heat source can be deduced. Points on the spiral heat source can be located by the coordinate \( \varphi \) with the other coordinates being \( r = r_0 \) and \( z = z_0 \) in the cylindrical coordinates, as shown in Figure 5.

![Figure 5 Spirial coil pipe buried in pile GHE](image)

Consequently, while the slope of the spiral is neglected, the coordinates of a point on the pipe outer circumference can be determined approximately as:

\[
\begin{align*}
  r &= r_0 + r_p \cdot \cos \alpha \\
  \varphi &= \varphi \\
  z &= z_0 + r_p \cdot \sin \alpha
\end{align*}
\]

By introducing the dimensionless variables \( R = \frac{r}{r_0} \), \( Z = \frac{z}{z_0} \), \( R_p = \frac{r_p}{r_0} \), \( Z_0 = \frac{z_0}{z_0} \), \( B = \frac{b}{r_0} \), the dimensionless coordinate values of points located on the coil pipe wall becomes:

\[
\begin{align*}
  R &= R_p \cos \alpha + 1 = R_p \cos \alpha + 1 \\
  Z &= \frac{Z_0}{z_0} \sin \alpha + Z_0 \sin \alpha + \frac{B \varphi}{2\pi}
\end{align*}
\]

Then the temperature distribution on the coil pipe wall can be deduced according to the Equation (3):
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\[ \Theta_{\text{spiral pipe}}(\alpha, \phi, Fo) = \frac{B}{16\pi^2} \int_{0}^{1} \frac{1}{(Fo - Fo')} \exp \left[ \frac{(R_o \cdot \cos \alpha + 1)^2 + 1}{4(Fo - Fo')} \right] \cdot \exp \left[ \frac{2(R_o \cdot \cos \alpha + 1) \cos(\phi - \phi')}{4(Fo - Fo')} \right] \cdot \exp \left[ \frac{(R_o \cdot \sin \alpha + B/2\pi \cdot (\phi + \phi'))^2}{4(Fo - Fo')} \right] \cdot d\phi' dFo' \] (6)

For engineering application, the integral mean temperature response of the coil pipes wall over its circumference in \( \xi \)-direction can be selected to represent the pipe wall temperature response of the whole FPGHE.

**Coil Pipe Wall Temperature Response to the Practical Heat Current**

The practical mutative heat currents imposed on the FPGHE can be approximated by the sum of a serious of rectangular pulse heat currents. Further, a single pulse heat current can be approximated by superposition of two step heat currents with the initial one \( q_{in} = 0 \). Therefore, concepts of \( p \)-function and \( q \)-function which represent the non-dimensional temperature response of the pipe wall to the step heat current and the pulse heat current, respectively, are developed to calculate the pipe wall temperature response to practical heat current which varies from time to time:

\[ p(\tau) = \frac{k}{\rho c p} \theta_{\text{p}, \phi, \rho}, \quad q(\tau - \tau_i) = p(\tau - \tau_i) - p(\tau) \] (7)

According to superimposing theory, the temperature response of the borehole wall at moment \( \tau \) can be deduced based on the short time step pulse heat currents \( q_i \):

\[ \theta_{\text{p}, \phi, \rho} = \frac{1}{k} \sum_{i=1}^{n} (q_i - q_{in}) \cdot p(\tau - \tau_i) \]

\[ = \frac{1}{k} \left[ \sum_{i=1}^{n} q_i \cdot p(\tau - \tau_i) - \sum_{i=1}^{n} q_{in} \cdot p(\tau - \tau_i) \right] = \frac{1}{k} \left[ \sum_{i=1}^{n} q_i \cdot p(\tau - \tau_i) - q_{in} \cdot p(\tau - \tau_i) - \sum_{i=1}^{n} q_{in} \cdot p(\tau - \tau_i) \right] \]

\[ = \frac{1}{k} \sum_{i=1}^{n} q_i \cdot [p(\tau - \tau_i) - p(\tau - \tau_i)] = \frac{1}{k} \sum_{i=1}^{n} q_i \cdot [q(\tau - \tau_i)] \]

**Temperature of Circulating Fluid**

The temperature of fluid circulating in the coil pipe is of primary importance to design and simulate the FPGHE. Compared with the ground outside the pile, both the dimensional scale and thermal mass of the coil pipe are much smaller. Moreover, the temperature variation inside the coil pipe is usually slow and minor. Thus, the heat transfer of fluid inside the FPGHE can be approximated as a steady-state process. The heat transfer resistance between the coil pipe external wall and circulating fluid can be calculated:

\[ R_p = \frac{1}{2\pi h_j} \ln \left( \frac{r_e}{r_p} \right) + \frac{1}{2\pi r_p h_j} \]

(9)

where \( h_j \) is the convection heat transfer coefficient between the pipe wall and circulating fluid.

\[ h_j = \varepsilon_k \cdot \frac{k_f - Nu}{2\pi} \]

(10)

\( Nu \) is the Nusselt number for a straight pipe. \( \varepsilon_k \) is an amendatory factor (Rohsenow and Hartnett 1983) of the convection heat transfer coefficient due to the augment effect of centrifugal force of fluid flow in the spiral channel.

\[ \varepsilon_k = 1 + 10.3 \left( \frac{2r_e}{\varepsilon_k} \right)^{0.4} \]

(11)

Due to the steady-state process simplification, the average temperature of circulating fluid can be calculated:

\[ T_f = \frac{T_{in} + T_{out}}{2} = T_p + \frac{q \cdot b}{\sqrt{2\pi \cdot r_p^2} + b^2} \cdot R_p \]

(12)
In view of heat balance for the FPGHE, one also has:

$$\Delta T = T_i' - T_i'' = \frac{q_i(b_i-h_i)}{M \cdot C}$$  \hspace{1cm} (13)

Then the entering and effusing fluid temperatures of the FPGHE can be determined by:

$$T_i' = T_p' + \frac{q_i \cdot b \cdot R_p}{\sqrt{(2\pi \cdot t)_f} + b^2} + \frac{q_i(h_i-h_f)}{2 \cdot M \cdot C_p}$$

$$T_i'' = T_p'' + \frac{q_i \cdot b \cdot R_p}{\sqrt{(2\pi \cdot t)_f} + b^2} - \frac{q_i(h_i-h_f)}{2 \cdot M \cdot C_p}$$  \hspace{1cm} (14)

**Operation Simulation of a Sample FPGHE**

A FPGHE with normal pile configurations for engineering application is selected as a sample in this study: $r_0=0.4 \text{m}$, $h_1=2 \text{m}$, $h_2=22 \text{m}$, $b=0.4 \text{m}$, $r_p=20 \text{mm}$ and $r_f=32 \text{mm}$. The flow velocity of fluid inside coil pipe is 0.5m/s for ensuring the turbulent flow. The undisturbed temperature of surrounding soil is 12.5°C. The hourly heat extraction/injection loads afforded by the sample FPGHE is assigned by an air conditioning loads software simulation program, as shown in Figure 6(a). Then the hourly operation parameters of the sample FPGHE in one year are simulated according to the proposed analytical model and plotted in Figure 6(b). According to simulative results, the maximum and minimum temperature effusing the FPGHE for cooling and heating provision is 28.42°C and 3.81°C, respectively. The heat exchange capacity of the sample FPGHE is about 212W/m.

![Figure 6](image)

**CONCLUSION**

This paper presents a model of a FPGHE with spiral coil, which can offer higher heat transfer efficiency, reduce pipe connection complexity, prevent air blocking and decrease the thermal “short-circuit” between the feed and return pipes. The design and manufacture key points of proposed FPGHE are illustrated.

In order to provide an appropriate and convenient tool for design and thermal analysis of the FPGHE with spiral coils, the analytical finite spiral heat source model is established in this study based on the Green’s function theory, the virtual heat source theory, and the superposition method. The temperature responses of the spiral heat source, the coil pipe wall, and the circulating water entering/effusing the FPGHE to the short time step heat transfer loads are deduced based on the established analytical model. Then the operation performance and the heat exchange capacity of the FPGHE is investigated. Analytical analysis developed in this study can provide an proper method for design and thermal analysis of the FPGHE with spiral coils as well as other similar engineering problems. For further research, the optimal design parameters, such as coil pitch, of proposed FPGHE should be discussed based on the established spiral heat source model and other relevant mechanical models.
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NOMENCLATURE

\( \alpha \) = Angular parameter (rad)
\( \Theta \) = Dimensionless temperature excess
\( \theta \) = Temperature excess (K)
\( \rho \) = Density (kg m\(^{-3}\))
\( \tau \) = Time (s)
\( \varphi \) = Angular coordinate (rad)
\( a \) = Thermal diffusivity (m\(^2\) s\(^{-1}\))
\( b \) = Coil or spiral pitch (m)
\( c \) = Specific heat of fluid (J kg\(^{-1}\) K\(^{-1}\))
\( Fo \) = Fourier number
\( h_f \) = Convective heat transfer coefficient (W m\(^{-2}\) K\(^{-1}\))
\( h_{1,2} \) = Pile depth (m)
\( k_f \) = Thermal conductivity of circulating fluid (W m\(^{-1}\) K\(^{-1}\))
\( M \) = Mass flow rate of circulating water (kg/s)
\( q_I \) = Heating rate per length of the heat source (W m\(^{-1}\))
\( r_0 \) = Spiral radius (m)
\( r_p \) = Exterior radius of coil pipe (m)
\( r_{pi} \) = Interior radius of coil pipe (m)
\( R_p \) = Heat transfer resistance between coil pipe wall and circulating fluid (m K W\(^{-1}\))
\( T_f' \) = Temperature of fluid entering the pile GHE (K)
\( T_f'' \) = Temperature of fluid effusing the pile GHE (K)

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Numerical simulation of pile geothermal heat exchanger with spiral tube considering its thermo-mechanical behavior

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ABSTRACT
Pile geothermal heat exchanger (PGHE), which utilizes the building foundation piles as part of the geothermal heat exchangers (GHEs) for a ground-coupled heat pump (GCHP) system, has been attracting the interests of researchers and engineers. However, the continuous heat rejection/extraction of the PGHE to/from the piles will cause significant temperature variations (up to 25°C) of piles and the surrounding soil, which can influence the mechanical behavior of the pile foundation severely. A modified direct shear apparatus has been developed to investigate the interface behavior between soil and pile. Then, based on the experiment results, the thermo-mechanical behavior of PGHE with spiral coils was investigated by a 3-D simulation model. The thermal loads induce additional compressive stress when the temperature rise, and the local compressive stress can reach to 9.35MPa near the heat exchanger pipe. Additionally, heat extraction led to a decrease of friction angle and normal contact pressure at the interface between soil and pile, and as a consequence, the shear force decreases with the temperature drop. Compared with no thermal disturbance, the ultimate friction resistance of pile is weakened by 15.37%.

INTRODUCTION
Ground-coupled heat pump (GCHP) technology, utilizing the shallow geothermal energy and heat capacity, has become an attractive alternative for space heating and cooling. Traditionally, the ground heat exchanger (GHE) consists of connected high-density polyethylene (HDPE) U-tubes buried in a number of ground boreholes with depth ranging from 50 to 120m and separated space in 4 to 5m (Wang et al., 2016). The heat exchange fluid is circulated inside the HDPE pipes. In general, the GHE with vertical boreholes requires a higher initial cost and a large plot of land for borehole installation, which has consequently hindered the wide applications of the technology in densely-populated cities. Luckily, a novel pile GHE (PGHE), which utilizes the building foundation piles as part of the GHE for a GCHP system, has been attracting the interests of researchers and engineers.

However, the continuous heat rejection/extraction of the PGHE to/from the piles will cause significant temperature variations (up to 20°C) of piles and the surrounding soil, which can influence the mechanical behavior of the pile foundation (Laloui et al., 2006). With temperature rise/drop and different thermal expansion coefficients of pile and soil, the thermal deformation between piles and the surrounding soil will become inconsistent and the stress
on the pile-soil interface would change (Knellwolf et al., 2011). The pile expands with the increase of its temperature, and accordingly the axial fixity at the head and toe of the pile and the lateral constraint from the surrounding soil would cause additional axial normal stress and lateral shear stress respectively (Ng et al., 2014). This will increase the compressive stress in the pile.

Most previous studies are focusing on the investigation of thermal induced inner stress variation, but little on the influence of friction force of energy pile. Additionally, investigations of the thermo-mechanical behavior of PGHE with U-tube and W-tube has been partially studied, but there is a lack of thorough and comprehensive thermo-mechanical study on the PGHE with spiral coils. Therefore, the present work aims to investigate the interface behavior between pile and soil and the thermo-mechanical behavior of PGHE with spiral coils. An interface behavior test is introduced for analyzing the element behavior of concrete-soil under different thermal loads. A detailed experiment design concept and experiment progress are presented. Based on the experimental results, a numerical simulation model is established to systematically analyze the thermo-mechanical performance of PGHE with spiral-tubes. Heating and cooling conditions are both simulated.

INTERFACE BEHAVIOUR TEST

The interface behavior between structural materials and soil is a critical parameter for the design and safety assessment of the pile foundation. Thus, the influence of thermal load on the interface behavior should be investigated first before any theoretical or numerical analysis for PGHE with spiral-tube.

New interface test apparatus

The traditional interface test is usually conducted by a direct shear apparatus, Figure 1 (a), which consists of two shear boxes. The specimens (structural materials and soil), contained in the box, is subjected to a constant normal load while a constant horizontal displacement is applied to the lower shear box. This displacement causes an increasing force at the interface, and a shear failure occurs when this force is beyond the shear strength. However, the traditional apparatus is failed to control the temperature and saturability during the shearing process, thus it cannot use to investigate the thermal effect.

Based on the traditional shear apparatus, a new direct shearing apparatus with temperature control (DSA-T) was designed, developed and employed. The new apparatus consists of an air pressure chamber, an air circulation pump, a heating/cooling system, a solution system and measuring/monitoring system, as shown in Figure 1 (b).

![Figure 1](image)

**Figure 1** Schematic diagram of (a) a traditional direct shear apparatus and (b) the new direct shear apparatus with temperature control.
One of the major differences of the new apparatus is the temperature control system that can simulate the heating and cooling process of energy pile. Another one is the suction control method. A solution circulation system is applied to control the humidity level in the air chamber so that the suction will be maintained during the shearing process. To enhance the heat and mass transfer in the air chamber, an air circulation pump is installed and has a maximum flow rate of 1 L/min. Thermal and humidity sensors are applied to monitor the heat and mass transfer in the shear box. The data logger reads all the measurements and communicates to the computer during the whole experiment.

**Experiment program**

Concrete is selected as the structure material in this soil-structure interface behavior experiment. For the soil material, two typical soil materials are selected. One is quartz sand, and the other one is clay. The experimental campaign can be divided into two groups: 1) interface test of sand-concrete and 2) interface test of clay-concrete. For the first part, interface test on sand-concrete, all the sand samples were sheared under dry conditions. All the prepared samples firstly experienced a consolidation process, and then a target thermal load was applied to the test chamber until the end of each shear test. After 24 hours, it is assumed that the test sample achieves the thermal equilibrium. Then, the sample was sheared under certain normal stress. Three normal stress, 50kPa, 100kPa, and 150kPa, are considered as the effective normal stress both for the interface test of sand and clay. For the interface test on clay-concrete, the loading path is basically the same with the sand-concrete test, but with a little difference in some aspects. All the clay soil samples are run under full saturated vapor pressure conditions. Both for these two interface behavior tests of sand-concrete and clay-concrete, three different temperature conditions, 8°C, 24°C and 60°C are investigated.

**Concrete.** The concrete was prepared in the laboratory mixing cement, water, and aggregates, based on the JGJ 55-2011. The target density of concrete is assumed to be around 2100 kg/m$^3$, thus the volume of aggregates is 250g and the cement is 125g mixed with 250g water. The normal river sand is used as the aggregate with particles diameter equal or smaller than 1mm.

**Quartz sand.** The quartz sand, extracted from a China quarry, was selected for the interface experiment of the sand-concrete. The grain size of the test sand ranges between 0.008 and 1.0 mm. The grain size distribution is presented in Figure 2 (a).

**Sandy clay.** The red clay soil, which is widely distributing in the east of China and used as a backfill material locally, was used in this study. The original soils used in this test were collected from a clay quarry at Hebei province, China. It can be seen in Figure 2 (b), that the clay has a fine fraction, as shown in the grain size distribution curve. This material is composed of Fe$\_2$O$\_3$ (15.2%), Al$\_2$O$\_3$ (30.03%), SiO$\_2$ (46.85%), K$\_2$O (3.16%), MgO (2.09%). The soil material was dried in an oven at 105°C over 24 hours and conserved tightly sealed. Before the shearing test, the specimens were mixed with distilled water to a target water content of 23% and filled into the shear box.

![Figure 2](threshold)  Particle size distribution of (a) quartz sand (b) sandy clay.
NUMERICAL SIMULATION

Based on the experimental results, a numerical simulation model is established to systematically analyze the thermo-mechanical performance of PGHE with spiral-tubes. The simulations were carried for a single energy pile with a diameter of 1.2m and a depth of 15m. It is assumed that a spiral-tube was buried in the pile with a loop diameter of 1.0 m and a spiral pitch of 0.4 m. The depth of the soil domain was set to be the double of the pile depth, and its radius was considered to be 15 times of the pile radius. The large enough soil domain is to ensure that the boundary condition of the soil has less influence for the heat transfer and mechanical performance of PGHE within the simulation period. Two typical heat transfer conditions (heating and cooling) was taken into consideration to simulate the working status of PGHE in winter and summer. As the influence of thermo-mechanical behavior of PGHE is not obvious in short time operation, it would be preferable to carry out a long-term simulation, i.e., 30 days, which would be a better way to understand the thermal and mechanical performances of PGHE.

The model and the mesh generation were built on the ABAQUS CAE. The sensitivity to the grid resolution of this numerical model has been considered and validated. In this analysis, the spiral tube is simplified as a series of separated ring-coils, and the whole calculation domain is axial symmetry. Thus, this three-dimensional problem can be simplified as an axial symmetry model. Three kinds of physical mechanisms were taken into consideration: 1) the heat transfer process from the geothermal heat exchanger to concrete pile and soil, 2) the mechanical behavior between pile and soil, and 3) the additional thermal stress induced by heat transfer.

Boundary conditions and material properties

As the radius of soil domain is more than ten times than that of the pile, the far-field boundary of soil was treated as adiabatic surfaces and blocked the displacement in the radial direction. The bottom boundary of soil domain was given a constant temperature of 20°C, which was equal to the initial temperature of the whole calculation domain. The boundary condition at the top surface of soil and pile was also assumed to be a constant temperature of 20°C. A fixed mechanical boundary condition was applied at the soil bottom to ensure that the vertical and horizontal displacements at this position are zero. As the main focus is the thermo-mechanical performance of pile and soil, the fluid dynamic within the spiral heat exchanger was not be simulated in this study, which was simplified as a heat flux boundary condition applied on the pipe surface. The heat injection rate is assumed to be 120W/m to simulate the heating process of GHE, and the heat extraction rate is considered to be 40W/m.

Both the pile and soil were assumed to be a purely elastic material, and the plastic deformation of soil is not considered. The metrical properties of soil are based on the results of the interface behavior tests, proposed in the last chapter, and the main properties for this numerical study are summarized in Table 1. Under each heat transfer condition (heating or cooling), two mechanical load behaviors were conducted: one is that no load force was applied on the pile top and the stress was only induced by gravity and temperature variation; the other one is that the pile is under an axis displacement, and both the mechanical and thermal loads were applied on the pile. The displacement rate of pile top was assumed to be 5 mm/day.
Table 1  Summary of concrete and soil properties for the simulation cases

<table>
<thead>
<tr>
<th>Item</th>
<th>Symbol</th>
<th>Unit</th>
<th>Concrete</th>
<th>Soil</th>
</tr>
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<tbody>
<tr>
<td>Conductivity</td>
<td>k</td>
<td>W/(m*K)</td>
<td>1.628</td>
<td>1.82</td>
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<tr>
<td>Density</td>
<td>ρ</td>
<td>kg/m³</td>
<td>2500</td>
<td>2500</td>
</tr>
<tr>
<td>Specific Heat</td>
<td>C_p</td>
<td>J/(kg*K)</td>
<td>837</td>
<td>880</td>
</tr>
<tr>
<td>Young Modulus</td>
<td>E</td>
<td>Pa</td>
<td>2.8E+10</td>
<td>5.2E6</td>
</tr>
<tr>
<td>Poisson’s Ratio</td>
<td>μ</td>
<td>1</td>
<td>0.25</td>
<td>0.35</td>
</tr>
<tr>
<td>Coefficient of Thermal Expansion</td>
<td>α</td>
<td>1/K</td>
<td>1.2E-5</td>
<td>1.75E-5</td>
</tr>
<tr>
<td>Internal Friction Angle</td>
<td>δ</td>
<td>°</td>
<td>-</td>
<td>15.51/17.66/22.05</td>
</tr>
<tr>
<td>Cohesion</td>
<td>c</td>
<td>kPa</td>
<td>-</td>
<td>4.57/4.61/7.51</td>
</tr>
</tbody>
</table>

RESULTS AND DISCUSSION

Interface behavior test

The test results of the sand-concrete and the clay-concrete interface are summarized in Figure 3.

![Figure 3](image-url)

Figure 3  Relationships between shear strength and net normal stress under different thermal load during (a) sand-concrete and (b) clay-concrete interface test.

As shown in Figure 3 (a), the interface response of sand-concrete either at high or low temperature conditions does not have the obvious difference to that at normal temperature condition. The friction angle (δ) of sand-concrete is found to be 25.51 degree on average. But for the case of clay-concrete, Figure 3 (b), the interface friction properties (friction angle and adhesion strength) are enhanced by the high temperature. Compared with the case of normal temperature, the adhesion strength has an improvement of about 63%, and the friction angle increased by 24% at high temperature. The water content loss in high temperature condition should be the main causes of the increases of the adhesion strength and the friction angle. According to the experiment data, the water content of 60 °C is 18.5%, and in the case of 24 °C is 28.3%. Conversely, the friction properties decrease with the as the temperature dropped. In the simulation model, the different relationship lines of failure criteria were applied on the different thermal conditions, and in each thermal condition the failure criteria is appropriate for all stress levels.
Thermo-mechanical response with no axis load

If there is no additional mechanical force applied on the pile top, the stress within the pile foundation should only come from the gravity. Therefore, the initial stress increases linearly with the pile depth, and the frictional shear force should be closed to zero, as shown in Figure 4.

Figure 4  Distribution of (a) vertical stress along the pile axis and (b) shear stress along the pile side at cooling and heating conditions with no axial force

When the spiral heat exchanger injects heat energy into the concrete pile, the thermal stress was excited and pushed the pile to expand from the null point to the top and bottom. In this case, the null point was located at the depth of 8.5 m. A positive shear stress (assuming the direction from the pile bottom to the top is positive) is mobilized under this null point, and a negative one occurs above this point. An inflection point can be observed near the top of the pile, which should be the consequence of the decrease of the normal contact force. An additional thermal stress was mobilized simultaneously. As shown in the stress distribution curves, this thermal load induced stress along the axis is obvious but less than 5% of the concrete ultimate compressive strength. The maximum compressive stress is equal to 5.75Mpa, which was detected in the area near the spiral heat exchanger pipe.

For the case under the cooling condition, a similar phenomenon was shown, but the friction direction is opposite to the case of the heating condition. The temperature change induced a contraction behavior of the pile, and the friction was mobilized to restrict this behavior. The stress distribution curves along the pile axis show that the pile suffered an additional pressure stress with the temperature increase and tensile stress with the temperature decrease. But, it should be pointed out that the tensile stress only occurs near the pile top, and the tensile stress fades away with an increase in depth. The maximum tensile stress detected near the spiral heat exchanger is 0.4Mpa, which has almost reached 50% of the ultimate tensile strength of normal concrete pile.
Thermo-mechanical response under axial force

Figure 5 shows that the shear stress along the pile side and the stress distribution along the pile axis under different thermal loads. Clearly, the temperature rise can induce a stress increase in the pile foundation. It could be noted that the stress curve of heating increase greatly near the pile top. This is due to the ununiformed distribution of the load force. The load force at the center of pile surface is always lower than that of the around area. The maximum compressive stress is found to be 6.25MPa at the depth of 1.12m, and at the same position, the pressure stress in the case of no thermal disturbance is only 4.96MPa. A 0.5MPa decrease was shown in the case of the cooling condition.

Figure 5  Distribution of (a) vertical stress along the pile axis and (b) shear stress along the pile side at cooling and heating conditions under axial force.

The shear stress along the pile is almost with linear distribution. It is because the compressive deformation of the pile is relatively small, and the slip between pile and soil is uniform. As seen in Figure 5 (b), the shear stress can be enhanced by the temperature rise, and weakened by the heat absorbing process. It should be noted that all the stress force is compressive stress, and no tensile stress is mobilized in the pile when the pile is loaded with head force. The temperature does not only increase the friction angle for the interface between soil and pile, but the expansion deformation is also increased by the normal pressure on the interface. These two behaviors are combined to generate an enhanced ratio of 48% on the first third of the pile, when the PGHE operation in the heating mode. This enhancement effect is more remarkable at the top of the pile, and decreases with the pile depth. A similar phenomenon can also be observed in the case of cooling simulation. The average enhancing ratio at 30’ days of heating operation is equal to 34.40 %. On the contrary, the temperature decrease caused a weakening effect on the shear stress, with an average weaken ratio of 15.37%.
CONCLUSION

The application of pile GHE faces an innovative challenge for geotechnical design. The temperature within the concrete pile and soil always changes during the heating or cooling season. This temperature change can induce additional thermal deformation and stress in the pile and physical property changes of soil.

The interface test shows that the friction parameters (friction angle and adhesion strength) of the clay-concrete interface are enhanced by the high temperature. Compared with the case of normal temperature, the adhesion strength has an improvement of about 63%, and the friction angle increased by 24%. Conversely, the friction properties decrease slightly with the temperature reduction. Based on these tested results, a simulation model was established to investigate the thermo-mechanical performance of PGHE with spiral-tubes in full size. As the simulation results demonstrated, the heat exchange process of GHE has a great effect on the skin friction behavior of the pile. A 34.4% increase of bearing capacity is observed in the case of heating simulation and a 15.37% capacity decrease has been found in the case of cooling simulation.

ACKNOWLEDGMENTS

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NOMENCLATURE

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Definition</th>
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<tr>
<td>α</td>
<td>Thermal diffusivity (m²/s)</td>
</tr>
<tr>
<td>k</td>
<td>Thermal conductivity (W/m·K)</td>
</tr>
<tr>
<td>ρ</td>
<td>Density (kg/m³)</td>
</tr>
<tr>
<td>C_p</td>
<td>Specific heat capacity (J/kg·K)</td>
</tr>
<tr>
<td>E</td>
<td>Young modulus (Pa)</td>
</tr>
<tr>
<td>μ</td>
<td>Poisson's ratio</td>
</tr>
<tr>
<td>δ</td>
<td>Internal friction angle (°)</td>
</tr>
<tr>
<td>c</td>
<td>Cohesion (kPa)</td>
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</table>

REFERENCES


Minimum Well Separation for small groundwater heat pump (GWHP) systems in Korea: preliminary analysis based on regional aquifer properties

Byeong-Hak Park Seung-Wook Ha Kang-Kun Lee*

ABSTRACT
Shallow geothermal energy has been estimated to have an excellent applicability in Korea, and its applications for space heating and cooling have steadily increased in recent years. Such application as ground source heat pump (GSHP) system can be classified into closed- and open-loop. In recent years, studies have been conducted to minimize the environmental impacts resulting from pumping/injection and to enhance the efficiency of groundwater heat pump (GWHP) system that is the open-loop system. These studies suggest that the characteristics of the aquifer have a significant role in designing efficient GWHP systems. This study considers various hydrogeological properties of Korea. An open-source numerical code called TRS was used for preliminary and sensitivity analyses of GWHP systems. In the analyses, arrival time when thermal plume arrives at pumping well and temperature change at pumping well were observed with different pumping/injection rates, hydraulic gradient, and well separation. Thus, we derived adequate well arrangement for efficient GWHP operation.

INTRODUCTION
Among available new and renewable energy sources, shallow geothermal energy that takes advantage of stable underground temperatures has been estimated to have excellent applicability in Korea (Mok et al., 2010). Since the “Promotional Law of New and Renewable Energy Development, Use and Dissemination” was enacted in 2004, the use of ground source heat pump (GSHP) systems in Korea has steadily increased in recent years because of their benefits and the governmental support (Lee, 2009; KEMCO, 2011; Kwon et al., 2012). Such application as GSHP system can be classified into closed- and open-loop systems.

An open-loop system extracts groundwater from one well, exchanges heat energy with the water, and injects the water into another well. In recent years, studies have been conducted to minimize the environmental impacts resulting from pumping/injection and to enhance the efficiency of groundwater heat pump (GWHP) systems (Lo Russo et al., 2011; Zhou et al., 2013). The applicability of groundwater for cooling large-scale facilities has also been estimated (Al-Zyoud et al., 2014). These studies suggest that the aquifer characteristics play an important role in designing the open-loop systems.

However, characterization of the aquifer, which is required to design the efficient system, is very expensive in terms of time and cost, and is not reasonable for small applications. In this context, an open-source numerical code called TRS (Casasso and Sethi, 2015) was selected for preliminary and sensitivity analyses of small-scale GWHP systems because it can deal with flow and heat transport in a well doublet and does not requires much input data. In the analyses,
arrival time when thermal plume arrives at pumping well and temperature change at pumping well were observed with different pumping/injection rates, hydraulic gradient, and well separation. Thus, we derived adequate well arrangement for efficient GWHP operation.

METHODOLOGY

The numerical model of TRS is based on a finite-difference approximation of the potential flow theory and deals with thermal recycling phenomenon between injection and production wells with groundwater flow (Casasso and Sethi, 2015). In this study, the model was used to estimate the well arrangement for small GWHP systems which can avoid thermal recycling between wells. In the model, injection well was assumed to be downstream. To consider hydrogeological properties of South Korea, the data obtained from the national groundwater monitoring stations (NGMSs) were analyzed. Figure 1 shows the location of 126 NGMSs considered in this study. The monitoring wells were installed at alluvial layers, which consist of clay, sand, gravel, and weathered rock. Groundwater levels are located at average 4.97 m below ground surface. Hydraulic conductivities of the alluvial aquifer ranged from $1.12 \times 10^{-5}$ to $2.34 \times 10^{-1}$ cm/s with geometric mean of $9.56 \times 10^{-4}$ cm/s (Figure 2). The range corresponds to the values of sand or gravel (Domenico and Schwartz, 1990) and volumetric heat capacity of such media varies between 2.2 and 2.8 MJ/m$^3$K (Stauffer et al., 2013). System size for small application was determined to be less than 23.4 kW, based on pumping/injection rates and the statistics of geothermal application (KEA, 2015). Injection temperature was assumed to be 5°C higher than temperature of pumped water, and thus the system operated in the cooling mode. Since the numerical model can simulate only a continuous operation, injection temperature of 5°C higher or lower make no difference in the analysis results. Pumping/injection rates were set to satisfy the system size. The simulation was performed for ten years, and arrival time when thermal plume arrives at pumping well and temperature change at pumping well were observed with different well flow rates, well distances, hydraulic conductivities, and hydraulic gradients. The input parameters used in the analysis are listed in Table 1.

![Location of the national groundwater monitoring stations considered in this study.](image_url)
Table 1. Model input parameters

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Symbol</th>
<th>Unit</th>
<th>Value</th>
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<tbody>
<tr>
<td>Well flow rate</td>
<td>Q</td>
<td>m³/d</td>
<td>20 to 100</td>
</tr>
<tr>
<td>Temperature difference</td>
<td>ΔT</td>
<td>K</td>
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<tr>
<td>Well radius</td>
<td>r</td>
<td>m</td>
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</tr>
<tr>
<td>Hydraulic conductivity</td>
<td>K</td>
<td>m/s</td>
<td>10⁻⁵ to 10⁻³</td>
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<tr>
<td>Effective porosity</td>
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<td>-</td>
<td>0.2</td>
</tr>
<tr>
<td>Hydraulic gradient</td>
<td>i</td>
<td>-</td>
<td>0.001 to 0.01</td>
</tr>
<tr>
<td>Thermal capacity of soil</td>
<td>ρsC₅</td>
<td>MJ/m³K</td>
<td>2.5</td>
</tr>
<tr>
<td>Thermal capacity of water</td>
<td>ρwC₆</td>
<td>MJ/m³K</td>
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</tr>
<tr>
<td>Aquifer thickness</td>
<td>b</td>
<td>m</td>
<td>15</td>
</tr>
</tbody>
</table>

Results

Analysis results are shown in Figures 3-5. Minimum well separation is defined as the distance between pumping and injection wells in which thermal recycling does not occur even with a continuous operation during ten years. Under high flow condition (i = 0.01), the minimum well separation to avoid thermal interferences varied from 2 to 50 m when hydraulic conductivity is larger than 10⁻⁴ m/s. As flow velocity decreases, the required distance increases and is in the range from 10 to 50 m when hydraulic conductivity is larger than 4×10⁻⁴ m/s (Figure 5). Considering the low hydraulic conductivity value reported for some monitoring wells, the necessary distance become too large to apply small GWHP systems when groundwater flow is very low (Figure 5). Therefore, groundwater flow condition play a
significant role in small-scale facilities, and such applications can be limited by available space especially in case of very low flow velocity.

**Figure 3** Minimum well separation according to various pumping/injection rates and hydraulic conductivity under high flow condition (hydraulic gradient $i = 0.01$).
Figure 4  Minimum well separation according to various pumping/injection rates and hydraulic conductivity under medium flow condition (hydraulic gradient $i = 0.005$).

Figure 5  Minimum well separation according to various pumping/injection rates and hydraulic conductivity under low flow condition (hydraulic gradient $i = 0.001$).
CONCLUSION

The distribution of hydraulic conductivity was estimated from the data obtained from 126 NGWSs over South Korea. Based on the estimated hydraulic properties, the well distance to avoid thermal interferences was calculated with different pumping/injection rates and hydraulic gradients. The results indicated that groundwater flow condition is an important parameter in the design of small GWHP systems, and such applications can be limited by available space especially when groundwater flow is very slow (See Figure 5). Further studies are needed to make the guideline for the design of small GWHP systems.

ACKNOWLEDGMENTS

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Carla Montagud
Julio Martos
José Miguel Corberán
Álvaro Montero

ABSTRACT

Ground source heat pump (GSHP) systems for heating and cooling represent an efficient alternative to conventional air source heat pump systems, always provided that they present an efficient design and operation. In this context, the development of energy optimization strategies becomes essential, with the aid of integrated dynamic models of the system, specially the ground source heat exchanger.

In previous works, a single U borehole heat exchanger (BHE) dynamic model, called Borehole-to-Ground (B2G), was developed and experimentally validated. The B2G model is based on the thermal network approach, combined with a vertical discretization of the borehole. However, the thermal properties of the surrounding ground were modelled as an average, constant with depth. For homogeneous type of soils, this assumption might be acceptable but, when considering heterogeneous type of soils, modelling the presence of different layers with different materials could provide more accurate results.

In this work, the B2G model has been adapted in order to account for the effect of a heterogeneous ground profile on the fluid temperature evolution along the borehole depth. Experimental data corresponding to a Thermal Response Test (TRT) performed in a real BHE existing at the Universitat Politècnica de València, were used to validate this new feature of the B2G model. Finally, it is concluded that the model can serve as a ground thermal properties estimation tool.

INTRODUCTION

Ground properties needed for the design of Ground-Source Heat Pump (GSHP) systems are estimated with a standard method, the Thermal Response Test (TRT). After injecting or extracting a constant heat power, the test analyzes the ground thermal response with a few measurements, inlet and outlet temperatures of the fluid at the top of the borehole heat exchanger (BHE) and fluid mass flow rate. Effective ground thermal conductivity, borehole thermal resistance and undisturbed ground temperature are outputs of this test. In this context, models able to predict the BHE thermal performance can contribute to analyze this type of tests and to determine such outputs. In the last years, several approaches have been proposed in order to reproduce the thermal behavior of different BHE configurations (a complete review is reported by Yang, et al. (2010)). In general, models analyzing this data assume that ground
thermophysical properties are homogeneous and isotropic; therefore, this approach is only able to estimate an effective ground thermal conductivity representing an average of the thermal conductivity of the different layers crossed by the perforation. As pointed in Aranzabal, et al. (2016), if a thermal conductivity profile of the ground as a function of depth is to be estimated, two additional inputs are needed; first, a measurement of the borehole temperature profile and, second, an analysis procedure taking into account that the ground is not homogeneous. In Aranzabal, et al. (2016) an analysis procedure, complementing the standard TRT analysis, was presented to estimate the thermal conductivity profile from a temperature profile along the borehole during the test. The analysis procedure was implemented by a 3D Finite Element Model (FEM) in which depth depending thermal conductivity of the subsoil was estimated by fitting simulation results with experimental data. The methodology was evaluated by the recorded temperature profiles throughout a TRT in a BHE monitored facility.

It should be noted that the influence of the different conductivity layers might be relevant for the short-term, and it has less influence in the modelling of the long term behaviour, as pointed out by Lee (2011) who stated that the adoption of an effective ground thermal conductivity and an effective ground volumetric heat capacity for a multi-layer ground determined from a TRT analysis led to very little error in the simulated long term system performance with differences lower than 0.5K. This effect will also be investigated in the present research work.

On the other hand, recent studies highlight the possibility of using dynamic simulation models to assist the determination of the ground effective thermal conductivity. In this context, Pasquier (2015) presented a program designed to analyze thermal response tests by deterministic or stochastic inversion, for single U BHEs with a thermal resistance and capacity model, which aims to foster investigation of new testing strategies with a possible reduction in the tests duration. The present contribution shows an alternative analysis procedure based on the recently developed borehole-to-ground (B2G) model, which is a thermal resistance and capacity model (5C6R), specially addressed to describe short term thermal processes in BHEs (Ruiz-Calvo, et al. (2015)). A new feature was included in the model to consider different ground layers in non homogeneous type of soil. Using the same experimental data analyzed in Aranzabal, et al. (2016), and the thermal conductivity profile estimated in that contribution, the purpose of this analysis is to experimentally validate and check the ability of the B2G model to reproduce the evolution of the measured ground temperature profile. This analysis is a first step towards the use of the B2G model as a complementary method to analyze TRT data in shorter time periods and adapted to non homogeneous type of soils.

**B2G MODEL**

**Model Description**

The Borehole-to-Ground (B2G) dynamic model was developed at the Instituto de Ingeniería Energética (IIE) – Universitat Politècnica de València in order to reproduce the short-term performance of a single U-tube BHE during the daily injection/extraction of heat in an on/off GSHP system operation. It is able to calculate the instantaneous evolution of the fluid temperature taking into account the geometrical characteristics of the BHE and the thermophysical properties of the grout and the surrounding ground. This model has been presented and validated against experimental TRT data from different BHEs located at Universitat Politècnica de València, Spain (De Rosa, et al. 2015) and in Stockholm, Sweden (Ruiz-Calvo, et al. 2015). It has also been validated against experimental data from a GSHP system on/off operation during one month (Ruiz-Calvo, et al. 2016) and one year (Ruiz-Calvo 2015). For this purpose, the B2G model was coupled with the g-function model in order to also predict the long-term response of the BHE.

The BHE is discretized vertically in several divisions; in each z-depth, a 2D thermal network represents the heat transfer between the different parts of the BHE. This thermal network is made of five nodes, each node represents one part of the BHE: $T_1$ represents the downward pipe fluid, $T_2$ represents the upward pipe fluid, the grout of the borehole is divided in two zones ($T_{b1}$ and $T_{b2}$) and the surrounding ground is represented by $T_g$. The nodes are interconnected with thermal resistances that consider the conductive and convective heat transfer between the different parts of the BHE and each node includes the thermal capacitance of its part, representing its thermal inertia. Figure 1 (a) shows the
thermal network of the BHE, where the different nodes and thermal resistances are represented.

Initially, the B2G model was developed to reproduce the instantaneous behavior of the BHE, for this purpose, only the portion of ground directly affected during the desired heat injection/extraction time is considered (around 10 hours for a normal on/off GSHP system operation). However, it is able to reproduce longer periods of injection/extraction of heat if a larger portion of ground is considered (and thus a higher thermal capacity of the ground). In this case, the accuracy of the short-term behavior decreases. In order to consider the corresponding amount of ground, a ground penetration diameter \( (D_{gp}) \) is set and it defines the position of the ground node.

The original B2G model neglected the vertical conduction, although the advection in vertical direction for the fluid nodes was taken into account in the transient energy balance equations (Ruiz-Calvo, et al. 2015). The entire model results in \( n \) thermal networks (\( n \) represents the number of vertical divisions of the BHE). At each depth, 5 thermal capacitances and 6 thermal resistances are considered (a 5C6R-\( n \) model), the thermal properties of the ground, the grout and the pipes are also taken into account. The model can be solved by numerical procedures, solving the system of ordinary differential equations as described in (Ruiz-Calvo, et al. 2015), where the model is described in more detail.

**Adaptation to heterogeneous ground**

Initially, the B2G dynamic model considered a homogeneous ground, so it used an effective ground thermal conductivity and an effective ground volumetric heat capacity if the ground was heterogeneous. This work is focused on the adaptation of the B2G model in order to consider different thermal conductivities in a multi-layer ground. In order to achieve this, an array of ground thermal conductivities depending on the depth is introduced in the model. Furthermore, the vertical conduction in the grout and ground zones is taken into account by means of vertical thermal resistances between the nodes of adjacent thermal networks, as it is shown in Figure 1 (b). The vertical thermal resistance of the grout nodes is described in equation (1). It depends on the vertical thermal conductivity of the grout \( (k_{g,p}) \), the vertical distance between nodes \( (dz) \) and the grout horizontal surface \( (D_b) \). The borehole diameter \( (D_p,e) \) is the external pipe diameter of the U-tube).

\[
R_{vb1} = R_{vb2} = \frac{dz}{\pi \cdot (D_b^2 - 2 \cdot D_p,e^2) \cdot k_{g,p}}
\]  

Equation (1)

Analogously, the vertical thermal resistance of the ground nodes is described in equation (2). In this case, the vertical thermal conductivity of the ground \( (k_{g,p}) \) varies with the depth.

\[
R_{vg}(z) = \frac{dz}{\pi \cdot (D_{gp}^2 - D_b^2) \cdot k_{g,p}(z)}
\]  

Equation (2)
Now, the energy balance equations corresponding to the different nodes of the thermal network are presented in equations (3)-(7).

\[
\frac{\partial T_1(z)}{\partial t} = -v \frac{\partial T_1(z)}{\partial z} - \frac{1}{C_f} \left( \frac{T_1(z) - T_{b1}(z)}{R_{b1}} + \frac{T_1(z) - T_2(z)}{R_{pp}} \right)
\]

(3)

\[
\frac{\partial T_2(z)}{\partial t} = -v \frac{\partial T_2(z)}{\partial z} - \frac{1}{C_f} \left( \frac{T_2(z) - T_{b2}(z)}{R_{b2}} - \frac{T_1(z) - T_2(z)}{R_{pp}} \right)
\]

(4)

\[
C_{b1} \frac{\partial T_{b1}(z)}{\partial t} = \frac{T_1(z) - T_{b1}(z)}{R_{b1}} + \frac{T_{b2}(z) - T_{b1}(z)}{R_{bb}} + \frac{T_g(z) - T_{b1}(z)}{R_{g}} + \frac{T_{b1}(z - 1) - T_{b1}(z)}{R_{vb1}} + \frac{T_{b1}(z + 1) - T_{b1}(z)}{R_{vb1}}
\]

(5)

\[
C_{b2} \frac{\partial T_{b2}(z)}{\partial t} = \frac{T_2(z) - T_{b2}(z)}{R_{b2}} + \frac{T_{b1}(z) - T_{b2}(z)}{R_{bb}} + \frac{T_g(z) - T_{b2}(z)}{R_{g}} + \frac{T_{b2}(z - 1) - T_{b2}(z)}{R_{vb2}} + \frac{T_{b2}(z + 1) - T_{b2}(z)}{R_{vb2}}
\]

(6)

\[
C_g \frac{\partial T_g(z)}{\partial t} = \frac{T_{b1}(z) - T_g(z)}{R_{g}} + \frac{T_{b2}(z) - T_g(z)}{R_{g}} + \frac{T_g(z - 1) - T_g(z)}{R_{vg}} + \frac{T_g(z + 1) - T_g(z)}{R_{vg}}
\]

(7)

The ground penetration diameter that sets the position of the ground nodes is calculated using an effective ground thermal conductivity and volumetric thermal capacity. For this purpose, the heat transfer equation for a region bounded internally by a circular cylinder and constant heat flux in its surface (Carslaw and Jaeger 1959) is used; this equation considers the period of time of heat injection/extraction and the thermal properties of the ground. The ground node \(D_g\) is placed at the mean diameter between the penetration diameter \(D_{pp}\) and the borehole diameter \(D_b\). Regarding the position of the grout nodes \(D_x\), they can be placed between the equivalent diameter of the pipes \(D_{eq}\) and the borehole diameter \(D_b\). To calculate the equivalent diameter, the Pasquier and Marcotte (2012) approach is used.

\[
D_{eq} = D_{p,e} \sqrt{\frac{4W}{\pi D_{p,e}} + 1}
\]

(8)

In this equation, \(W\) represents the shank space between pipes.
THE BOREHOLE HEAT EXCHANGER AND EXPERIMENTAL DATA

At the Universidad Politécnica de Valencia campus there exists a BHE of 40 m depth, 160 mm drill diameter and two geothermal independent pipes, ALB GEROTHERM PE-100 of 40 mm diameter and 29 and 39 m long, respectively. The pipes are disposed with a turn of 90° between them; keeping uniform the distance between the pipes of the geothermal probes with separators of polyethylene distributed every meter depth. A scheme of the facility is shown in Figure 2. Further information about the facility can be found in Aranzabal, et al. (2016).

THERMAL RESPONSE DATA

Same data analyzed in Aranzabal, et al. (2016) are used in this contribution. The TRT started on 9th March 2011 at 11:00 with 1 kW heat injection, using the geothermal pipe of 29 meters deep, and leaving the one of 39 meters filled with water in order to use it as an observer pipe and to measure the temperature profile during the TRT. From the beginning of the TRT, the temperature profile was obtained in one of the 39 meters pipe, which was not used in this TRT for heat exchanging. Details of the measured data can be found in the work by Aranzabal, et al. (2016).

![Figure 2](image.png)

Figure 2: Scheme of the borehole heat exchanger facility, vertical layout on left figure and horizontal section on the right figure (Aranzabal, et al. 2016).

EXPERIMENTAL VALIDATION FOR HETEROGENEOUS GROUND

In order to validate the adaptation of the B2G dynamic model to consider a multi-layer ground, the test conducted in the borehole located at Universitat Politècnica de València (explained in the previous sections) was used. Furthermore, the ground conductivities along the borehole depth calculated in Aranzabal, et al. (2016) were used as inputs in the model. The B2G model has been developed for a single U-tube BHE configuration while the borehole used for the experiments is a double U-tube BHE. However, only one tube is used in the TRT, while the other one is used as an observer tube to measure the temperature evolution along the borehole depth. So, the B2G model is used to calculate the temperatures along the borehole depth on the grout nodes and to compare them with the measured temperatures from the observer tube. The position of the grout nodes has been set based on the grout temperature distribution maps in Esen, et al. (2009) to simulate the position of the observer tube. Regarding the observer tube, a maximum of 2K temperature difference along the depth of the BHE was observed. Therefore, the natural convection flows due to the temperature difference between the top and the end of the BHE are considered negligible. The main parameters adopted
in the simulations are shown in Table 1. The number of vertical nodes that have been used is 117, in order to obtain a good accuracy and a low computational time. The same ground thermal conductivity is considered in the vertical and radial direction for each depth. The mass flow rate and the temperature of the inlet fluid is introduced in the model as an input. The model calculates the temperature of the outlet fluid and the temperature in all the grout nodes for each simulation step.

<table>
<thead>
<tr>
<th>Table 1 Main Parameters Adopted In The Model</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Thermo-physical properties</strong></td>
</tr>
<tr>
<td>Ground thermal conductivity (See Figure 4)</td>
</tr>
<tr>
<td>Grout thermal conductivity</td>
</tr>
<tr>
<td>Ground volumetric thermal capacitance</td>
</tr>
<tr>
<td>Grout volumetric thermal capacitance</td>
</tr>
<tr>
<td>Fluid</td>
</tr>
</tbody>
</table>

In order to compare the results obtained by the B2G model with the experimental measurements, four simulations have been carried out: for an injection time of 2.5 hours (9th March 2011 13:30h), 24 hours (10th March 2011 11:00h), 48 hours (11th March 2011 11:00h), and 72 hours (12th March 2011 11:00h). The TRT results for the simulation of 72 hours of heat injection are presented in Figure 3, considering a homogeneous ground with a single fixed effective conductivity and initial temperature (Figure 3 (a)) and considering a multi-layer ground with different conductivities and initial temperatures (Figure 3 (b)). It can be seen that the outlet temperature calculated by the B2G model is very similar to the experimental one in both cases, although the accuracy in case (b) is slightly higher (the root mean square error of case (a) is 0.16 K and in case (b), it is 0.13 K). The accuracy of the results is lower during the first hours of the simulation due to the fact that it has been used a penetration diameter according to a heat injection of 72 hours, losing accuracy in the first hours of the simulation, because the amount of ground considered (and thus the ground thermal capacity) is too high for the short-term response.

![Figure 3](image1.png)  
**Figure 3** Comparison between the experimental measures and the results calculated by the B2G model for a TRT during 72 hours of heat injection. a) homogeneous ground; b) heterogeneous multi-layer ground.

Figure 4 shows a comparison between the experimental temperature measurements inside the observer tube and the simulated temperature of the grout nodes that has been calculated in the four simulations, as well as the initial temperature profile for these two cases considered (homogeneous ground and heterogeneous multilayer ground). It can be observed that, in case (b), with different ground layers, the evolution of the simulated temperatures is very similar to the experimental ones. However, along the first 3-5 meters, the difference between the simulations and the experimental measurements is higher. This result is due to the fact that the physical phenomena that occur between the ground surface and the top of the BHE are not well known (all the instrumentation equipment is placed in the space between
the BHE and the surface) and have not been modeled. So the simulated temperature evolution along the first meters is not very accurate. Moreover, the temperature difference in the region located at 25 m deep is also higher, this fact may be explained because of groundwater advection effects, as it has been explained in Aranzabal, et al. (2016).

On the other hand, in case (a), the calculated temperature along the borehole depth is almost constant for a given time, because a homogeneous ground has been considered.

**Figure 4**  Comparison between the observer tube temperature profile along the borehole depth and the simulated grout nodes’ temperatures; experimentally measured (marked lines) and calculated by the B2G model (continuous lines). Values at different heat injection times (initial temperature, 2.5 hours, 24 hours, 48 hours and 72 hours). a) homogeneous ground; b) heterogeneous multi-layer ground.

Taking into account the results presented in this contribution, it is concluded that the B2G dynamic model could be a useful tool to assist the calculation of the ground thermal conductivities along the borehole depth from the measured temperatures along the observer tube, by using an estimated initial conductivity and an approximation algorithm, and applying the same methodology as the one explained in Aranzabal, et al. (2016). Thus, leveraging the low computational cost of the B2G dynamic model which allows making a great number of simulations in a short period of time (20 seconds for a 72 hours TRT, 6 seconds for a 24 hours TRT on a modern PC). This way, it would be possible
to obtain an estimation of the ground thermal conductivity and the overall borehole thermal resistance with a reduction in injection times needed to calculate the ground conductivity. This would mean a reduction of the cost of the TRT and a consequent increase in its economic feasibility. On the other hand, the calculation of the ground thermal conductivity layer by layer can help in the future design of BHEs for the same type of ground, as it is possible to identify substratums with higher thermal conductivity and then size the borehole depth taking advantage of the different layers.

CONCLUSIONS

This contribution presents the B2G dynamic model as a tool to assist the TRT analysis. The model is able to reproduce a TRT test with a low computational cost (less than 20 seconds for a 72 hours TRT). A new feature was included in the model to consider different ground layers in non homogeneous type of soil, as well as the vertical conduction between the ground and grout nodes. The model was experimentally validated in two different ways: 1) analyzing the temperature evolution along a 72 hours TRT; and 2) analyzing the evolution of the water temperature profile in a multi-layer heterogeneous ground. In 1) the difference between considering a homogeneous ground with a single effective conductivity and initial temperature or a multi-layer ground with different thermal conductivities and initial temperatures, is analyzed. It is concluded that considering an effective ground thermal conductivity is a good estimation presenting a root mean square error of 0.16 K for a TRT simulation without the need of determining the conductivities layer by layer. In 2) the same effect is analyzed for the water temperature profile along the BHE’s depth. In this case, the influence of considering the conductivity of each ground layer results of a higher relevance, specially in order to identify the layers with highest conductivities and to be able to assist in the design of the BHE’s depth.

In conclusion, the B2G dynamic model is able to reproduce the behavior of a single U-tube BHE in a multi-layer heterogeneous ground with a low computational cost. This means that it could be a helpful tool in order to estimate the thermal conductivity of different layers in a heterogeneous ground and to assist in the design of future BHE’s for a given site.

ACKNOWLEDGMENTS

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NOMENCLATURE

\[ \begin{align*}
C &= \text{Thermal capacitance (J/K)} \\
D &= \text{Diameter (m)} \\
k &= \text{Conductivity (W/(m·K))} \\
R &= \text{Thermal resistance (K/W)} \\
t &= \text{Time (s)} \\
T &= \text{Temperature (°C)} \\
v &= \text{Velocity (m/s)} \\
z &= \text{Borehole depth coordinate (m)}
\end{align*} \]

Subscripts

\[ \begin{align*}
bb &= \text{Borehole node to borehole node} \\
f &= \text{Fluid} \\
b &= \text{Convection} \\
pp &= \text{Pipe to pipe node}
\end{align*} \]
REFERENCES


An investigation of thermo-hydro-geochemical processes in a standing column well intersected by a fracture

F. Eppner  P. Pasquier  P. Baudron

ABSTRACT
Local thermal and chemical conditions may favor minerals’ scaling which can cause undesirable operational problems to standing column well systems. Precipitation may reduce heat exchange in the well and the heat exchanger and increase pressure drops in the systems components, as well as the drawdown in the well. In this paper, a coupled thermo-hydro-geochemical model simulating the operation of a standing column well is used to illustrate the link between the temperature and the reaction rate of calcite. The results demonstrate that bleeding only 10% of the pumped water allows to get rid of calcium ions in solution in the well and reduces substantially the precipitated calcite mass. Our simulations clearly indicate that groundwater flow induced by the bleed mainly occurs in the fracture since significant chemical variations are observed downstream of the fracture. We expect that the findings presented in the paper will help demystify the complex thermo-hydro-geochemical behaviour of standing column wells installed in sedimentary rocks.

INTRODUCTION
Standing column wells (SCWs) present an important energy savings potential in dense urban centers. Although most applications of SCWs in Northeastern United States did not report mineral scaling problems, it is known from past experiences with open-loop systems (Stanaseal et al., 2006; Gunnlaugsson 2012; Ma et al., 2012) that mineral precipitation can impede the operation of the heat exchangers and possibly clog the fractures surrounding a SCW. It is likely that these issues will become more common as more SCWs will be installed in sedimentary environments. Understanding the chemical mechanisms induced by the operation of a SCW in sedimentary rocks is then of primary importance.

Indeed, the chemical composition of groundwater in a SCW undergoes important variations throughout the year. These variations are mainly due to fluctuations of groundwater temperature induced by heat pumps’ operation, CO2 degassing caused by pumping and reinjection of groundwater and/or groundwater supply from a fracture having a different chemical signature or temperature. Operational problems arise in response to the increase in roughness of pipes' surface, to pressure drops and to the drawdown in the SCW, as well as to the decrease in heat transfer with the geological formation and the coolant in the heat exchanger.

A thermo-hydro-geochemical (THG) model coupling groundwater flow, heat transfer and reactive transport of dissolved species has been developed recently (Eppner et al., 2015, 2016; Pasquier et al.; 2016) to investigate the risks of calcite dissolution and precipitation in the vicinity of a SCW. In this article, this THG model is used to explain...
complex interactions between thermal, hydraulic and geochemical processes which may cause precipitation and dissolution of calcite. The beneficial impact of the bleed on the calcite precipitation rate in presence of a fracture is also presented and discussed. In addition, the modification of chemical groundwater composition in the well caused by supply of water from a fracture is illustrated.

**GEOCHEMICAL REACTIONS INVOLVING CALCITE**

Precipitation and dissolution of calcite (CaCO$_3(s)$) in carbonate rocks is quite common. Indeed, this widespread mineral may chemically react with groundwater and release some mobile ions in solution, thus promoting local dissolution of the carbonate rock. Alternatively, if the local conditions in the aquifer promote precipitation, some ions may combine together and create solid calcite precipitates (Brantley et al., 2008). The calcite crystals will grow from a solid nucleus (usually a mineral surface or a solid particle in suspension in water) and reduce the fracture’s aperture or accumulate in areas of low velocity such as horizontal pipes (see Figure 1).

Two types of chemical reactions control the precipitation or dissolution rate of calcite. The first type involves six fast equilibrium reactions describing the reactions taking place in the aqueous phase, while the second type involves three slow reversible kinetic reactions occurring at the surface of solid particles in suspension or minerals. These nine chemical reactions are:

Equilibrium reactions – Fast  
\[ H^+ + OH^- \rightleftharpoons H_2O \]
\[ H^+ + CO_3^{2-} \rightleftharpoons HCO_3^- \]
\[ Ca^{2+} + HCO_3^- \rightleftharpoons CaHCO_3^+ \]
\[ H^+ + HCO_3^- \rightleftharpoons H_2CO_3 \]
\[ Ca^{2+} + CO_3^{2-} \rightleftharpoons CaCO_3(s) \]
\[ H^+ + CaOH^+ \rightleftharpoons Ca^{2+} + H_2O \]

Kinetic reactions - Slow  
\[ CaCO_3(s) + H^+ \xrightleftharpoons[k_4]{k_1} Ca^{2+} + HCO_3^- \]
\[ CaCO_3(s) + H_2CO_3 \xrightleftharpoons[k_{2.1}]{k_2} Ca^{2+} + 2HCO_3^- \]  \hspace{1cm} (1)
\[ CaCO_3(s) + H_2O \xrightleftharpoons[k_3]{k_{-1}} Ca^{2+} + HCO_3^- + OH^- \]
All these reactions are temperature dependent (see Appelo and Postma 1993; Plummer et al., 1978), which has some practical consequences. For instance, an increase of water temperature reduces calcite solubility and promotes its precipitation, which explains why calcite crusts are often found in domestic hot water tanks. Similarly, elevating water temperature decreases CO₂ solubility and promotes its degassing from the water free surface to the atmosphere, which again promotes calcite precipitation. Since the chemical species involved in dissolution/precipitation reactions are relatively mobile in the aqueous phase, the ions can be transported by advection, diffusion and dispersion mechanisms and precipitate anywhere else in the system where the local conditions are favorable to precipitation.

THERMO-HYDRO-GEOCHEMICAL MODEL

This section describes briefly the SCW model used in this study to allow the reader appreciate the results presented hereinafter. More details on the simulation framework, boundary conditions, relation between reaction constants ($k_i, k_{-i}$) and temperature, or connection between fracture' aperture and calcite precipitation can be found in Eppner et al., (2015), Nguyen et al., (2015) and Pasquier et al., (2016).

Governing equations

The SCW model couples three models, namely a groundwater flow model (Eq. 2), a heat transfer model (Eq. 3) and a geochemical model (Eq. 4) supplying the species concentrations (c) over the simulation domain. The temperature is obtained by considering heat transfer by advection and conduction mechanisms, while the species concentrations are obtained by considering that advection, diffusion and dispersion mechanisms drive species transport. Recall that all chemical reactions are temperature dependent. The groundwater flow is simulated by solving the conservation equation and Darcy’s law, as expressed by the following equations:

$$\rho S_s \frac{\partial p}{\partial t} + \nabla \cdot (\rho v) = 0 \quad \text{with} \quad v = -\frac{K}{\rho g} (\nabla p + \rho g \nabla D_v)$$

where $\rho$ is the fluid density, $S_s$ is the storage coefficient, $p$ is the pressure, $t$ is the time, $v$ is Darcy’s velocity, $K$ is the hydraulic conductivity, $g$ is the gravitational acceleration and $D_v$ is the vertical coordinate. Darcy’s velocity is integrated in the heat transfer equation with the aim to define the temperature ($T$) over the entire domain through:

$$\rho C_p \frac{\partial T}{\partial t} + \rho C_p \nabla T \cdot \nabla = \nabla \cdot (\lambda \nabla T)$$

with $\lambda$ and $\rho C_p$ which are the equivalent thermal conductivity and volumetric heat capacity respectively. Finally, $v$ and $T$ are integrated in the reactive transport equation to define the activity of the species at each node of the model by:

$$\varphi \frac{\partial}{\partial t} \Gamma = \nabla \cdot (D \nabla \Gamma) - \nabla \cdot (v \Gamma) + US_r \Gamma^r$$

where $\varphi$ is the ground porosity, $\Gamma$ is the vector of species’ total concentration, $D$ is the diffusion coefficient of the total concentration which includes the molecular diffusion and dispersivity, $U$ is a transformation matrix, $S_r$ is the stoichiometric matrix for the kinetic reactions, $r_k$ is a vector of the reaction rates of the kinetic reactions and ’ is the transpose operator. The matrix and the vectors are described in details in Eppner et al., (2015).

The species involved in the model (see Eq. 1) are grouped in three total concentrations ($\Gamma$), allowing to solve only three transport equations (Eq. 4) instead of nine according to the Tableaux method (Morel and Hering 1993; Eppner et al., 2015). This operation greatly simplifies the simulation burden and reduces calculation time. To simulate
the geochemical reactions and to couple them with transport processes, a system of nine equations and nine unknowns (one for each concentration) has also to be built and solved over the simulation domain. For the first six equations, the following formulation is used (Saaltink et al., 1998; Holzbecher 2012):

\[ S_e \cdot \log \alpha - \log K_{eq} = 0 \]  

where \( S_e \) is the stoichiometric matrix for the equilibrium reactions, \( \alpha \) is the vector of the activity of the species and \( K_{eq} \) is the vector of the equilibrium constants of the equilibrium reactions. To complete the system of nine equations, three additional equations are added for each node of the simulation domain:

\[ U \cdot c - \Gamma = 0 \]  

The kinetic reactions described by the term \( US^*_k r_k \) in Eq. 4 are simulated by the following source term:

\[ US^*_k r_k = \begin{pmatrix}
\hat{R}_1 + \hat{R}_2 + \hat{R}_3 \\
-\hat{R}_1 - \hat{R}_2 - \hat{R}_3 \\
-\hat{R}_1 - \hat{R}_2 - \hat{R}_3 
\end{pmatrix} \]  

where \( \hat{R}_i \) is the reaction rate of the kinetic reactions (Eppner et al., 2015 and Plummer et al., 1978). The matrix \( U \), used in Eq. 6 for equilibrium reactions and in Eq. 7 for kinetic reactions, allows to link both reaction types.

**Boundary conditions and input parameters**

Geometry, boundary conditions and input parameters used to solve Eq. 2 to 7 reflect the materials, geometry (water-filled borehole, raiser pipe, aquifer, fracture) and operation of a typical SCW. Indeed, the model integrates especially the pumping and reinjection operations, the drawdown caused by the bleed (if activated) and the temperature variation induced by the heat pump at the SCW’ inlet defined by \( Q_g/(\rho \cdot c_p \cdot T) \), where \( Q_g \) is the heating or cooling load presented in Figure 2 and \( \dot{V} \) is the pumping rate. It is worth noting that water in the aquifer (beyond the model radius) has a constant temperature throughout the year of about 10 °C. The initial species concentration and the concentrations prescribed along the lateral boundary are at chemical equilibrium and vary with depth and temperature (a \( pH \) of 7 and a \( PCO_2 \) of \( 10^{-1.39} \) atm have been used to define the initial groundwater composition).

Finally, the main thermal and hydraulic parameters integrated in the model are presented in Table 1 below.

**Table 1. Parameters used in the simulations.**

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Unit</th>
<th>Symbol</th>
<th>Fluid (f)</th>
<th>Soil (a)</th>
<th>Pipe (p)</th>
<th>Fracture (fr)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Hydraulic conductivity</td>
<td>m/s</td>
<td>( K )</td>
<td>1000</td>
<td>1.0e-6</td>
<td>1.0e-9</td>
<td>5.26 (initial)</td>
</tr>
<tr>
<td>Pumping rate</td>
<td>l/min</td>
<td>( \dot{V} )</td>
<td>189</td>
<td>-</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>Specific storage</td>
<td>1/m</td>
<td>( S_s )</td>
<td>4.0e-6</td>
<td>5.0e-5</td>
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<td>2.5</td>
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<td>2160</td>
<td>1560</td>
<td>2160-4200*</td>
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<td>1.0e-5</td>
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<td>-</td>
<td>0.1/0</td>
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* The fracture is consisting of soil and water, meaning that this parameter varies between the values applied for the soil and water.

*IGSHPA Conference & Expo 2017*
To illustrate the THG behavior of a SCW intersected by a fracture, the equations and parameters presented previously were integrated in a SCW model of 300 m containing a fracture of 3 mm thick at a depth of 150 m. Two one-year simulations (from January 1 to December 31) were achieved (with a constant bleed of 10 % and without bleed) using the heating and cooling loads illustrated in Figure 2 (a). Notice that negative and positive loads are respectively used for heating and cooling.

ILLUSTRATION OF THG PROCESSES

To illustrate a first THG process, one can see on Figure 2 (b) and (c) the evolution of the SCW inlet and outlet fluid temperature along with the evolution of Ca\(^{2+}\). This figure exemplifies the strong influence of temperature on calcite reaction rate and therefore on Ca\(^{2+}\) concentrations (as well as on the other species involved in Eq. 1). The correlation observed between \(T\) and Ca\(^{2+}\) is easily explained by the fact that as the fluid temperature rises, calcite precipitation is promoted at the well’ entrance, which consumes ions of Ca\(^{2+}\) and CO\(_3^{2-}\) to form CaCO\(_3\).

![Figure 2](image)

Figure 2  Temporal evolution of (a) heating and cooling loads, (b) temperature and (c) Ca\(^{2+}\) concentration at the inlet and outlet of the well, with and without a 10% bleed (B).

The THG solutions over the simulation domain is also instructive. For instance, Figure 3 compares Darcy velocities, streamlines, temperatures and Ca\(^{2+}\) concentration for a simulation time step corresponding to 12 months of operation (December 31\(^{th}\)). In winter, heat extraction in the well cools the surrounding geological formation, whereas a warm zone is observed between 0.5 and 8 m, corresponding to the effect of heat injection during the previous summer. Temperature oscillations greatly influences Ca\(^{2+}\) concentration. Indeed, the hottest areas have a lower Ca\(^{2+}\) concentration, while the coolest zones are calcium-enriched. This behavior is explained by the fact that an increase in temperature promotes the reactions in the direction allowing to decrease calcium concentration. Finally, the distribution of calcium shows that advective transport is not a dominant mechanism in most of the aquifer, but an important one around the fracture.

Figure 4 presents vertical profiles of the temperature, Ca\(^{2+}\) concentration and overall calcite reaction rate \(\dot{R} = \dot{R}_1 + \dot{R}_2 + \dot{R}_3\) in the ascending and descending fluids of the SCW in July and December. Note that a negative overall rate indicates that the rate of precipitation exceeds the rate of dissolution and inversely when the rate is positive. We can observe that without bleed, injection of warm water in the SCW during summer (Fig. 4 (a)) increases precipitation of calcite (see the negative rate on Fig. 4 (c) from \(z=0\) to 100 m) and decreases Ca\(^{2+}\) concentration (Fig. 4 (b)). Around a depth of 100 m, the reaction rate reaches zero and becomes positive between a depth of 100 and 300 m due to the simultaneous drop of temperature and calcium concentration, leading to a state of undersaturation with respect to calcite. Dissolution of calcite is then favored at the base of the well and in the central pipe, increasing slightly calcium concentration. In winter, the opposite behavior is observed and the injection of cold water at the top of the SCW causes dissolution of calcite along the borehole wall until a depth of 150 m (Fig. 4 (d, e, f)), and precipitation along the remaining path.
CONTROLLING SPECIES CONCENTRATION BY BLEED

Bleed activation is a key component of the operation of SCWs as it generates a beneficial groundwater flow toward the well (recall that the far field temperature is constant at about 10 °C). Comparing Fig. 4 (a) and (d) clearly confirms the fact that bleed allows reducing the fluid temperature in cooling mode and to increase it in heating, thus leading to better heat pump performances. From a geochemical perspective, bleeding a fraction of the pumped water also allows to get rid of the ions in solution and attenuates the annual fluctuations of species concentration. This has a direct impact on the calcite reaction rate and on the chemical signature of the water circulating in the SCW. Indeed, the mean precipitation rate observed in July and December (Fig. 4 (c and f)) are closer to zero when the bleed is activated, indicating an environment less reactive and therefore less prone to calcite dissolution and precipitation.

Figure 3  Solutions of (a) the hydraulic model (Darcy velocity and streamlines), (b) the heat transfer model (temperature) and (c) the geochemical model (Ca²⁺ concentration) at the end of the year (December 31st). Notice that only calcium has been presented here but the concentration of the other species is also available.

IMPACT OF THE FRACTURE

It is clear from Figure 4 that groundwater flow induced by the bleed mainly occurs in the fracture since significant variations of the parameters presented are observed in the descending fluid at a depth of 150 m. Indeed, in July the reaction rate of calcite varies abruptly due to the supply of calcium and the temperature decline caused by the groundwater inflow from the fracture (Fig. 4(a, b, c)). Similarly, at the fracture’s depth in December the overall reaction rate varies locally owing to the calcium depleted groundwater inflow. Notice that in December, the Ca²⁺ concentration is relatively high in the descending fluid due to the important rate of dissolution, explaining why the water supplied by the fracture reduces the calcium concentration in the well. It is worth noting that the water below the fracture is closer to chemical equilibrium (\( \dot{r} \) tends to 0) throughout the year due to the supply of water from the aquifer. In this study, a single fracture located at a depth of 150 m was considered but additional work should be done to confirm the generality of this result.

LOCATION OF PRECIPITATION AND DISSOLUTION

The overall reaction rate is useful for understanding the mechanisms leading to precipitation and dissolution of calcite. However, this variable does not easily represent the magnitude of the processes within the well. The cumulative mass of calcite is more meaningful for quantifying and localizing precipitation and dissolution processes. Figure 5 illustrates the cumulative calcite mass after one year between the riser pipe and the geological formation, with and without a continuous 10% bleed. Negative values indicate a dissolved mass, while positive values express a precipitated mass. At the top of the SCW, an accumulation of about 0.45 kg of calcite is observed without bleed.
against 0.2 kg with bleed. In addition, when the bleed is not active, calcite is dissolved at the base of the SCW, which consequently releases Ca$^{2+}$ ions in solution and increases the risk of precipitation elsewhere in the system. The precipitated calcite mass is however relatively small for the size of the well. Indeed, the mass balance after an operation of one year in the SCW annulus is 0.95 kg with bleed and 0.25 kg without bleed.

Notice that in some specific cases, the bleed may increase the rate of precipitation of calcite. For instance, assuming that CaCO$_3$ precipitates on particles in suspension, which will accumulate at the base of well, an operation without bleed is expected to be more advantageous, since the base of the descending fluid is undersaturated with respect to calcite, preventing calcite scaling. Moreover, when CO$_2$ degassing is observed at the top of the SCW, calcite scaling is more important when the bleed is active. Indeed, CO$_2$ supply from the aquifer allows to maintain a high CO$_2$ flux in direction to the atmosphere, significantly increasing the rate of precipitation. Contrariwise, without bleed, the reserve in CO$_2$ decreases in function of time, therefore reducing the negative impact of this phenomenon.

![Graphs showing temperature, Ca$^{2+}$ concentration, and overall rate of reaction of calcite as a function of depth in the ascending and descending fluids with a 10% bleed (in dashed line) and without bleed (in solid line) in July (a, b, c) and December (d, e, f).]

**Figure 4** Profiles of the temperature, the concentration in Ca$^{2+}$ and the overall rate of reaction of calcite as a function of depth in the ascending and descending fluids with a 10% bleed (in dashed line) and without bleed (in solid line) in July (a, b, c) and December (d, e, f).

**PREVENTING GEOCHEMICAL PROBLEMS**

The significative precipitation rate at the SCW inlet suggests that calcite precipitation may occur in the heat pump, heat exchanger, above ground pipes and upper portion of the SCW, which may consequently decrease the heat exchange between the groundwater and the refrigerant, increase pressure drops in the systems components, and possibly the drawdown in the SCW. Therefore, controlling mineral scaling in the mechanical equipments and SCW may prove important. Our observations indicate that a constant 10% bleed helps reducing calcite precipitation in the
SCW and could therefore be used as a simple and effective mitigation measure for the SCW itself. However, if the groundwater bled from the SCW is reinjected in the original aquifer through an injection well (as required in some jurisdictions), it is likely that precipitation will occur in the injection well and surrounding fracture network. Additional works will be required to quantify the clogging risk of the injection well and identify effective mitigation measures such as, for example, use of water treatment system.

**CONCLUSION**

With the aim to control minerals scaling in SCWs, identifying processes leading to fluid sursaturation with respect to a certain mineral is important. This study demonstrates that temperature controls the rate of reaction of calcite and therefore, the chemical signature of the water in the well. However, the bleed can be used to mitigate yearly temperature fluctuations in the well and consequently reduce the accumulation of calcite on borehole walls at the top of the SCW. Moreover, when a fracture is present in the aquifer, groundwater flow induced by the bleed mainly occur in this preferential flow path. Chemical signature of the descending fluid before and after the fracture are therefore significantly different. We hope that this new knowledge favors the installation of SCWs in carbonate rocks. Notice that the results presented in this paper are valid for the parameters fixed in the model. With another initial groundwater composition for instance, the results and conclusions could be somehow different.

**NOMENCLATURE**

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
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<td>$\alpha$</td>
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<td>$\lambda$, $\Phi$, $\Gamma$</td>
<td>Equivalent thermal conductivity (W/(K·m)), vector of species’ total activity (-) and total activities (kg/m³)</td>
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<td>$\rho$, $\rho C_p$</td>
<td>Fluid density (kg/m³) and equivalent volumetric heat capacity (J/K·m³)</td>
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<td>$C_v$</td>
<td>Volumetric heat capacity (kJ/(m³·K))</td>
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<td>Vertical coordinate (m)</td>
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<td>Vector of reactions rates of kinetic reactions (-)</td>
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<td>Storage coefficient (1/Pa)</td>
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<td>$S_k$, $S_e$, $U$</td>
<td>Stoichiometric matrix for kinetic and equilibrium reactions (-) and transformation matrix (-)</td>
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<td>$v$, $\dot{v}$</td>
<td>Darcy velocity (m/s) and pumping rate (l/min)</td>
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REFERENCES


Modelling and Experimental Validation of a Novel Co-Axial Spiral Borehole Heat Exchanger

Antonio Cazorla-Marín
Roon Hylkema
Carla Montagud
José Miguel Corberán
Henk Witte

ABSTRACT
In order to optimize the operation of a ground source heat pump (GSHP) system, the development of dynamic models that integrate all the system components is a key factor. Particularly, the modelling of the ground source heat exchanger and its coupling to the heat pump operation becomes important. Usually, this kind of systems present an on/off operation, which makes it necessary to have an accurate prediction of both the short and long thermal response of the borehole heat exchanger (BHE). In this context, the novel B2G dynamic model was developed and experimentally validated in previous works for a single U-loop BHE.

This work presents the adaptation and experimental validation of the B2G dynamic model to a novel co-axial spiral BHE configuration designed in the framework of a HORIZON 2020 European Project, GEOT€CH (Geothermal Technology for Economic Cooling and Heating). The results show that the B2G approach applied to this specific configuration produces a model that can accurately predict the behavior of the BHE.

INTRODUCTION
During the last years, Ground Source Heat Pump (GSHP) systems have proven to be one of the most efficient systems for heating and cooling in buildings (Luo, et al. 2016), achieving a significant reduction of the energy consumption when compared to conventional heating and cooling systems. For example, in comparison with conventional air-to-water heat pumps, GSHP systems can provide energy savings around 40% of primary energy in the European Mediterranean coast (Urchueguía, et al. 2008).

In order to obtain an efficiency as high as possible, it is important to optimize not only the design of the components but also the integrated system’s operation. For this purpose, a dynamic model of the whole system is a very useful tool, since it allows a detailed prediction of the behavior of the system during its operation. In GSHP systems, the most important component, and also the most expensive, is the Borehole Heat Exchanger (BHE). An accurate dynamic model of this component can greatly help in the prediction and overall optimization of the system behavior, especially in an ON/OFF operation GSHP system. Furthermore, an optimized design of the GSHE is key in order to obtain a good energy efficiency of the system at a reasonable cost. Providing not only energy savings, but also a higher return on investment when used instead of a conventional system. For this reason, the GSHE should not be under-
sized (low cost but lower efficiency) nor over-sized (higher efficiency but high cost).

In this context, several GSHE models have been developed in order to reproduce the thermal behavior of different BHE configurations (a complete review is reported by Yang, et al. 2010). Some of them are focused on the prediction of the long-term response of the surrounding ground and other models able to predict the BHE short term behavior of a single heat exchanger with high accuracy are usually based on FEM technique, or employ very large and refined thermal grids and implicit numerical schemes, so requiring very long computational times and making them less attractive for integrated system simulations. In this context, the B2G dynamic model was developed for a single U-tube BHE configuration. The B2G model is based on the thermal network approach (5C6R), combined with a vertical discretization of the borehole. The model attempts to find the simplest thermal grid for the borehole and nearby surrounding ground, which is able to retain the accuracy for the instantaneous heat transfer along one day with low computation cost (less than 6 seconds for a 24 hour simulation on a modern PC). This model has been implemented in TRNSYS environment (De Rosa, et al. 2015) and validated against experimental data (Ruiz-Calvo, et al. 2015). It is able to predict the short-term behavior of the BHE with high accuracy.

The B2G dynamic model was developed for a single U-tube BHE, but this configuration is not the most efficient from the point of view of the heat transfer due to the interference between the downward pipe and the upward pipe. Especially when the mass flow rate is low and Reynolds numbers drop below 2300 (laminar flow conditions) the thermal resistance of the borehole heat exchanger can be dramatically increased, resulting in a reduced thermal efficiency.

To improve efficiency at low Reynolds numbers and, at the same time, reduce the thermal losses between down-and upflowing channels, a new coaxial heat exchanger, with an insulated inner pipe and a spiral fluid flow path in the outer pipe, has been developed by Geothex BV (http://geothex.nl). This novel heat exchanger will be further developed and optimized within the framework of the GEOTECH European Horizon 2020 project, Geothermal Technology for Economic Cooling and Heating. Preliminary investigations showed a significant increase of efficiency compared to conventional heat exchanger designs (50% lower thermal resistance compared to a U-tube heat exchanger with turbulent flow and over 75% lower thermal resistance compared with a U-tube heat exchanger at laminar flow or compared with a conventional concentric heat exchanger), especially at low Reynolds numbers (Witte 2012).

The B2G model has been adapted to the innovative co-axial configuration with spiral flow path as a part of the GEOTECH project and validated against experimental data from a Thermal Response Test (TRT) carried out in a real BHE located at the Geothex BV facilities in Houten, Netherlands (Witte, et al 2002).

In this work, the B2G model is described as well as the adaptation to the novel co-axial BHE with spiral flow path. The description of the BHE is presented as well as the TRT carried out. The model has been validated against the experimental data from this TRT, and predicts the behavior of the BHE with high accuracy (root mean square error (RMSE) lower than 0.1 K).

**B2G MODEL**

In order to accurately model the dynamic behavior of a single U-tube BHE, the B2G model was developed and presented previously in (Ruiz-Calvo, et al. 2015) and (De Rosa, et al. 2015). This model is able to reproduce with high accuracy the short-term behavior of the BHE in terms of water temperature throughout the pipe. To reduce computational time, only the portion of surrounding ground directly affected by the considered heat injection/extraction period is taken into account (normally around 10-15 hours for a GSHP system that switches off during the night).

The BHE is discretized vertically in \( n \) divisions and, in each borehole depth, a 2D thermal network represents the radial heat transfer. The thermal network consists of 5 nodes connected by 6 thermal resistances. Each node represents one of the parts of the BHE: the downward and upward fluid inside the pipe, the borehole backfilling and the surrounding ground. Each node also includes a thermal capacitance, taking into account the thermal inertia of each part.

Although the vertical conduction is neglected, for the fluid nodes, the advection in vertical direction has been taken into account in the transient energy balance equations. The entire model consists of a 5C6R-n model (5 thermal
capacitances and 6 thermal resistances in each \( z \)-depth and \( n \) vertical divisions of the BHE), this is a system of ordinary differential equations that can be solved using standard numerical procedures, as described in (Ruiz-Calvo, et al. 2015).

The B2G model has been validated against experimental data from a real borehole located in Stockholm, Sweden, using different step-tests (Ruiz-Calvo, et al. 2015). Furthermore, it has been validated against experimental data from a real borehole located at Universitat Politècnica de València, Spain, under different operation conditions: a step-test and the normal ON/OFF operation of the GSHP system in which the borehole is installed (De Rosa, et al. 2015).

**ADAPTATION OF THE B2G MODEL TO THE NEW CONFIGURATION**

**Model Description**

The B2G dynamic model has been adapted to the new co-axial configuration with spiral flow path. For this purpose, the thermal network has been modified, taking into account the different parts of the new BHE configuration: the fluid in the inner pipe is represented by \( T_i \), the fluid in the outer pipe is represented by \( T_o \), the grout is represented by \( T_b \) and the surrounding ground is represented by two nodes, \( T_{g1} \) and \( T_{g2} \). The reason for considering two ground nodes, instead of only one, is to obtain a higher accuracy both on the short-term and the mid-term behavior. The first ground node is located in order to consider a short period of heat injection/extraction (for example 1 hour), while the second considers a larger period of injection/extraction of time (for example 15 hours). These ground nodes’ positions are represented by the ground penetration diameters \( D_{gp1} \) and \( D_{gp2} \), respectively and are calculated according to the equation for a region bounded internally by a circular cylinder and constant heat flux on its surface (Carslaw and Jaeger 1959). It takes into account the thermal properties of the ground and the injection/extraction period of time. Figure 1(a) shows the thermal network on the borehole layout.

The BHE is discretized vertically from the top to the bottom in \( n \) 2D thermal networks, each thermal network has five nodes with their respective thermal capacitances and four thermal resistances. Furthermore, the vertical conduction along the borehole depth is considered on the grout and the ground. So, each thermal network is connected to its adjacent thermal networks via vertical thermal resistances as it is shown in Figure 1(b).

![Figure 1](image)

**Figure 1** Thermal network of the coaxial configuration model: a) borehole layout; b) vertical discretization.

The energy balance equations for the different nodes are described in the equations (1)-(5), the vertical advection in the fluid nodes is taken into account and the vertical conduction between grout and ground nodes. The velocity of
the fluid in the inner pipe and the outer pipe is represented by $v_i$ and $v_o$, respectively.

$$\frac{\partial T_i(z)}{\partial t} = v_i \frac{\partial T_i(z)}{\partial z} - \frac{1}{C_i} \left( \frac{T_i(z) - T_o(z)}{R_{io}} \right) \quad (1)$$

$$\frac{\partial T_o(z)}{\partial t} = -v_o \frac{\partial T_o(z)}{\partial z} - \frac{1}{C_o} \left( \frac{T_o(z) - T_i(z)}{R_{io}} + \frac{T_o(z) - T_b(z)}{R_{ob}} \right) \quad (2)$$

$$C_b \frac{\partial T_b(z)}{\partial t} = \frac{T_o(z) - T_b(z)}{R_{ob}} + \frac{T_{g1}(z) - T_b(z)}{R_{bg1}} + \frac{T_b(z - 1) - T_b(z)}{R_{vb}} + \frac{T_b(z + 1) - T_b(z)}{R_{vb}} \quad (3)$$

$$C_{g1} \frac{\partial T_{g1}(z)}{\partial t} = \frac{T_b(z) - T_{g1}(z)}{R_{bg1}} + \frac{T_{g2}(z) - T_{g1}(z)}{R_{g1g2}} + \frac{T_{g1}(z - 1) - T_{g1}(z)}{R_{vg1}} + \frac{T_{g1}(z + 1) - T_{g1}(z)}{R_{vg1}} \quad (4)$$

$$C_{g2} \frac{\partial T_{g2}(z)}{\partial t} = \frac{T_{g1}(z) - T_{g2}(z)}{R_{g1g2}} + \frac{T_{g2}(z - 1) - T_{g2}(z)}{R_{vg2}} + \frac{T_{g2}(z + 1) - T_{g2}(z)}{R_{vg2}} \quad (5)$$

Regarding the spiral flow path inside the outer pipe, an equivalent section and an equivalent hydraulic diameter in the calculation of the hydraulic and thermodynamic properties are considered. The numerical resolution of the entire model is analogous to the resolution for the single U-tube model.

**Parameter Calculation**

The main parameters of the model are the thermal capacitances and the thermal resistances, which can be determined taking into account the thermo-physical properties and the geometrical characteristics of the borehole, similarly to the single U-tube B2G model (Ruiz-Calvo, et al. 2015).

**Thermal Capacitances** ($C$) are calculated with the volumetric thermal capacitance ($c$) and the volume of the zone in each vertical division ($dz$). The ground and grout capacitances are calculated according to the equations in (6), where $D_b$ represents the borehole diameter and $D_{eo}$ represents the external diameter of the outer pipe.

$$C_{g1} = \frac{\pi}{4} \left( D_{gp1}^2 - D_b^2 \right) c_{g1} dz \quad ; \quad C_{g2} = \frac{\pi}{4} \left( D_{gp2}^2 - D_{gp1}^2 \right) c_{g2} dz \quad ; \quad C_b = \frac{\pi}{4} \left( D_b^2 - D_{eo}^2 \right) c_b dz \quad (6)$$

The thermal capacitance of the fluid nodes is calculated taking into account the heat capacity ($C_p$), the fluid density ($\rho$) and the volume, according to the equation (7), where $D_{el}$ represents the inner diameter of the inner pipe, $D_{el}$ represents the inner diameter of the outer pipe and $D_{co}$ represents the outer diameter of the inner pipe.

$$C_i = \frac{\pi}{4} D_{el}^2 c_{pi} \rho_i dz \quad ; \quad C_o = \frac{\pi}{4} (D_{el}^2 - D_{co}^2) C_{p,o} \rho_o dz \quad (7)$$

**Thermal resistances** are calculated as an addition of conductive and convective cylindrical thermal resistances. The nodes are located at an equivalent diameter in order to calculate the conductive resistance. The equivalent diameter is calculated as a mean diameter of the zone according to the equation (8), where $D_x$ is the borehole node diameter, $D_{g1}$ corresponds to the short-term node diameter and $D_{g2}$ corresponds to the mid-term node diameter.
\[ D_x = \frac{D_b + D_{eo}}{2} \quad ; \quad D_{g1} = \frac{D_{gp1} + D_b}{2} \quad ; \quad D_{g2} = \frac{D_{gp2} + D_{gp1}}{2} \]  

The convective thermal resistance is calculated using the mean convective heat transfer coefficient \( (h) \) of the fluid in the inner pipe \( (h_i) \) and in the outer pipe \( (h_o) \) according to the equation (9). The Nusselt number \( (Nu) \) is calculated depending on the flow regime (e.g. (Gnielinski 2010)), and \( k \) represents the thermal conductivity of the fluid.

\[ h = \frac{Nu \cdot k}{D} \]  

For the inner pipe, the internal diameter is considered; for the outer pipe, it is considered an equivalent hydraulic diameter, taking into account the spiral flow path.

On the other hand, the conductive thermal resistances are calculated taking into account the conductivities of the inner pipe \( (k_{ip}) \), the outer pipe \( (k_{op}) \), the grout \( (k_b) \) and the ground \( (k_g) \). The total thermal resistances between the different nodes in the thermal network are described in the equations (10)-(13).

\[ R_{io} = \frac{1}{\pi D_{ei} \cdot dz \cdot h_i} + \frac{\ln(D_{eo}/D_{ei})}{2 \pi k_{ip} \cdot dz} + \frac{1}{\pi D_{eo} \cdot dz \cdot h_o} \]  

\[ R_{ob} = \frac{1}{\pi D_{ei} \cdot dz \cdot h_o} + \frac{\ln(D_{eo}/D_{ei})}{2 \pi k_{op} \cdot dz} + \frac{\ln(D_x/D_{eo})}{2 \pi k_b \cdot dz} \]  

\[ R_{bg1} = \frac{\ln(D_b/D_x)}{2 \pi k_b \cdot dz} + \frac{\ln(D_{gx}/D_b)}{2 \pi k_g \cdot dz} \]  

\[ R_{g1g2} = \frac{\ln(D_{g2}/D_{g1})}{2 \pi k_g \cdot dz} \]  

Regarding the vertical thermal resistances between nodes of adjacent thermal networks, they are calculated according to the equations in (14), depending on the thermal conductivity, the vertical distance between nodes \( (dz) \) and the annulus surface. It is considered the same thermal conductivity for the vertical and radial direction.

\[ R_{vb} = \frac{dz}{\pi \cdot (D_b^2 - D_{eo}^2) \cdot k_b} \quad ; \quad R_{vg1} = \frac{dz}{\pi \cdot (D_{g1}^2 - D_{gp1}^2) \cdot k_g} \quad ; \quad R_{vg2} = \frac{dz}{\pi \cdot (D_{g2}^2 - D_{gp2}^2) \cdot k_g} \]  

**EXPERIMENTAL VALIDATION**

**Geothex® Borehole Heat Exchanger**

The main innovations of the Geothex® heat exchanger consist of an insulated inner pipe to minimize heat loss between the inner and outer flow channel and spiralling vanes in the annular space to enhance heat transfer, especially at low Reynolds numbers. As can be seen in Figures 1(a) and 2, the edge of the vane and the inner wall of the outer pipe are not in full contact. In fact, there is a gap of about 3.25 mm on average. Although the vanes will touch the inner wall of the outer pipe at different places, the spiral flow path is not completely closed. The reason for the existence of this gap is twofold. First of all, manufacturing tolerances of the special inner pipe material are not as high as in conventional extruded pipes. Secondly, and perhaps more importantly, if the fit would be very tight it would be very difficult to insert...
the inner pipe in the outer pipe due to friction, especially when the length of the heat exchanger increases.

Fluid flowing through the gap may generate a local higher turbulence and hence an increase in the heat transfer coefficient from the fluid in the outer pipe to the grout/ground. In the following two scenarios, one without gap flow and one with gap flow, will be considered.

![Figure 2](image_url)  
**Figure 2**  Gap between the spiral rib and the outer pipe wall

**Validation of the model**

The new model has been implemented as a TRNSYS type where the geometrical characteristics and the thermal properties are set as parameters. The temperature and the mass flow rate of the inlet fluid are introduced as inputs and the model calculates the outlet temperature and the heat transferred to the surrounding ground.

In order to validate the model, data from a TRT carried out using the BHE described in the previous section is used. The inlet temperature and mass flow rate are introduced, and the calculated outlet temperature is compared to experimental measurements. The temperature measurements during the TRT were carried out with two PT100 sensors, calibrated in situ with a standard deviation of the temperature measured of 0.028K. The error of the flow measurement is less than 0.2% of the flow rate. A complete description of the experimental setup and error analysis is given in Witte (2012). The difference between the simulated and the experimental outlet temperature is analyzed and the root mean square error (RMSE) is calculated. The amount of heat transferred to the ground is also compared. For the simulation, a test length of 15 hours is used (which is a common daily operation schedule for ON/OFF GSHP systems that switch off at night) and the TRT data is introduced in intervals of 1 minute. Regarding the penetration diameters, the short-term ground node, $D_{g1}$, is placed according to a heat extraction time of 1 hour; and the mid-term ground node, $D_{g2}$, according to 15 hours of heat extraction. This means a penetration diameter $D_{gp1} = 0.3\ m$ and $D_{gp2} = 0.7\ m$ for the type of soil considered according to the equation for a region bounded internally by a circular cylinder and constant heat flux in its surface presented by Carslaw and Jaeger (1959). Independency from the parameter $D_{gp1}$ was checked and it was concluded that for the short term behavior of the water along the BHE, the higher the penetration depth for the first ground node $D_{gp1}$, the higher the RMSE. So, for the TRT analyzed in this work, $D_{gp2}$ should be determined for the total injection period, and $D_{gp1}$ should be set to injection periods lower than or equal to 1 hour. The number of vertical divisions adopted is 150.

<table>
<thead>
<tr>
<th>Thermo-physical properties</th>
<th>Geometrical characteristics</th>
</tr>
</thead>
<tbody>
<tr>
<td>Inner pipe conductivity</td>
<td>0.20 W/m·K</td>
</tr>
<tr>
<td>Outer pipe conductivity</td>
<td>0.42 W/m·K</td>
</tr>
<tr>
<td>Ground thermal conductivity</td>
<td>2.13 W/m·K</td>
</tr>
<tr>
<td>Grout thermal conductivity</td>
<td>1.56 W/m·K</td>
</tr>
<tr>
<td>Ground volumetric thermal capacitance</td>
<td>2410 kJ/m³·K</td>
</tr>
<tr>
<td>Grout volumetric thermal capacitance</td>
<td>3500 kJ/m³·K</td>
</tr>
<tr>
<td>Percentage of propylene-glycol in the fluid</td>
<td>20 % (vol.)</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Thermophysical properties</th>
<th>Geometrical characteristics</th>
</tr>
</thead>
<tbody>
<tr>
<td>Inner diameter of the inner pipe</td>
<td>0.0285 m</td>
</tr>
<tr>
<td>Outer diameter of the inner pipe</td>
<td>0.0445 m</td>
</tr>
<tr>
<td>Inner diameter of the outer pipe</td>
<td>0.057 m</td>
</tr>
<tr>
<td>Outer diameter of the outer pipe</td>
<td>0.063 m</td>
</tr>
<tr>
<td>Angle of the spiral rib</td>
<td>85.7°</td>
</tr>
</tbody>
</table>

A sensitivity analysis was carried out for the grout conductivity and it was concluded that assuming a ±25%
uncertainty in the determination of the grout conductivity could produce up to a 70% increase in the RMSE (RMSE=0.049+0.034K). Regarding the spiral flow along the outer pipe of the BHE, two scenarios have been considered: a) all the fluid is following the spiral path along the outer pipe; b) there exists a part of the fluid that flows through the gap between the rib and the outer pipe wall, this phenomenon generates a higher turbulence and hence, the convective heat transfer is increased. In order to account for this phenomenon in the model, an enhancement factor in the mean convective heat transfer coefficient from the fluid along the outer pipe to the grout is defined. The heat transfer coefficient from the fluid to the inner pipe is not modified. The enhancement factor has been chosen to accurately fit the simulated results with the experimental data. In this case, the value is 1.5 (the convective heat transfer coefficient is increased by 50%).

RESULTS

For the two cases considered (with and without gapflow), if the enhancement factor is set to 1, no gapflow is assumed; and if it is set to a value of 1.5, then enhanced heat transfer due to gapflow is assumed. In Figure 3 the calculated and experimentally measured BHE outlet temperature as well as the difference between the calculated and the measured outlet temperatures are depicted for both cases. It can be seen that, in both cases the model reproduces the measured values accurately, presenting the smallest error in the second case (Figure 3b) with enhanced heat transfer. The RMSE in case a) is 0.095 K and in case b), it is 0.049 K. The highest temperature difference is 0.308 K in case a) and 0.212 K in case b), and it takes place during the first half hour of the TRT.

![Figure 3](image-url)  
**Figure 3** Comparison between the experimental outlet temperature and the calculated by the B2G model. a) fluid following the spiral path, b) Part of the fluid through the gap, enhancement factor=1.5.
The heat transferred to the ground and the deviations between the experimental and simulation results are summarized in Table 2 for both cases.

<table>
<thead>
<tr>
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</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>43022.6 kJ</td>
<td>41487.7 kJ</td>
<td>-3.57%</td>
<td>43097.2 kJ</td>
<td>0.17%</td>
</tr>
</tbody>
</table>

Other important parameters of the model are shown in Table 3. The mass flow rate corresponds to the mean value introduced in the model from the experimental data; the other parameters are calculated by the model.

<table>
<thead>
<tr>
<th></th>
<th>Hydraulic parameters</th>
<th>Convective heat transfer coefficient</th>
</tr>
</thead>
<tbody>
<tr>
<td>Volumetric flow rate</td>
<td>0.307 ± 0.018 (m³/hr)</td>
<td>Inner pipe 79 W/(m²·K)</td>
</tr>
<tr>
<td>Reynolds number (inner pipe)</td>
<td>1240</td>
<td>Outer pipe (case a) 149 W/(m²·K)</td>
</tr>
<tr>
<td>Reynolds number (outer pipe)</td>
<td>321</td>
<td>Outer pipe (case b) 224 W/(m²·K)</td>
</tr>
</tbody>
</table>

CONCLUSION

The B2G dynamic model has proven to be an accurate model for the prediction of the short term behavior of a single U-tube BHE. In this paper it is shown how it can be adapted to a novel co-axial configuration with spiral flow path. The adapted model has been validated against experimental data from a TRT carried out at the Geothex BV facilities. In order to validate the model, two scenarios have been studied: the first one considers that all the fluid follows the spiral path along the outer pipe, the second one considers that part of the fluid flows through the gap between the rib and the pipe wall, generating a higher turbulence and hence, enhancing the convective heat transfer. In order to take into account this phenomenon, the convective heat transfer coefficient between the fluid along the outer pipe and the grout has been experimentally fitted and an enhancement factor of 1.5 in the heat transfer coefficient was obtained (50% better heat transfer). Both cases generate accurate results, with a RMSE lower than 0.1 K (a typical tolerance for the temperature sensors), although the precision of the second case (enhancement factor=1.5) is higher (RMSE<0.05 K). To summarize, the B2G dynamic model adapted to the new co-axial configuration is able to predict the short term behavior of the new coaxial BHE with high accuracy, especially when it is considered that there exists an increase in the convective heat transfer due to the fact that a small part of the fluid along the outer pipe flows through the gap.

ACKNOWLEDGMENTS

The present work has been supported by the European Community Horizon 2020 Program for European Research and Technological Development (2014-2020) inside the framework of the project 656889 – GEOTeCH (Geothermal Technology for Economic Cooling and Heating) and by the Generalitat Valenciana inside the program “Ayudas para la contratación de personal investigador en formación de carácter predoctoral (ACIF/2016/131)”. 
REFERENCES


Assessment of effective borehole thermal resistance from operational data

Olga Mikhaylova  Ian W. Johnston  Guillermo A. Narsilio

ABSTRACT

Ground source heat pump (GSHP) systems use the ground as a source of sustainable thermal energy for heating and cooling of buildings. Efficient design of the ground heat exchangers (GHEs) for these systems is important so that long-term operation is adequate, efficient and cost-effective. Several design methods have been developed to size GHEs, and many of these methods, including the widely used ASHRAE method, use an effective borehole thermal resistance to model thermal processes in boreholes. A correct estimation of this parameter is crucial for an adequate sizing of borehole GHEs. This study estimates an experimental effective borehole thermal resistance of the borehole GHEs of an operating GSHP system based on monitoring data collected during the Elizabeth Blackburn School of Sciences full-scale shallow geothermal operational study in Melbourne, Australia. The experimental resistance is compared with the resistances predicted using several analytical and numerical methods. It was found that the experimental resistance can be significantly different from the resistances predicted by these other methods. The paper discusses possible reasons for such differences.

INTRODUCTION

For adequate design of ground heat exchangers (GHEs), their thermal performance has to be predicted for the expected thermal loads applied to the ground over the lifetime of a ground source heat pump (GSHP) system. Classic analytical models, such as infinite line source, finite line source or infinite cylindrical source models, estimate GHE wall temperatures at a particular time of GHE operation (Li and Lai, 2015). To calculate temperatures of the fluid circulating in the GHEs, thermal fluxes from the GHE walls to the fluid have to be estimated or assigned. In most borehole models, heat transfer inside GHEs is assumed to be steady-flux and thermal capacity of GHEs is not taken into consideration (see an extensive literature review on this topic in Shirazi and Bernier (2013)). Considering this, thermal processes inside borehole GHEs are often described by steady-state borehole thermal resistances. The borehole thermal resistances that consider thermal short-circuiting between tubes of GHEs are referred to as the effective borehole thermal resistances, \( R^* \) (Hellström 1991; Beier and Spitler 2016; Spitler, et al. 2016).

For relatively short boreholes of depths less than about 100 m, short-circuiting does not affect the GHE resistance much (Spitler, et al. 2016). In such cases, \( R^* \) can be estimated as a parameter that relates the mean fluid temperature of the GHE, \( T_m \), to its mean borehole wall temperature, \( T_b \), through a constant unit heat transfer rate applied to the borehole and averaged along its depth, \( q_b \) (for example, Lamarche, et al. (2010)) as shown in Equation 1.

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\[ T_m = T_b + R^* b q_b \] (1)

In practice, \( R^* b \) can be evaluated by conducting an in-situ thermal response test (TRT) on a pilot GHE (Spitler and Gehlin, 2015). If a TRT is not performed on site, the design \( R^* b \) can be estimated analytically or numerically using anticipated GHE geometry and grout and ground thermal properties. When models are used, thermal properties of the ground and grout as well as geometry of GHEs have to be well predicted for an accurate estimation of \( R^* b \) values. In addition, analytical and numerical solutions are built on certain assumptions, so the accuracy of \( R^* b \) estimated by these methods can be affected by these assumptions.

This study estimates experimental effective borehole thermal resistance \( R^* b \) of an operational GSHP system. The monitoring data collected for the full-scale GSHP system of the Elizabeth Blackburn School of Sciences in Melbourne, Australia was used for the estimations. The \( R^* b \) value determined is compared with the values of borehole thermal resistances \( R_b \) calculated analytically and numerically. The comparison shows that the values of \( R_b \) estimated by models can be significantly different from the value of \( R^* b \) estimated experimentally. Probable reasons of these differences are discussed herein.

### EXPERIMENTAL EFFECTIVE BOREHOLE THERMAL RESISTANCE

A 120 kW (34.1 tons) GSHP system was installed in the 1,500 m² (16,146 ft²) two-storey Elizabeth Blackburn School of Sciences (EBSS) in Melbourne, Australia to provide heating and cooling energy for the building. The system has twenty-eight 50 m (164 ft) deep double U-loop borehole GHEs installed under and around the building footprint. Based on the continuous core samples collected from the site, the site is underlain by effectively intact and imperious Silurian mudstone from around 1.5 m below the ground surface. The system was instrumented to monitor its performance and ground thermal responses to the GHEs. The system commenced operation in March 2014. More details about the set-up of this operational study and instrumentation can be found in Mikhaylova, et al. (2015). The operational performance of the system has been recorded continuously at 3-minute time intervals since the beginning of its operation. More than 2 years of this data has been collected.

In this study, a line of seven GHEs connected in parallel was selected for the analysis. A general view and a plan view cross-section of the GHEs of this line are shown in Figure 1. The GHEs of the line were installed at least 5 m (16.4 ft) apart from each other and the GHEs of neighbouring lines. The ground temperatures observed suggest that there was no significant thermal interference between the GHEs of the selected line during the first 2 years of the system operation. In the estimations of \( R^* b \) of the GHEs selected, the first 2 years of the monitoring data was considered and the individual GHEs in the line were treated as stand-alone GHEs.

The values of ground and grout thermal properties were measured in a laboratory. In the calculations, the ground and grout thermal conductivity and diffusivity were taken as average values along the depths of the GHEs. Table 1 summarises the thermal properties of the ground and grout and the “as designed” geometry of the GHEs. It should be noted that the actual geometry of the GHEs could be different from the “as designed” geometry due to installation processes. While extensive local experience with the Silurian mudstone suggests that changes in the borehole diameter are unlikely, as will be addressed later, the positions of the U-loops inside the boreholes can change significantly.
Figure 1  GHEs: a) A GHE being installed; b) A plan view cross-section of a GHE.

<table>
<thead>
<tr>
<th>Table 1.  Properties and geometry of the ground and GHEs</th>
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<tbody>
<tr>
<td><strong>Parameter</strong></td>
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<tr>
<td>Ground thermal conductivity, ( k )</td>
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<td></td>
</tr>
<tr>
<td></td>
</tr>
<tr>
<td>Ground thermal diffusivity, ( \alpha )</td>
</tr>
<tr>
<td>Grout thermal conductivity, ( k_{\text{grout}} )</td>
</tr>
<tr>
<td></td>
</tr>
<tr>
<td>Grout thermal diffusivity, ( \alpha_{\text{grout}} )</td>
</tr>
<tr>
<td></td>
</tr>
<tr>
<td>U-loop thermal conductivity, ( k_{\text{pipe}} )</td>
</tr>
<tr>
<td></td>
</tr>
<tr>
<td>Radius of borehole, ( r_{\text{bore}} )</td>
</tr>
<tr>
<td></td>
</tr>
<tr>
<td>External radius of U-loop pipes, ( \rho_{\text{ext}} )</td>
</tr>
<tr>
<td></td>
</tr>
<tr>
<td>Internal radius of U-loop pipes, ( \rho_{\text{in}} )</td>
</tr>
<tr>
<td></td>
</tr>
<tr>
<td>Centre-to-centre distance between U-loops, ( L_u )</td>
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<tr>
<td></td>
</tr>
<tr>
<td>Internal convection coefficient, ( h_{\text{conv}} )</td>
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</table>

Since the GHEs of relatively short depths of 50 m are under investigation, Equation 1 is used to estimate effective borehole thermal resistances. To do so, the experimental effective borehole thermal resistance \( R_{\text{be},i} \) at any particular time step \( i \) was estimated from the general relationship between the unit power \( q_{b,i} \) applied to the GHE at this time step and the difference between the mean fluid temperature \( T_{m,i} \) and the mean borehole wall temperature \( T_{b,i} \) at the same time step as

\[
R_{\text{be},i} = \frac{T_{m,i} - T_{b,i}}{q_{b,i}}
\]

(2)
The values of $R^{*}_{be,i}$ were calculated at 3-minute time steps for the times when the heat pumps were running. To do so, at each time step, the $T_{b,i}$ values were calculated considering a 4-month ground thermal load history aggregated into three constant power pulses and ground thermal resistances to these pulses. Considering this and Equation 2, $R_{be,i}$ was calculated as

$$
R^{*}_{be,i} = \frac{T_{m,i} - T_g - q_{5h,i}R_{5h} - q_{720h,i}R_{720h} - q_{2160h,i}R_{2160h}}{q_{5h,i}}
$$

(3)

where $R_{5h}$, $R_{720h}$ and $R_{2160h}$ are the ground thermal resistances to the $q_{5h,i}$, $q_{720h,i}$ and $q_{2160h,i}$ thermal pulses and $T_g$ is the initial undisturbed ground temperature measured on the site. This particular selection of the pulse durations is in line with some common GHE sizing guidelines (for example, ASHRAE (2011)) in regards to the selection of immediate and intermediate ground thermal pulse durations. It should be noted that this is one of many possible aggregations of ground thermal loads for such calculations. The infinite line source model (see, for example, Li and Lai (2015)) was used to calculate the resistances in Equation 3 using the ground and grout thermal properties measured in the laboratory (Table 1).

![Figure 2](image-url)  
Figure 2: An estimation of $R^{*}_{be,i}$ and $q_{5h,i}$ for 11 February 2015: a) the parameters are given in SI units and b) the parameters are given in US customary units.

For the $R^{*}_{be}$ analyses, the days when the highest daily energy was applied to the GHEs were selected to minimise ground load averaging errors in estimations of $q_{5h,i}$. During these days, the heat pumps tended to work continuously for long periods applying close to constant ground power to the GHEs. In total, eleven such days were selected. As an example, Figure 2 presents $q_{5h,i}$ and $R^{*}_{be,i}$ of the GHEs estimated for 11 November 2015. The $R^{*}_{be,i}$ values estimated for all eleven days selected were plotted together in Figure 3. In Figures 2 and 3, $R^{*}_{be,i}$ is presented at two scales to show changes in this parameter over time more clearly.
From Figures 2 and 3, the initial high values of $R_{be,i}$ estimated for the periods from 7:00 to around 15:00 indicate that the thermal absorption of the grout, the U-loop pipes and the water influenced the thermal fluxes inside the GHEs at the beginning of daily heat pump operations. However, it should be noted that the initial values of $R_{be,i}$ estimated at each day can be affected by the load averaging errors. This is likely the case because, firstly, during the periods from 7:00 to 12:00 the $q_{5h,i}$ values represent load history of the times before 7:00 when no power was applied to the GHEs. Secondly, during the first few hours of daily operation, the applied ground thermal power fluctuated significantly because of relatively moderate building cooling power demands at these times. Therefore, the values of the $R_{be,i}$ presented here for the periods from 7:00 to around 15:00 should be considered with caution.

For the above reasons, the values of $R_{be,i}$ estimated at periods from 15:00 to 18:00 only were used for the estimation of $R_{be}$ of the GHEs. During these periods, when the values of $R_{be,i}$ become nearly stable, the thermal processes inside the GHEs reach quasi steady-flux conditions. Such nearly stable values of $R_{be,i}$ represent the resistance of the borehole GHEs to thermal power applied to the ground through them as defined by Equation 1 and are considered as effective borehole thermal resistances $R_{be}$ determined experimentally. From Figure 3, between the times of 15:00 to 18:00 of the eleven days selected, the $R_{be,i}$ values of the GHEs fluctuated around the average value of 0.05 m·K/W (0.085 h·ft·°F/Btu) level. Hence, the representative value of the experimental effective borehole thermal resistance of the GHEs is estimated as $R_{be} = 0.05$ m·K/W (0.085 h·ft·°F/Btu). It should be noted that the estimated values of $R_{be,i}$ plotted in Figure 3 can be affected by measurement errors similar to the measurement errors in TRT results (for an example of a TRT measurement error estimation see Witte (2013)). The measurement errors in $R_{be,i}$ values obtained in this study will be addressed in future publications.

**ANALYTICAL AND NUMERICAL BOREHOLE THERMAL RESISTANCES**

In this section, the borehole thermal resistances of the GHEs of the EBSS GSHP system were estimated using analytical and numerical models. In these estimations, $R_0$ was calculated as
\[ R_b = R_{\text{grout}} + \frac{R_p + R_{\text{conv}}}{4} \]  

where \( R_{\text{grout}} \) is the grout resistance, \( R_p \) is the conduction resistance for each tube of the U-loop and \( R_{\text{conv}} \) is the convection resistance inside each tube of the U-loop. \( R_p = 0.080 \text{ mK/W} \) (0.136 \text{ h·ft·°F/Btu}) and \( R_{\text{conv}} = 0.016 \text{ mK/W} \) (0.027 \text{ h·ft·°F/Btu}) were estimated for the GHEs of the system (the calculations were made according to Philippe, et al. (2010)).

To calculate \( R_{\text{grout}} \) of GHEs the following three methods were selected:

- The two-dimensional multipole equation for double U-loop borehole GHEs suggested by Conti, et al. (2016);
- The equivalent diameter method proposed by Shonder and Beck (2000);
- A two-dimensional numerical model of the GHE cross-section.

In the numerical model, the GHE was modelled using the same approach as was used by Loveridge and Powrie (2014) in their “pile only model” estimating thermal resistances of piles. In particular, a two-dimensional model of the GHE cross-section was considered in the finite element software TEMP/W. The model simulated only the grout; the plastic pipes of the U-loops and the tremie pipe as well as the GHE fluid were not considered in these simulations. As in Loveridge and Powrie (2014), the constant temperatures of 20 °C (68 °F) and 10 °C (50 °F) were applied to the borehole wall and the U-loop pipe walls respectively. The heat flux between the borehole wall and the U-loop pipe walls was calculated to find the borehole thermal resistance of the grout at steady state conditions.

During the installation of the GHEs, it was observed that many spacers designed to keep the U-loops of the GHEs in place were lost which might affect the actual positions of the U-loops inside the GHEs. In addition, other factors might influence their positions such as grouting. Hence, the actual positions of the U-loops in the GHEs might not be the same as designed. It is not possible to establish the exact locations of the U-loops considering that there are many possible positions that the U-loops might take. To illustrate possible influences of the U-loop positions to \( R_b \) of the GHEs, two extreme locations of the U-loops inside the boreholes were considered in addition to the “as designed” geometry of the borehole cross-section shown in Figure 4a. These locations are:

- “Pipes apart”: the pipes of the U-loops are apart and touching the borehole sides, \( L_{\text{LU}} = 0.086 \text{ m} \) (0.282 \text{ ft}) (Figure 4b);
- “Pipes at centre”: the pipes of the U-loops are at the centre of the GHE and touching, \( L_{\text{LU}} = 0.035 \text{ m} \) (0.115 \text{ ft}) (Figure 4c).

The positions of the individual pipes in both cases are symmetrical. Clearly, there are many other non-symmetrical locations of the pipes but these are not considered herein.

The values of \( R_b \) estimated for the three U-loop locations by the three methods are plotted in Figure 5. From the figure, the values of \( R_b \) for a particular U-loop location estimated by the multipole equation and the numerical method are very similar. For example, the multipole equation estimated \( R_b \) for the “as designed” case as 0.066 mK/W (0.112 h·ft·°F/Btu), which is only marginally larger than the value of 0.062 mK/W (0.105 h·ft·°F/Btu) estimated by the numerical method for the same case. Shonder and Beck's method calculated the highest value of \( R_b \) for the “as designed” case among the values calculated by the all three methods which is equal to 0.086 mK/W (0.146 h·ft·°F/Btu).
The estimations by the numerical model and multipole equation demonstrate the importance of the position of U-loop pipes for the value of $R_b$. For the particular case considered here, the value of $R_b$ can be in between 0.045 m·K/W (0.077 h·ft·°F/Btu) to 0.085 m·K/W (0.145 h·ft·°F/Btu) depending on pipe positions as estimated by the numerical model (Figure 5). This range can be even higher if the grout of lower thermal conductivity than the grout considered here is used for the GHEs. Shonder and Beck's method does not consider the location of U-loops inside GHEs, so the values of $R_b$ estimated by this method are the same for all three locations considered.

### COMPARISON OF OPERATIONAL AND MODELLED BOREHOLE THERMAL RESISTANCES

In Figure 5, the value of $R_{be}^{*}$ is plotted along with the values of $R_b$ estimated analytically and numerically. As observed, the value of $R_{be}^{*} = 0.05$ m·K/W (0.085 h·ft·°F/Btu) is lower than $R_b = 0.062$ m·K/W (0.105 h·ft·°F/Btu) and 0.066 m·K/W (0.112 h·ft·°F/Btu) estimated by the numerical model and the multipole equation respectively for the “as designed” position of the U-loops. At the same time, the $R_{be}^{*}$ value is in between the values for the “as designed” and “pipes apart” cases calculated by both these methods. This might indicate that the actual location of the U-loops inside the GHEs is somewhere between these two positions. Such a location of the U-loops is likely. Indeed, the GHEs have a grouted tremie pipe in between the U-loops (see Figure 1b), so the U-loops cannot be closer together than the “as designed” case. At the same time, it is likely that a proportion of the U-loops may have moved...
from the centres towards edges of the boreholes during grouting.

From Figure 5, the $R'_{be}$ value does not compare well with the $R_b$ values calculated by the equivalent diameter method proposed by Shonder and Beck (2000). The $R_b = 0.086 \text{ m} \cdot \text{K/W} (0.146 \text{ h} \cdot \text{ft} \cdot ^\circ\text{F}/\text{Btu})$ calculated by this method is substantially different from the experimental $R'_{be} = 0.05 \text{ m} \cdot \text{K/W} (0.085 \text{ h} \cdot \text{ft} \cdot ^\circ\text{F}/\text{Btu})$. Shonder and Beck's method assumes one-dimensional thermal processes inside GHEs and simplifies the two U-loops of a double U-loop GHE to a single pipe with the radius of $2 \cdot r_{p,ext}$ at the centre of a borehole. This assumption seems to be close to the “pipes at centre” case of the U-loops location inside the GHEs. Considering this, it is logical that the value of $R_b$ calculated by Shonder and Beck’s method is close to the values of $R_b$ calculated by the numerical model and the multipole equation for the “pipes at centre” location. This seems to be the case.

The comparison of $R'_{be}$ and $R_b$ indicates that the $R_b$ values estimated analytically and numerically for the “as designed” geometry of the GHEs are substantially different from the $R'_{be}$ values observed experimentally. The estimations suggest that a deviation of the actual U-loop positions inside the boreholes from their “as designed” positions can be one of the reasons of such differences. The second reason can be the specific assumptions of the models which might not be accurate for particular GHEs. The differences between actual values of borehole thermal resistances and the values of these parameters estimated during design can lead to oversizing or undersizing of GHEs (Mikhaylova, et al. 2016).

Overall, designers should be aware of the significant influence of the U-loop pipe position on the values of borehole thermal resistances. During installation of U-loops into boreholes, it is hard to ensure the correct spacing between the pipes of the U-loops. In some cases, spacers are used to separate the pipes, but some contractors prefer not to use them since they reduce flexibility of U-loops and make the installation process harder. Also, the spacers are frequently lost during installations. A possible measure to verify calculated values of $R_b$ is in-situ testing (TRT) of GHEs. During a TRT, $R_b$ of an actual pilot GHE is measured. Since other GHEs of the same borefield would be installed following the same technological procedures, the TRT can provide additional information about likely effective borehole thermal resistances of the GHEs in the borefield to consider during their design. However, the values of the resistances obtained from TRTs can be affected by measuring errors. Also, it is not clear to what extend TRT results of a single GHE can represent all GHEs in a borefield. More comparisons, as the one performed here, may help to evaluate possible differences between calculated and measured resistances.

**CONCLUSION**

This study compares an experimentally determined effective borehole thermal resistance $R'_{be}$ of double U-loop GHEs with the resistances $R_b$ predicted analytically and numerically for the same GHEs. The monitoring data of a full-scale commercial GSHP system was used for this comparison. The analysis showed that the $R_b$ values estimated by models based on the design geometry of the GHEs can be significantly different from the $R'_{be}$ value determined experimentally. The $R_b$ value estimated by the equivalent diameter method differs from $R'_{be}$ the most. Possibly, the U-loop design positions inside the boreholes changed during the installation of the GHEs because of installation processes which might have caused the differences between the predicted and observed resistances values.

The analysis showed the importance of the positions of U-loops inside borehole GHEs to the values of borehole thermal resistances. If these positions are changed during an installation, but $R_b$ estimated analytically for the design geometry of GHEs, the GHE design lengths can be overestimated or underestimated. Designers should be aware of such possible errors in design lengths of GHEs. To improve estimations of $R_b$, in-situ thermal response tests can be performed on pilot GHEs to contribute to the estimation of $R_b$ values calculated by models.

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High time-resolution analytical models for heat transfer through U-shaped ground heat exchangers

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ABSTRACT
This paper reports a set of high time-resolution analytical models for heat transfer of GHEs, including a full scale line-source model for heat transfer outside U-shaped pipes and a quasi-3D heat transfer model for the circulating fluid inside U-shaped pipes. The full scale line-source model is a composite expression consisting of a composite medium line-source solution for short-term temperature responses, an infinite line-source solution for mid-term temperature responses, a finite line-source solution for long-term temperature responses, and a finite line-source solution for the thermal interaction between adjacent boreholes. The quasi-3D model for heat transfer inside U-pipes tackles the variation of the fluid temperature along the U-shaped channels. The fluid temperatures in the descending and ascending legs are derived as functions of time and borehole depth. The quasi-3D model, together with the full scale G-function, constitutes a complete theoretical model for heat transfer by ground heat exchangers, providing a computational-efficient approach to computer simulation, analysis, and optimum design of ground-coupled heat pump and ground heat storage systems.

INTRODUCTION
A great challenge in heat transfer analysis of ground heat exchangers (GHEs) is the involved diverse time and space scales (Li and Lai 2015; Li, et al. 2016). As shown in Li and Lai (2015) and Li et al. (2016), four space scales ranging from several centimeters to hundreds meters and eight time scales ranging from minutes to decades are probably involved in the heat transfer by borehole GHEs. A successful model should provide an accurate and efficient approach to calculating the diverse-scale thermal problem in the ground. From the perspective of accuracy, the heat transfer analysis should use a model of time resolution ranging from sub-hour to decades, corresponding to a space range from several centimeters to more than one hundred meters. From the efficient viewpoint, the analysis should tackle the complete spectrum of the broad time-length scales in an analytical way.

To meet the challenge in modeling GHEs, researchers have developed a vast number of analytical heat transfer models. Among them, conventional finite line-source models (FLS) appear to be the most suitable and efficient models for calculating the long-term temperature response in the ground (Claesson and Javed 2011), but they are unsuitable for modeling the short-term thermal response (Claesson and Javed 2011; Yang and Li 2014); the Infinite line-source solution (ILS) is only applicable to calculating the mid-term temperature response (Li, et al. 2016); and the infinite composite medium line-source model emerges as a valid model for calculating the short-term temperature response of GHEs (Li and Lai 2012; Li and Lai 2013; Yang and Li 2014), but it is invalid for the long-term process. An efficient analytical model that can address the entire time-space spectrum will be beneficial. Very few attempts...
have been made to develop such a model, however.

Claesson and Javed (2011) attempted to develop a heat transfer model covering time scales from minutes to decades; but they used the equilibrium-diameter assumption for their short-term model so that the used short- and long-term solutions are not guaranteed to blend. Li et al. (2014) developed a full scale line-source model by combining the composite-medium line-source solution and the conventional ILS and FLS solutions using the idea of matched asymptotic expansions. But the solution ignores the thermal interaction between adjacent boreholes and the vertical variation in fluid temperature along U-shaped tubes. The purpose of this paper is, based on our previous work, to extend the full scale line-source model to deal with these two thermal processes.

HEAT TRANSFER MODEL

Heat transfer by borehole GHEs spans a wide range of time and space scales; it is necessary to decompose the thermal process into two parts to simplify the analysis. We propose here decomposing the heat transfer of borehole GHEs by the outer wall of the U-shaped pipes: assume the heat transfer from the circulating fluid to the outer wall of the U-pipe to be a quasi-steady (or steady-flux) process and the heat transfer from the outer U-pipe wall to the ground to be a transient process. Further suppose that \( T_f \) and \( T_g \) are the fluid temperatures in the two legs of the U-pipe, which are functions of time \( t \) and coordinate \( z \) in the depth direction (\( z = 0 \) on the ground surface). The energy balance equations for \( T_f \) and \( T_g \) can be written as

\[
\begin{align*}
\frac{c_p m}{\bar{z}} \frac{\partial T_{f1}(t, z)}{\partial z} &= \frac{T_{p1}(t, z) - T_{f1}(t, z)}{R_p} \\
\frac{c_p m}{\bar{z}} \frac{\partial T_{f2}(t, z)}{\partial z} &= \frac{T_{p2}(t, z) - T_{f2}(t, z)}{R_p}
\end{align*}
\]

Eqs. (1) and (2) ignore the heat conduction in the fluids along the z-direction as a result of the small temperature gradient in this direction, but consider both the local horizontal thermal process through the U-shaped tube and the vertical variation of the fluid temperature along the U-shaped channel (Ma, et al. 2015). \( R_p \) is the fluid-to-pipe thermal resistance of the U-pipe (unit is W-m/K):

\[
R_p = \frac{1}{2\pi k_p} \left( \ln \frac{r_o}{r_i} + \frac{k_p}{\alpha r_i} \right)
\]

The boundary conditions of Eqs. (1) and (2) are the temperature conditions on the ground surface (\( z = 0 \)) and at the bottom of the borehole (\( z = H \)), respectively: \( T_f(t, 0) = T_{f, in} \), \( T_g(t, H) = T_f(t, H) \), where \( T_{f, in} \) is the inlet temperature of GHEs. Having these conditions, it is easy to solve Eqs. (1) and (2) analytically. And the final expression for the outlet temperature of GHEs, \( T_{f, out} \), is (Ma, et al. 2015)

\[
T_{f, out}(t) = T_u(t) + \frac{H}{T_{f, in}(t) - T_u(t)} \eta \exp(-2\eta)
\]

where \( \eta \) is a dimensionless quantity having the similar implication as the number of transfer units (NTU):

\[
\eta = \frac{H}{c_p m R_p}
\]

Eq. (4) is applicable for GHEs using single U-tube and parallel-connected double U-tube. The similar idea is
applicable to the case of series connected double U-tube (or W-shaped tube), and the outlet temperature response is (Ma, et al. 2015)

\[ T_{f,\text{out}}(t) = T_a(t) + \frac{q_f}{T_f,\text{in}}(t) - \frac{T_a(t)}{\eta} \exp(-4\eta) \]  

(6)

\( T_a \) occurred in Eqs. (4) and (6) is the mean temperature of the outer walls of U-shaped pipes, which is unknown and needs to be determined. There are several approaches to determining \( T_a \). The approach we used is by imposing a constant heat flux boundary condition on the outer walls because the heat flux can be easily determined by the heating and cooling loads and COP of the heat pump. By this assumption, \( T_a \) can be determined using the concept of unit-step response function (i.e., G-function) as follows (Li et al. 2016; Li and Lai 2015):

\[ T_a = T_0 + q_l G(t) \]  

(7)

\( G \) function is the mean temperature of the outer wall of the U-shaped pipe due to a unit step change in the heat flux (i.e., \( q_l = 1 \) in Eq. (7)). \( G \) function used here has the same dimension as thermal resistance \( R_p \) and their forms differ slightly from the dimensionless counterparts. It should be noted that Eqs. (4) – (7) embrace the natural way of the operation of GHEs: GHEs are driven by the inlet temperature of GHEs and the flow rate of the circulating fluid; that \( T_{in} \) and \( m \), in conjunction with \( T_0 \), are used in Eqs. (4) – (7) as the boundary/driven conditions for computing \( T_{f,\text{out}} \) reproduces the physical reality.

Figure 1  The line-heat-source assumption used in the heat transfer model for U-shaped GHEs
Following the idea, \( G \) must be known to calculate \( T_a \) and \( T_{out} \). This work uses the full scale line-source model for the \( G \) function, which is a composite expression consisting of four temperature response functions as follows (Li, et al. 2014; Li, et al. 2016):

\[
G(t) = G_i + G_o - G_m + \Delta G \tag{8}
\]

Here, \( G_m \) is an infinite line-heat-source solution to the temperature response on the borehole wall (Carslaw and Jaeger 1959):

\[
G_m(t) = \frac{1}{4 \pi k_s} \int_{I_{m, s}}^{Y} \frac{\exp(-s)}{s} ds = \frac{1}{4 \pi k_s} E_1 \left[ \frac{r_b^2}{4a_t} \right] \tag{9}
\]

\( G_o \) in Eq. (8) is a finite line-heat-source solution for the temperature of the borehole wall. According to Claesson and Javed (2011), the expression for the mean temperature of the borehole wall is:

\[
G_o(t) = \frac{1}{4 \pi k_s} \int_{I_{o, s}}^{Y} \frac{\exp(-r_b^2 u^2)}{u^2} \frac{I(H_u, H_u)}{H_u^2} du \tag{10}
\]

where \( I \) is a special function defined as follows:

\[
I(x_1, x_2) = 2 \cdot \operatorname{ierf}(x_1) + 2 \cdot \operatorname{ierf}(x_1 + x_2) - \operatorname{ierf}(2x_1 + 2x_2) - \operatorname{ierf}(2x_2) \tag{11}
\]

Here, \( \operatorname{ierf}(x) \) denotes integral of the error function \( \operatorname{erf}(x) \):

\[
\operatorname{ierf}(x) = \int_0^x \operatorname{erf}(s) ds = x \cdot \operatorname{erf}(x) - \frac{1}{\sqrt{\pi}} \left[ 1 - \exp\left(-x^2\right) \right] \tag{12}
\]

In Eq. (8), \( G_i \) is a composite medium line-source solution to the average temperature of the outer wall of the U-shaped pipes. The key idea of the composite medium line-source solution is that the legs of U-shaped tubes (not the borehole) are modeled as lines of heat sources placed in a cylindrical composite medium. Thus it is possible to obtain the transient temperature field inside the borehole and the average temperature \( T_o \). To simplify calculation, \( T_o \) is approximated by the average temperature of points A and B as labeled in Fig. 1. Readers can find more details about this model in the references (Li and Lai 2012; Li and Lai 2013). For single U-shaped tubes, \( G_i \) is

\[
G_i(t) = \frac{1}{2 \pi k_b} \int_{I_{i, a}}^{Y} \frac{\exp(-u^2 a_t^2)}{a_t} \left[ J_{2i}(ur_a) + J_{2i}(ur_B) \right] J_{2i}(ur) \left( \rho g - \psi f \right) du \tag{13}
\]

For double U-shaped pipes, the expression of \( G_i \) is
\[ G_i(t) = \frac{1}{2\pi k_{b\omega}} \hat{a} \hat{g} \exp(-u^2\alpha_{b\omega}t) \]

\[ \frac{J_{4i}(ur_A) + J_{4i}(ur_B)}{2u(\psi^2 + \psi^2)} \int_0^\infty \pi \]

where \( r_A \) and \( r_B \) are the radius coordinates of points A (e.g., \( r_A = D - r_i \)) and B (e.g., \( r_B = D + r_i \)) (see Fig. 1); functions \( \varphi \), \( \psi \), \( \phi \), and \( \psi \) are defined as

\[ \varphi = akJ_n(ur_B)J_n^\prime(aur_B) - J_n^\prime(ur_B)J_n(aur_B) \]  
(15a)

\[ \psi = akJ_n(ur_B)Y_n^\prime(aur_B) - J_n^\prime(ur_B)Y_n(aur_B) \]  
(15b)

\[ f = akY_n(ur_B)J_n^\prime(aur_B) - J_n^\prime(ur_B)J_n(aur_B) \]  
(15c)

\[ g = akY_n(ur_B)Y_n^\prime(aur_B) - Y_n^\prime(ur_B)Y_n(aur_B) \]  
(15d)

where \( a \) and \( k \) are dimensionless variables \( k = k_s/k_b \), \( a = (a_s/a_b)^{1/2} \); \( J_n \) and \( Y_n \) denote the Bessel functions of the first and the second kind of order \( n \), respectively; \( J_n^\prime \) and \( Y_n^\prime \) are the derivatives of \( J_n \) and \( Y_n \); the order \( n \) is equal to \( 2i \) and \( 4i \), respectively, in Eqs. (13) and (14).

\[ \Delta G \] in Eq. (8) denotes the superimposed temperature due to the thermal interaction between adjacent boreholes. It is calculated by the finite line-source model:

\[ \Delta G(t) = \sum_{i=1}^{M-1} \frac{1}{4\pi k_s} \int_0^\infty \exp(-\beta^2u^2) \frac{I(Hu, H_s)}{Hu^2} du \]  
(16)

In summary, the \( G \) function defined in Eq. (8) is a composite expression consisting of a finite line-source solution, an infinite line source solution, a composite medium line-source solution, and a finite line-source solution to the thermal interaction between boreholes. The underlying idea of this combined formula is the principle of matched asymptotic expansions (Li, et al. 2014; Li, et al. 2016). In fact, referring to Eq. (8), we can find that the short-term (high-frequency) temperature response is calculated only by \( G_i \) if \( G_o \) and \( G_m \) cancel each other out within short time scales (e.g., times smaller 10 hr); the mid-term response is calculated by \( G_m \) if \( G_o \) can offset \( G_i \) within this time period (from 10 hr to several months); and the long-term response is determined only by \( G_o \) if \( G_i \) cancels out \( G_m \) during large time periods (times larger several months). Therefore, it is essential for Eq. (8) that the \( G \) functions (i.e., \( G_i \), \( G_o \), and \( G_m \)) must be derived from correct and complete theoretical models. Otherwise, they cannot cancel out each other in the overlapped time ranges. Since Eq. (8) synthesizes the advantages of short-, mid-, and long-term solutions, it is suitable for calculating GHE's temperature responses from several minutes to decades.

RESULTS

This section first provides a collection of graphs for \( G \) functions generated by using the full-scale line-source solutions (i.e., Eqs. (8) – (16)). Compared to conventional \( G \)-functions, the new \( G \)-functions, which are anchored to the theoretically complete models, are more direct for use and, more importantly, are expected to be more reliable and accurate, being a new contribution to this field.

The new model integrates with the finite line-source model by the principle of superposition, making it feasible
to calculate the thermal interaction between boreholes in a theoretical way. While this improvement is not new, this paper proposes a very fast but accurate way of calculating thermal interaction between large numbers of boreholes, a way that is based on systematic parametric analysis. Our parametric analysis reveals that the average temperature response of a GHEs cluster can be well approximated by the average of the maximum and the minimum temperatures of the GHEs cluster (Fig. 2). Fig. 2 illustrates that the average of the minimum and the maximum temperatures agrees with the corresponding mean temperatures. The relative errors are only about 2%. Additionally, parametric analysis validates that this approximation is also applicable to other GHEs configurations, such as the linear and the L-shaped arrangements of GHEs.

Figure 2  Temperature responses of $5 \times 5$ and $10 \times 10$ GHEs matrices: the maximum, the minimum, the average, and the approximated average temperature responses.

Figure 3  Temperature responses of GHEs matrices with different borehole numbers and spacing (from single borehole to 10 000 boreholes).
Fig. 3 shows general variations of the mean temperature responses of square GHEs clusters (i.e., $G$ functions). Similar charts have been reported by some researchers (Eskilson 1987; Claesson and Javed 2011). Fig. 3A is complementary to previous charts by providing $G$ functions for extremely large GHE clusters (up to 10000 boreholes). It is important to remember that these curves are inadequate and incomplete for practical design because $G$ functions involve too many parameters to summarize in several 2D graphs. In practice, computerized approaches to $G$ functions would be more desirable and accurate.

Another improvement of the new model is the integration of the composite medium line-source model, which can accurately predict short-term temperature responses and avoid the complex computation of effective borehole thermal resistance $R_b$. As far as we know, the composite medium line-source solution is probably the best analytical solution (at least so far) to the short-term heat transfer of GHEs. This solution can yield transient temperature field inside boreholes, making the quasi-steady-state assumption unnecessary. Accordingly, $R_b$ used in the conventional models is not needed anymore.

The authors would like to draw more attention to the elimination of $R_b$ from the simulation. The authors tend to argue that $R_b$ is redundant for the thermal analysis of GHEs because not only it is more difficult to determine than $R_p$ but also it is a derivative of the assumption about the heat transfer inside boreholes. The computation of $R_b$ is difficult and uncertain, depending on the installation of U-tubes, the thermal properties of the ground and the backfilling material. Several theoretical formulas for $R_b$ exist, for example, the steady-state line-source and the first-order multipole expressions for single U-tube GHEs (Hellström 1991). $R_b$ can be estimated from a thermal response test (Spitler and Gehlin 2015). No matter what method is used, however, the computation of $R_b$ is more complicated than that of $R_p$. In the new model, all the complications are lumped into the $G$-functions. A complete set of $G$-charts should be useful for engineering applications. Furthermore, the quasi-steady-state assumption is a seeming simplification to heat transfer analysis of GHEs, which introduces significant errors in the calculations of short-term or high-frequency temperature responses (Yang and Li 2014). This is the key reason why the study of short-term heat transfer has received a spate of interest recently (Lamarche and Beuchamp 2007; Javed and Claesson 2011; Li and Lai 2012; Lei, et al. 2015; Ruiz-Calvo, et al. 2015; Wei, et al. 2016). The composite medium line-source solution used here has been validated to be a viable solution to the problem of short-term thermal process (Li and Lai 2013; Yang and Li 2014). Having this solution, it seems to us that the quasi-steady-state assumption is not needed anymore, neither is $R_b$.

CONCLUSION

This paper reports a set of theoretical solutions to heat transfer by borehole GHEs, which can predict temperature responses of GHEs from sub-hour to decades. The solutions use the inlet temperature of GHEs, the flow rate of the circulating fluid, as well as the initial ground temperature as the boundary/driven conditions for computing the outlet temperature of GHEs, reproducing the physical reality of the operation of GHEs and providing a very direct way of GHEs simulation. Another outstanding feature of the new model is the ability to predict the transient heat conduction inside the boreholes. This ability enables the widely used effective borehole thermal resistance to be unnecessary, a derivative of the quasi-steady state assumption about the heat transfer inside boreholes.

NOMENCLATURE

\begin{align*}
    a_b &= \text{Thermal diffusivity of the backfilling material (m}^2/\text{s}) \\
    a_s &= \text{Thermal diffusivity of soil (m}^2/\text{s}) \\
    B_i &= \text{Distance between the } i\text{th borehole and the borehole under consideration (m)} \\
    c_p &= \text{Specific heat of the circulating fluid (kJ/kg°C)} \\
    D &= \text{Half spacing between the legs of U-tube (cm)} \\
    H &= \text{Borehole depth (m)} \\
    H_1 &= \text{Distance between the ground surface and the top end of GHEs (m)}
\end{align*}
\[ k_b = \text{Thermal conductivity of the backfilling material (W/m°C)} \]
\[ k_p = \text{Thermal conductivity of the U-shaped pipe (W/m°C)} \]
\[ k_s = \text{Thermal conductivity of soil (W/m°C)} \]
\[ m = \text{Flow rate of the circulating fluid (kg/s)} \]
\[ M = \text{Total number of boreholes} \]
\[ q_l = \text{Heat transfer rate per unit length of the borehole (W/m)} \]
\[ r = \text{Radial distance from the line source (m)} \]
\[ r_b = \text{Borehole radius (cm)} \]
\[ r_o = \text{Outer radii of U-pipe legs (cm)} \]
\[ r_i = \text{Inner radii of U-pipe legs (cm)} \]
\[ s = \text{Dimensionless integral variable} \]
\[ T_0 = \text{Initial temperature of the ground (°C)} \]
\[ T_{p1} = \text{Mean temperature of the outer wall of the U-pipe leg 1 (°C)} \]
\[ T_{p2} = \text{Mean temperature of the outer wall of the U-pipe leg 2 (°C)} \]
\[ u = \text{Integral variable (1/m)} \]
\[ \alpha = \text{Convective heat transfer coefficient inside U-pipe (W/m²°C)} \]

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Development of TPRT (Thermal Performance–Response Test) for Borehole Heat Exchanger Design

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ABSTRACT

To obtain the effective thermal conductivity and borehole thermal resistance required to design borehole heat exchangers (BHEs), thermal response tests (TRTs) are usually conducted. Although many advanced TRT methods have been proposed, most TRTs cannot directly provide the actual thermal performance of an installed BHE. Because many uncertainties exist in constructing even conventional BHEs, examining the transient heat exchange rate allows inspection of the construction quality; the rate can also be used as a reference value for design. To determine the actual heat exchange rates of BHEs, it is necessary to conduct thermal performance tests (TPTs) under a constant inlet fluid temperature. However, TPT requires expensive equipment, including a water tank and a complex control system; thus, generally only TRT is conducted. To overcome the existing problems of TPT, in this study, we proposed a thermal performance–response test (TPRT) that combines TRT and TPT. This method involved the construction of a cost-effective TPRT apparatus by adding only a general PID controller and a solid-state relay to an existing TRT apparatus. Using the apparatus constructed by the proposed method, two TPRTs were conducted to confirm the performance of the apparatus and the validity of the TPRT method. Additionally, by defining the new parameter of the unit heat exchange rate, one potential simple and reliable design method for BHEs was explored.

INTRODUCTION

Thermal response tests (TRTs) are conducted to obtain the effective thermal conductivity of ground and the borehole thermal resistance, which are required to design borehole heat exchangers (BHEs). After Mogensen (Mogensen 1983) first proposed the idea of a TRT, many advanced TRT methods were developed (Acuña, et al. 2011; Fujii, et al. 2009; Raymond, et al. 2015). Although most TRTs can provide information regarding ground thermal properties, they cannot directly determine the actual thermal performance (i.e., the transient heat exchange rate) of constructed BHEs. After estimating the effective thermal conductivity of ground via TRT, the performance of a BHE can be estimated indirectly using numerical or analytical methods. However, this indirect estimation requires further data, such as the thermal properties of the backfill material and accurate geometric details of the BHE, which are not always available and can be difficult to obtain. If a ground-source heat pump (GSHP) is designed for intermittent operation, such unavailable information gains greater importance because it significantly affects the short-term performance and response of the BHE.

The temperature of the circulating fluid depends on the building load conditions and affects the heat exchange rate. Knowing the resulting heat exchange rate between the circulating fluid and ground is helpful for BHE design. For
ground heat exchangers (GHEs) with complex geometries, such as energy piles with multiple U- or W-shaped pipes and helical-coil GHEs, the heat exchange rate is especially helpful because the performance of such complex-geometry GHEs differs from that of conventional BHEs and the design method is not well established. To obtain the actual thermal performance of a complex-geometry GHE, a thermal performance test (TPT), which maintains a constant inlet fluid temperature, is usually conducted. To investigate the thermal performance of various types of GHEs, several TPTs have been reported (Gao, et al. 2008a, 2008b; Li, et al. 2006; Wang, et al. 2009). In addition to examining the performance of complex shape GHEs, TPTs can also be used to examine the actual performance of an installed conventional BHE. Obtaining the heat exchange rate is important because many uncertainties affect the construction and design of BHEs, such as the geometry of the U-tubes (torsion and bending of pipe legs), uneven packing of the grouting or backfill, unverified subsurface heat-transfer phenomena, and uncertain ground properties, as noted in Bernier (2002); Raymond, et al. (2011); Spitler and Gehlin (2015); Witte (2013).

As described above, although the information from TPT is as important as that from TRT, conducting both TRT and TPT is atypical because they require different apparatus. Typically, only TRTs are conducted. To overcome this situation, we propose a method that combines the TRT and TPT, termed the thermal performance–response test (TPRT). The proposed method is accompanied by a new apparatus and estimation method. Compared to the total cost of constructing conventional TRT apparatus, which generate constant heat rates from heaters, the proposed apparatus can be constructed with an additional cost of only $300–400, and the estimation can be conducted using the well-known infinite line source (ILS) model (Carslaw and Jaeger 1959; Ingersoll, et al. 1954) and a numerical optimization method. Therefore, the proposed apparatus and estimation method are practically appealing.

In this work, we constructed a TPRT apparatus using an existing conventional TRT apparatus. Two TPRTs with different inlet temperatures (25 °C and 30 °C) were conducted, and the estimation results were validated using previous TRT results. Additionally, a new parameter, termed the unit heat exchange rate $Q_u$, was proposed to predict GHE performance under certain operation conditions.

**EXPERIMENTAL SETUP AND TPRT APPARATUS**

TPRTs were applied to a BHE installed at the Chiba Experimental Station at the University of Tokyo (Inage Ward, Chiba, Japan). The site was stratigraphically divided into a top layer of loam and clay up to a depth of 8 m, followed by fine sand, silt, and fine sand again between the depths of 8–25 m, 25–31 m, and 31–60 m, respectively. The depth-averaged undisturbed ground temperature varied seasonally at around 17 °C. The BHE had an effective depth of 50 m, and the borehole had a diameter of 165 mm. A single high-density polyethylene U-tube with outer and inner diameters of 34 and 27 mm, respectively, was inserted into the borehole. The thermal conductivity of the U-tube was 0.38 W/(m·K). To maintain a shank spacing of 50 mm, spacers were attached between the U-tube legs at 10 m intervals. The borehole was grouted with Portland cement mixed with 20% silica sand. Water was used as the heat carrier fluid. More details can be found in (Choi and Ooka 2016a, 2015).

Reported TPT apparatus use water tanks as thermal buffers; researchers note that complex control systems are required, and the resulting high cost makes TPT apparatus impractical (Gao, et al. 2008a; Li, et al. 2006; Wang, et al. 2010). The apparatus proposed in this study does not require a water tank; only a solid-state relay (SSR) and a general proportional–integral–derivative (PID) controller are required to control the inlet fluid temperature. A schematic of the proposed system is shown in Figure 1. The process variable of this system is the inlet fluid temperature (outlet temperature of the TPRT apparatus), which is measured by a PT-100 sensor. Depending on the fluid temperature passing through the heaters, the PID controller calculates the desired power output and then sends signals to the SSR to regulate the power output of the heaters. Because the proposed method does not use a chiller, the apparatus cannot be used for heat extraction tests. However, compared to a conventional TRT apparatus, the additional cost is only approximately $300–400. Moreover, the proposed control system can be easily applied to any existing TRT apparatus.

In this study, to obtain good controllability, three coefficients—P, I, and D—were selected based on trial and error. The P, I, and D coefficients were set to 6, 64, and 15, respectively. The heaters installed in the existing TRT
apparatus had power outputs of 1, 2, and 4 kW. The total output was 7 kW.

When the set point temperature is 25 °C and the initial ground temperature is 17.7 °C, the rise time required to reach the set point temperature from the beginning of the test is approximately 6 min, and the amount of overshoot is only 0.45 °C (Figure 2). For higher-power heaters, a faster rise time can be expected. After an elapsed test time of 15 min, the temperature difference between the set point and inlet temperature is less than 0.2 °C. After an elapsed test time of 1 h, the temperature difference is less than 0.1 °C; this error can be regarded as a steady-state error of the control system. The developed TPRT apparatus shows that sufficiently fast, accurate, and robust control can be achieved without a water tank and complex control logic.

Figure 1  Schematic of TPRT system.

Figure 2  Early time temperature response and heat exchange rate (set point temperature is 25 °C).

PARAMETER ESTIMATION METHOD

The parameter estimation method in this study used the infinite line source (ILS) model (Carslaw and Jaeger 1959; Ingersoll, et al. 1954) as a physical model that reproduced the thermal response of the BHE and the quasi-Newton method as a numerical optimization algorithm. To consider the continuously varying heat exchange rate of TPRT, a temporal superposition of heat pulses was applied to the ILS model. The temporal superposition–applied ILS model with respect to the average fluid temperature $\bar{T}_{cf,\text{cal}}$ can be expressed as follows:
The volumetric heat capacity of soil \( C_v \) was assumed to be 2.8 MJ/m\(^3\)K. The depth-averaged undisturbed ground temperature \( T_0 \) was measured by circulating water in the BHE without heat generation. It varied according to the outdoor environment in the range of 16.5–17.0 °C. However, because the water in the aboveground hydraulic circuit is affected by the outdoor environment, the measured \( T_0 \) is higher than 17.0 °C and lower than 16.5 °C in summer and winter seasons, respectively. The transient heat exchange rate per unit length of the BHE \( q \) can be obtained from the flow rate \( \dot{V}_{cf} \), inlet temperature \( T_{cf,in} \), and outlet temperature \( T_{cf,out} \).

\[
q = Q_{BHE}/H = \rho_{cf} c_{cf} \dot{V}_{cf}(T_{cf,in} - T_{cf,out})/H
\]  

The heat exchange rate per unit length of BHE at a certain time step, \( q_n \), is a time-varying value. The heat rate \( q \) is regarded as squared pulses obtained by averaging the \( q \) values measured at 5-s intervals over a 2-h period.

The calculated average fluid temperature \( \bar{T}_{cf,cal} \) is a function of two unknown variables: the effective thermal conductivity \( \lambda_{eff} \) and the borehole thermal resistance \( R_b \). By minimizing the difference between the calculated fluid temperature \( \bar{T}_{cf,cal} \) and measured fluid temperature \( \bar{T}_{cf,exp} \) to less than \( 10^{-5} \) for every 2-h interval of time step, both \( \lambda_{eff} \) and \( R_b \) can be estimated. The objective function and stopping criterion are described in Eq. (3).

\[
f_{obj}(\lambda_{eff}, R_b) = (\bar{T}_{cf,exp} - \bar{T}_{cf,cal}(\lambda_{eff}, R_b))^2 \\
f_{obj}(\lambda_{eff}, R_b) \leq 10^{-5}
\]  

This objective function was minimized using the quasi-Newton method, and the Broyden–Fletcher–Goldfarb–Shanno (BFGS) method was used to approximate the Hessian. To enhance the speed of the estimation and alleviate the ill-posed nature of the parameter estimation, the search range was restricted based on the initial guess values of each time step, \( \lambda_{eff,int} \pm 0.5 \) W/(m·K) and \( R_{b,int} \pm 0.05 \) m·K/W. From the second time step onward, the estimated values of the previous time step were used as the initial guess values for the current time step. The details and advantages of the method can be found in Choi and Ooka (2015).

**APPLICATION OF IN SITU TPRT**

Two TPRTs were conducted for 96 h. The experimental conditions, average flow rate, and average heat exchange rate are summarized in Table 1. The set point inlet temperatures were 25 °C (TPRT_25) and 30 °C (TPRT_30), respectively. The temperature response and heat exchange rate are shown in Figure 3. The inlet temperatures of both TPRT_25 and TPRT_30 are very stable. Unlike in TRT, the diurnal disturbance of the outdoor environment, which affects the temperature response, was well controlled. Immediately after noon, when the disturbed heat flux was the highest because of solar irradiation (Choi and Ooka 2016b, 2016c), the inlet temperature was slightly perturbed. However, the largest error of the inlet temperature was less than 0.18 °C from 3 h to the end of test time, which can be considered steady-state control. Considering that logging was conducted at 5-s intervals, the results imply very good controllability of the developed apparatus, and thus, the TPRT can be conducted without a water tank or complex control system.
Both tests show approximately 7 kW for the heat exchange rate immediately after the experiments begin, but this rate decreases rapidly. At the final elapsed time of 96 h, the heat exchange rates of TPRT_25 and TPRT_30 are 1.03 kW and 1.69 kW, respectively. After about 12 h, the heat rate decreases linearly with the logarithm of time.

Estimations were conducted using the average fluid temperature (Figure 3) and Eq. (1)–(3). Because the early-time response dominated by the BHE itself cannot be accurately reproduced by the ILS model, which is derived based on the assumption of a homogeneous medium, estimations were conducted from 12 h onward. The estimation results are shown in Figure 4, and the final estimated values are listed in Table 2. The estimation behavior is stable, and the estimated values are very close to the values obtained from TRTs conducted in 2015 at the same site (Choi and Ooka 2016a, 2016c) as listed in Table 2.

Although the estimations in this study were successful, for TPRT on other GHEs, parameters should be estimated carefully. Parameter estimation requires a physical model that can appropriately reproduce the temperature response of the GHE. Using an inadequate model would produce unreliable estimated values.

### Table 1. Experimental Conditions of TPRT

<table>
<thead>
<tr>
<th>Test name</th>
<th>Set point temperature [°C]</th>
<th>Duration [h]</th>
<th>Flow rate [L/min]</th>
<th>Average heat rejection rate [W]</th>
</tr>
</thead>
<tbody>
<tr>
<td>TPRT_25</td>
<td>25</td>
<td>96</td>
<td>17.77</td>
<td>1193</td>
</tr>
<tr>
<td>TPRT_30</td>
<td>30</td>
<td>96</td>
<td>18.07</td>
<td>1965</td>
</tr>
</tbody>
</table>

### Table 2. Estimated Thermal Conductivity and Borehole Thermal Resistance

<table>
<thead>
<tr>
<th>Test name</th>
<th>Initial guess $\lambda_{eff}$ and $R_b$ for estimation [W/(m·K)], [m·K/W]</th>
<th>Final $\lambda_{eff}$ [W/(m·K)]</th>
<th>Final $R_b$ [m·K/W]</th>
<th>$\lambda_{eff}$ of past TRTs [W/(m·K)]</th>
<th>$R_b$ of past TRTs [m·K/W]</th>
</tr>
</thead>
<tbody>
<tr>
<td>TPRT_25</td>
<td>1.9, 0.15</td>
<td>1.91</td>
<td>0.150</td>
<td>1.86–1.98</td>
<td>0.146–0.159</td>
</tr>
<tr>
<td>TPRT_30</td>
<td>1.9, 0.15</td>
<td>1.95</td>
<td>0.149</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>
Because the heat exchange rate obtained from TPRT reflects all uncertain and unknown factors, such as the U-tube position, construction quality, and other unknown and neglected physical phenomena (e.g., natural convection and advection), it is valuable and reliable in indicating the performance of the installed GHE. Additionally, this value can be used for the simple and reliable prediction of GHE performance under certain operation conditions. To utilize the obtained heat exchange rate from TPRT, we define a new parameter that describes the heat exchange rate of GHE per unit temperature difference. We divide the heat exchange rate $Q_{GHE}$ obtained from TPRT by the absolute temperature difference between the average fluid temperature $T_{cf}$ and the undisturbed ground temperature $T_0$. We name this value the unit heat exchange rate $Q_u$ [W/K], which is expressed as follows: 

**Figure 4** Estimation results of the two TPRTs ((a) TPRT_25, and (b) TPRT_30).

**Figure 5** Semi-log plot of unit heat exchange rate and linear regression using data from 12–96 h ((a) TPRT_25, and (b) TPRT_30).
Assuming that the undisturbed ground temperature is 17 °C, we calculated $Q_u$ for two TPRTs using Eq. (4); the semi-logarithmic plots of $Q_u$ are shown in Figure 5. Regardless of the inlet set temperatures, the two graphs show very similar behaviors. This means that the obtained $Q_u$ can be extended to predict other operating conditions with different fluid temperatures. However, this extension may not hold if there is strong subsurface advection by groundwater flow.

The red dashed line in Figure 5 represents the linear regression of $Q_u$ using the data collected from 12–96 h. From the regression line, we find that $Q_u$ decreases linearly and it can be extended for a longer operation. Based on the findings of $Q_u$, we can use the TPRT for a simple performance prediction of the GHE under a certain operating condition. For example, if we know the length and magnitude of heat load $t_i$ and $Q_i$, respectively, then we can calculate the fluid temperature using an analytical or numerical response model with known thermal properties estimated from TPRT. By calculating $Q_u(t_i \cdot [T_{cf}(t_i) - T_0])$, the performance of the GHE under specific operating conditions can be easily predicted.

To extend this performance prediction method to the design of GHEs, further studies should be conducted. In the ASHRAE design method (ASHRAE 2007; Philippe, et al. 2010) for BHEs, three different heat pulses for yearly, monthly, and hourly time scales are required. If the yearly and monthly average heat pulses can be appropriately converted to the hourly pulse and, correspondingly, the length of the hourly pulse is modified, a new and simple design becomes possible.

**CONCLUSION**

In this study, using a general PID controller and an SSR, a cost-competitive TPRT apparatus was developed. The developed apparatus showed excellent controllability. Using the developed apparatus, two TPRTs were conducted on a BHE to obtain the transient heat exchange rates and estimate the effective thermal conductivity and borehole thermal resistance. The estimation results were very close to the estimated values from past TRTs at the same site.

Additionally, by defining the unit heat exchange rate $Q_u$, the possibility of a simple and reliable design method for GHEs was identified, and a draft of this new method was proposed. Some issues pertaining to the unit heat exchange rate were outlined, which require further examination and discussion.

**ACKNOWLEDGMENTS**

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**NOMENCLATURE**

\[
Q_u(t) = q_u(t) \cdot H = \frac{Q_{GHE}(t)}{[T_{cf}(t) - T_0]}
\]

$H$ = Total length of BHE (m)
$q$ = Heat exchange rate per unit length of GHE (W/m)
$Q_{GHE}$ = Total heat exchange rate of GHE (W)
$Q_i$ = Magnitude of heat load used to calculate fluid temperature (W)
$q_u$ = Unit heat exchange rate per unit length of GHE (W/(m·K))
$Q_u$ = Unit heat exchange rate of GHE (W/K)
$r_b$ = Radius of BHE (m)
$R_b$ = Borehole thermal resistance (m·K/W)
\[ t_l = \text{Length of heat load used to calculate fluid temperature and unit heat exchange rate (h)} \]

\[ T_{cf} = \text{Average temperature of circulating fluid (°C)} \]

\[ V_{cf} = \text{Volumetric flow rate of circulating fluid (m}^3/\text{s)} \]

\[ \lambda_{eff} = \text{Effective thermal conductivity of ground (W/(m} \cdot \text{K)}) \]

**Subscripts**

\[ cf = \text{circulating fluid} \]

\[ n = \text{n-th time step} \]

**REFERENCES**


Fast and accurate calculation of the soil temperature distribution around ground heat exchanger based on a Response Factor model

Tian You Xianting Li* Wenxing Shi Baolong Wang

ABSTRACT
Ground heat exchanger (GHE) is an important component of ground coupled heat pump system (GCHP). To calculate the soil temperature around GHE accurately and fast, a refined response factor model (RF model) is proposed. It combines the heat transfer inside and outside the U pipe through the temperature of pipe wall and the heat flux of U pipe. For the RF model, after calculating the response factors by CFD simulation, the soil temperature can be calculated by the deduced analytical equations. The sandbox experiment is built up to validate the the RF model. Based on the experiment, this case is also studied by the numerical simulation and the RF model. Results show that the soil temperature differences between the RF model and the experiment are only -0.21°C ~0.69°C at the 96th time step. The relative errors of the soil temperatures between RF model and numerical simulation at the 1800th time step are only 1.86%~3.94%. RF model consumes 30% time of the numerical simulation for the soil temperature calculation with 1800 time steps and consumes only 1% time of the numerical simulation for that with 350400 time steps. Therefore, the RF model is accurate and fast to calculate the soil temperature around the GHE with fluid inside.

1 INTRODUCTION

1.1 Background
Ground-coupled heat pump (Sanner et al. 2003, Pawel 2004, Lund et al. 2011, Mustafa and Hikmet 2004) is more and more popular in the world because it’s a clean and efficient technology for heating and cooling. The ground heat exchanger (GHE) is an important component in this system whose heat exchanging performance has greatly influenced the system design and operation (Florides and Kalogirou 2007, Yang et al. 2010).

The method of calculating the soil temperature around GHE and the heat exchanged by it is important to the design and the performance improvement of GHE. There are three main kinds of GHE models now: the analytical solution, the numerical solution, and the g-function model. The analytical solution (Zeng et al. 2003, Diao et al. 2004, Li and Lai 2013) needs many assumptions to simplify the problem and sacrifices its accuracy. The numerical solution (Lee and Lam 2008, Cui et al. 2008, Congedo et al. 2012) consumes long time because of its complexity. The g-function model (Yavuzturk and Spiter 1999, Yavuzturk et al. 1999, Yavuzturk 1999, Li et al. 2014) combines the analytical and numerical solution while ignores some important details, such as: the shape of U pipe and the non-uniform soil property. Calculating the soil temperature distribution accurately as well as at a fast speed needs to be studied further.

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1.2 Response factors

In our previous work, RF model (You et al. 2016) based on the response factors was proposed to calculate the soil temperature accurately and fast. Response factors represent the contribution of heat sources to the temperature variation of soil points, of which the definition is shown as Equation 1. It is equal to the excess soil temperature variation divided by average soil temperature variation caused by the heat flux at the initial time step. Therefore, the response factor is dimensionless and tends to be 1 when time becomes long. Physically, it means the heat pulse at the initial time step spreads to every corners of the soil heat retainer homogeneously at last and every soil points have the same temperature variation. The response factor is determined by the distance between the heat source and the soil points, having no relationship with the heat flux.

\[
\theta_p(j\Delta\tau) = \frac{\theta_p(j\Delta\tau)}{Q_n(\theta)\cdot\Delta\tau / \left(\sum_{k=1}^{K} \rho_k c_k V_k\right)}
\]

(1)

where, \(Y_{n,p}(j\Delta\tau)\) is the response factor of soil point p to the heat pulse of the nth heat source at the jth time step; \(\theta_p(j\Delta\tau)\) is the excess temperature of point p at the jth time step, [°C]; \(T_p(j\Delta\tau)\) is the temperature of point p at the jth time step, [°C]; \(T_{initial}\) is the initial soil temperature, [°C]; \(Q_n(\theta)\) is the heat flux released by the nth heat source at the initial time step, [W]; \(\rho_k\) is the density of the kth material in the soil heat retainer, [kg/m\(^3\)]; \(c_k\) is the specific heat capacity of the kth material in the soil heat retainer, [J/(kg·°C)]; \(V_k\) is the volume of the kth material in the soil heat retainer, [m\(^3\)]. Different materials in the soil heat retainer can be the grout, concrete, different soil layers.

Based on the definition of response factors, the soil temperature can be calculated by the accumulated contribution of heat fluxes at different time steps. The calculating method is shown in Equation 2, which is the accumulation of the heat fluxes timing the corresponding response factors during the period and divided by the soil heat capacity.

\[
\theta_p(j\Delta\tau) = \sum_{n=1}^{N} \sum_{i=0}^{j} Q_n(\theta)\cdot\Delta\tau \cdot Y_{n,p}[(j-i)\Delta\tau] / \left(\sum_{k=1}^{K} \rho_k c_k V_k\right)
\]

(2)

where, \(\theta_p(j\Delta\tau)\) is the excess temperature of soil point p at the jth time step, [°C]; \(n\) is the number of the different heat sources.

The RF model is the combination of the numerical simulation and the analytical solution. As shown in Equation 1, the response factors are based on the soil temperature under a heat pulse. These soil temperature can be calculated by the numerical simulation, like FLUENT solver, during the limited period. All the specific parameters, like the different soil thermal conductivities, different geometries of pipes, different borehole grouting materials, borehole group field can be accounted for at this stage. Different parameters contribute to different response factors. After the calculation of response factors, the soil temperature under any flux at any time can be calculated based on the analytical equations, like Equation 2.

1.3 The purpose of this paper

In the previous work (You et al. 2016), the definition of response factors is suitable to the heat sources with the known heat flux. However, for the GHE with the fluid inside the U pipe, the known variables are usually the inlet water temperature and the mass flow rate. Therefore, the RF model is refined to combine the heat transfer inside and outside the pipe in this paper. The principle of RF model is illustrated in detail. A case is studied to show the calculating procedure and the sandbox experiment is built up to validate the model. At last, the calculating speed and accuracy of RF model are compared with that of the numerical simulation.
2 PRINCIPLE OF RF MODEL

With the known inlet temperature and flow rate of U pipe, the heat flux of U pipe should be calculated in advance for the soil temperature calculation in equation 2. Thus, the heat transfer of U pipe is regarded as two parts: heat transfer outside the pipe and heat transfer inside the pipe. The temperature of pipewall and the heat flux of the pipe are used to connect them together. The heat transfer outside can be expressed by the response factors of the pipewall. The heat transfer inside can be expressed by the heat balance law. This is the principle of RF model and is illustrated in detail in this section.

2.1 Heat transfer outside the pipe

Based on the definition of response factors, the excess temperature of pipewall can be calculated by Equation 3. For the excess temperature of pipewall at jth time step, it is determined by the accumulated contribution of heat fluxes during 0~jth time steps. Taking the contribution of heat flux at ith time step, its temperature contribution is the heat flux at ith time step timing the response factors at the (j-i)th time step, because it takes (j-i) time steps to get the influence of the heat flux, and then divided by the soil heat capacity.

\[
\theta_{\text{wall}} (j\Delta \tau) = \sum_{i=0}^{j-1} Q_s(i\Delta \tau) \cdot \Delta \tau \cdot Y_{s,\text{wall}} [(j-i)\Delta \tau] \left( \sum_{k=1}^{K} \rho_k c_k V_k \right) + Q_s(j\Delta \tau) \cdot \Delta \tau \cdot Y_{s,\text{wall}} (0) \left( \sum_{k=1}^{K} \rho_k c_k V_k \right)
\]

(3)

where, \( \theta_{\text{wall}} (j\Delta \tau) \) is the excess temperature of pipewall at the jth time step, \([\^\circ C]\); \( Q_s(i\Delta \tau) \) is the heat flux released by the U pipe at the ith time steps, \([W]\); \( Y_{s,\text{wall}} [(j-i)\Delta \tau] \) is the response factor of pipewall to the U pipe at the (j-i)th time step.

2.2 Heat transfer inside the pipe

The heat flux of the pipe exchanged to the soil can be calculated by the heat capacity of the fluid timing the temperature difference of the inlet and outlet fluid, which is shown in Equation 4.

\[
Q_s(j\Delta \tau) = c_f m_f \left[ \theta_{\text{in}} (j\Delta \tau) - \theta_{\text{out}} (j\Delta \tau) \right]
\]

(4)

where, \( c_f \) is the specific heat capacity of the fluid, \([/\text{kg}^\circ \text{C}]\); \( m_f \) is the mass flow rate of the fluid, \([\text{kg/s}]\); \( \theta_{\text{in}} (j\Delta \tau) \) is the inlet fluid temperature of U pipe at the jth time step, \([\^\circ C]\); \( \theta_{\text{out}} (j\Delta \tau) \) is the outlet fluid temperature of U pipe at the jth time step, \([\^\circ C]\).

The heat flux also can be calculated by the heat convection between the fluid and the pipewall, as shown in Equation 5. The convective heat transfer coefficient can be calculated by Equation 6.

\[
Q_s(j\Delta \tau) = h_f F_{\text{pipe}} \left[ \theta_f (j\Delta \tau) - \theta_{\text{wall}} (j\Delta \tau) \right]
\]

(5)

\[
\theta_f (j\Delta \tau) = \frac{1}{2} \left[ \theta_{\text{in}} (j\Delta \tau) + \theta_{\text{out}} (j\Delta \tau) \right]
\]

\[
Nu = 0.023 \cdot Re_f^{0.8} \cdot Pr_f^{0.3} \quad \text{Re} > 10000
\]

\[
Nu = 0.116 \cdot (Re_f^{7/3} - 125) \cdot Pr_f^{1/3} \cdot (1 + \left( \frac{d_{\text{pipe}}}{l_{\text{pipe}}} \right)^{2/3}) \quad 2200 < \text{Re} < 10000
\]

\[
Nu = 1.86 \cdot (Re_f \cdot Pr_f \cdot \frac{d_{\text{pipe}}}{l_{\text{pipe}}})^{1/3} \quad \text{Re} < 2200, Pr > 0.6
\]

where \( h_f \) is the convective heat transfer coefficient of the fluid,\([W/(m^2\cdot^\circ \text{C}]\); \( F_{\text{pipe}} \) is the area of U pipe, \([m^2]\); \( \theta_f (j\Delta \tau) \) is the average fluid temperature inside U pipe at the jth time step, \([^\circ C]\); \( d_{\text{pipe}} \) is the inner diameter of the U pipe,\([m]\); \( l_{\text{pipe}} \) is the length of the U pipe,\([m]\).
2.3 Heat connection of inside and outside of the pipe

Combining Equation 3~5, once the inlet temperature is known, the heat flux of U pipe, the outlet temperature and the pipewall temperature can be calculated by the matrix, as shown in Equation 7. As a consequence, as long as the response factors are calculated by numerical simulation, the heat transfer process can be demonstrated by Equation 7 and then the soil temperature distribution can be calculated by Equation 2. Since the soil thermal conductivity and the geometry of pipes are accounted for already in the numerical simulation, they have influence on the response factors. In the Equation 7, there is no need to consider them again.

\[
\begin{pmatrix}
Q_i(j\Delta \tau) \\
\theta_{\text{out}}(j\Delta \tau) \\
\theta_{\text{wall}}(j\Delta \tau)
\end{pmatrix} =
\begin{pmatrix}
1 & c_f m_f & 0 \\
1 & -\frac{1}{2} h_f F_{\text{tube}} & h_f F_{\text{tube}} \\
-\Delta \tau \cdot Y_{\text{wall}}(0) \left( \sum_{k=1}^{K} \rho_{k} c_{k} V_{k} \right) & 0 & 1
\end{pmatrix}
\begin{pmatrix}
c_f m_f \cdot \theta_i(j\Delta \tau) \\
\frac{1}{2} h_f F_{\text{tube}} \cdot \theta_{\text{out}}(j\Delta \tau) \\
\sum_{i=1}^{i-1} Q_i(i\Delta \tau) \cdot \Delta \tau \cdot Y_{\text{wall}}[(j-i)\Delta \tau] \left( \sum_{k=1}^{K} \rho_{k} c_{k} V_{k} \right)
\end{pmatrix}
\]

(7)

Since the response factors has no relationship with the heat flux, when calculating it, the pipe can be considered as a solid without fluid inside and it releases the heat pulse at the initial time step. In this way, the calculating speed of numerical simulation for response factors can be greatly increased. Besides, the response factors tend to be 1 when time becomes longer. As a consequence, for a long term soil temperature calculation, only the response factors at the initial time steps needs to be simulated and those at the following time steps can be assumed as 1. The simulated time steps of real response factors are determined by the demand of accuracy.

3 CASE STUDIES

3.1 Sandbox experiment

To validate the RF model, the sandbox experiment is built up, which is composed of a sandbox placed with thermocouples, data acquisition system, a device providing constant temperature water, water pump and a flow meter, as shown in Figure 1.

Figure 1  Sandbox experiment

Figure 2  Side view of thermocouples placement within sandbox

(a) Geometry

(b) Meshing

Figure 3  The studied case in ANSYS
The sandbox is 1m × 1m × 1m with a U-pipe placed at the center of the sandbox and is filled with sand. The diameter and length of U pipe are 8mm and 950mm. The thermal properties of the sand are tested by the transient plane source method. The specific heat capacity, density and heat conductivity of the sand are respectively 757[J/(kg·K)], 1255[kg/m³], and 0.22655[W/(m·K)]. 18 thermocouples are placed to test the soil temperatures, as shown in Figure 2. The sandbox is wrapped by the thermal insulation material with the thickness about 100mm. So, the sandbox wall is regarded as adiabatic. The initial excess soil temperature is homogeneous and considered as 0. In the experiment, the water flows inside the U pipe at the speed of 1.725m/s and it heats the sand for 24 hours. The excess temperature of the U pipe inlet keeps constant at about 14°C. The data acquiring system records every 10 seconds.

3.2 Case design

To compare the calculation accuracy and speed of RF model to those of the experiment and numerical simulation, the model of the case based on the sand experiment is built in ANSYS. ANSYS is a general purpose finite element modeling package for numerically solving a wide variety of mechanical problems, including the fluid and heat transfer. The geometry and meshing of the case for CFD simulation are shown in Figure 3 respectively. As the sandbox is symmetric, the built geometry is half of it to reduce the mesh number and increase the calculation speed. The size, material and boundary conditions of the case for CFD simulation are kept the same with the sandbox experiment. There are 640,000 meshes and the size of simulating time step is 900s.

In the RF model, the excess soil temperatures under the initial heat pulse are simulated by the ANSYS model to further calculate the response factors. Besides, the soil temperature calculated by the numerical simulation is also based on this ANSYS model.

4 RESULTS AND ANALYSES

As for the studied case, the calculation results of RF model are demonstrated in this section. First, the accuracy of RF model is verified by the numerical simulation and validated by the sandbox experiment. Then, for the calculation of soil temperatures in a long term, when the time tends to be infinite, the response factors can be approximate to 1 to save the calculation time. The accuracy of RF model with the approximate response factors are compared to that with the real response factors. At last, the long-term soil temperatures calculated by RF model with approximate response factors are compared with those by the numerical simulation.

4.1 Validation by the numerical simulation and sandbox experiment

![Figure 4: The response factors of P1~P6 during the initial 96 time steps](image)

![Figure 5: The excess soil temperature comparison of P1~P6 calculated by three different methods](image)
To calculate the soil temperatures by RF model, the response factors at different time steps should be calculated in advance. The response factors of P1~P6 during the initial 96 time steps are shown in Figure 4. For the points next to the U pipe, the response factors first increase rapidly and then decrease, like that the response factor of P1 peaks at 67.16 at the 3rd time step. For the points far from the U pipe, the response factors keep increasing slowly, like the P6. The response factors of P1~P6 at the 96th time steps range from 0.42~4.53.

Based on the response factors, the soil temperatures of P1~P6 of RF model are calculated and compared with those of the numerical simulation and sandbox experiment, which is shown in Figure 5. Due to the heat of constant inlet temperature, the soil temperatures of P1~P6 increase. The soil temperature differences of P1~P6 between the RF model and the numerical simulation are 0.01°C~0.34°C at the 96th time step. And, the soil temperature differences of P1~P6 between the RF model and the experiment are -0.21°C~0.69°C at the 96th time step. When the meshing of model in CFD simulation becomes finer, the accuracy of RF model can be furtherly improved.

4.2 The accuracy of RF model with the approximate response factors

Taking P13~P19 as examples, the soil temperatures during 1800 time steps are calculated based on RF model. The response factors of P13~P19 during 0~1800 time steps are calculated under the initial heat pulse, as shown in Figure 6. The maximum response factors of P13, P14, P15 and P16 are respectively 67.20 at the 3rd time step, 39.29 at the 4th time step, 11.26 at the 11th time step and 3.82 at the 41th time step. All the response factors at the 900th and the 1800th time steps are respectively 1.030~1.048 and 1.030~1.035, which are the real values. It shows that when the time becomes longer, the variation of response factor becomes gentle and its value tends to be 1. To save the calculation time, the response factors of P13~P19 during 900~1800 time steps are assumed 1, which are the approximate values. So, the CFD simulation of response factors only takes 900 time steps, saving half of the calculation time steps.

The soil temperatures of P13~P19 calculated by the RF model with the real and approximate response factors are illustrated in Figure 7. The approximation of response factors only cause very small errors of soil temperatures. The relative errors at P13 are less than 0.27%.

4.3 Soil temperature calculation in long term

Because of the good accuracy, the approximate response factors are used by RF model to calculate the soil temperature of P13~P19. The calculation results are compared with those by numerical simulation, which are shown in Figure 8. The temperature difference between RF model and numerical simulation are only less than 0.24°C. The relative errors of the soil temperatures between RF model and numerical simulation at the 1800th time step are 1.86%~3.94%, they show the RF model has a very good accuracy.
The calculating time consumption between RF model and numerical simulation is compared in Table 1. For RF model, it takes 38min on CFD simulation for response factors and 67 seconds for the following soil temperature calculation. Therefore, the total time consumption of RF model is about 39min and 7sec, while that of the numerical simulation is about 120min. The RF model only consumes 30% time of the numerical simulation. When the soil temperature calculation becomes much longer to 10 years, the time consumption of RF model is about 255min22sec, but the time consumption of numerical simulation increases greatly to 23360min. The RF model only consumes 1% time of the numerical simulation. Consequently, the advantage of RF model on time saving becomes more obvious for a long-term soil temperature calculation.

![Graph showing soil temperature differences](image)

**Figure 8** Soil temperatures calculated by numerical simulation and RF model

<table>
<thead>
<tr>
<th>Time steps</th>
<th>Time (Day)</th>
<th>Calculating time consumption</th>
</tr>
</thead>
<tbody>
<tr>
<td>1800</td>
<td>19</td>
<td>39min7sec</td>
</tr>
<tr>
<td>350400</td>
<td>3650</td>
<td>255min22sec</td>
</tr>
</tbody>
</table>

### Table 1  Time consumption between RF model and numerical simulation

<table>
<thead>
<tr>
<th>Time steps</th>
<th>Time (Day)</th>
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<tr>
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</tr>
<tr>
<td>350400</td>
<td>3650</td>
<td>255min22sec</td>
</tr>
</tbody>
</table>

### 5 CONCLUSION

To improve the calculating speed and accuracy of soil temperature around the GHE, the refined RF model based on the response factors is proposed. It combines the heat transfer inside and outside the U pipe through the pipe wall temperature and the heat flux of U pipe. To validate the RF model, the sandbox experiment is built up and the model of CFD simulation is also established based on it. The results and analyses are as follows:

1. The soil temperatures of P1~P6 during 96 time steps calculated by RF model have small temperature difference with those by the numerical simulation and the experiment. The soil temperature differences between the RF model and the numerical simulation are 0.01°C ~0.34°C at the 96th time step. And, those between the RF model and the experiment are -0.21°C ~0.69°C at the 96th time step.

2. The RF model with approximate response factors has the nearly the same accuracy with that with the real response factors. Their relative errors of soil temperature at P13 are less than 0.27%.

3. The RF model has a good accuracy by the verification with the numerical simulation. The relative errors of the soil temperatures between RF model and numerical simulation at the 1800th time step are 1.86%~3.94%. What's more, RF model has a fast speed than the numerical simulation, consuming 30% time of the numerical simulation for the soil temperature calculation with 1800 time steps and consuming only 1% time of the numerical simulation for the soil temperature calculation with 350400 time steps.

### ACKNOWLEDGMENTS

The authors gratefully acknowledge the supports of Innovative Research Groups of the National Natural Science
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**NOMENCLATURE**

\[ \rho = \text{density (kg/m}^3) \]
\[ \theta = \text{excess temperature (°C)} \]
\[ c = \text{specific heat capacity (J/(kg \cdot °C))} \]
\[ d = \text{inner diameter of the U pipe (m)} \]
\[ F = \text{pipe area (m}^2) \]
\[ h = \text{convective heat transfer coefficient (W/(m}^2 \cdot °C)) \]
\[ I = \text{length of the U pipe (m)} \]
\[ m = \text{mass flow rate of the fluid (kg/s)} \]
\[ Q = \text{heat flux (W)} \]
\[ Y = \text{response factor} \]
\[ V = \text{Volume (m}^3) \]

**Subscripts**

\[ f = \text{fluid} \]
\[ in = \text{inlet of U pipe} \]
\[ k = \text{number of the soil layers} \]
\[ n = \text{number of the different heat sources} \]
\[ out = \text{outlet of U pipe} \]
\[ p = \text{soil point p} \]
\[ wall = \text{pipe wall} \]

**REFERENCES**


New methods to spatially extend thermal response test assessments

Jasmin Raymond  
Lorenzo Perozzi

Michel Malo  
Erwan Gloaguen

Louis Lamarche  
Carl Bégin

ABSTRACT

Thermal response tests (TRTs), used to evaluate the subsurface thermal conductivity when designing ground source heat pump systems, are spatially limited to the vicinity of the borehole where a test is carried out. The subsurface is heterogeneous and the thermal conductivity assessment provided by a TRT is likely to vary beyond the tested borehole. New methods have, therefore, been developed to extend subsurface assessments at the building site and the urban district scales. The first method relies on temperature profiles measured at equilibrium in ground heat exchangers that are reproduced with inverse numerical simulations to infer the terrestrial heat flow and the subsurface thermal conductivity beyond a first TRT. Inversion of temperature profiles was verified at a pilot site in the Appalachians where TRTs had been performed and showed a thermal conductivity estimate within less than 10% for both approaches. The second method is based on geostatistical simulations to map the distribution of the subsurface thermal conductivity in areas where several ground source heat pump installations are anticipated. A first mapping exercise was achieved to the north of Montreal in the St. Lawrence Lowlands with four TRTs and ten laboratory measurements interpolated with sequential Gaussian simulations.

INTRODUCTION

Conventional thermal response tests (TRTs; Rainieri et al. 2011; Raymond et al. 2011a; Spitler and Gehlin 2015), with heated water circulating in a pilot ground heat exchanger (GHE), have been successfully implemented in the commercial geothermal sector. The method is mostly used to evaluate the subsurface thermal conductivity when designing ground-coupled heat pump (GCHP) systems.

The subsurface assessment is spatially limited to a single pilot GHE commonly drilled before the installation of the complete GCHP system. The test radius of influence is on the order of 1-2 m (3.3-6.6 ft; Raymond and Lamarche 2014a), while GCHP systems installed in the commercial sector can enclose tens of boreholes covering hundreds of squared meters where subsurface conditions are likely non-uniform. TRTs have additionally been unable to penetrate the residential geothermal sector, where drilling of a pilot GHE before the installation of a GCHP system enclosing few, likely less than ten, boreholes is uneconomical. Life-cycle cost analysis of GCHP systems has, in fact, shown that a TRT can be uneconomical for small buildings (Robert and Gosselin 2014).

Research with the objective to spatially extend assessments provided by TRT has therefore been carried out to develop new methods for evaluating the subsurface thermal conductivity distribution at the building site and the urban district scales. The methods do not replace TRTs but allows extending the subsurface thermal conductivity assessment as an alternative to repeating TRTs. The first application developed relies on inverse numerical modeling.
of temperature profiles and can be used at the site scale for large projects where tens of GHEs are installed on the same site. The second application is based on geostatistical simulations and can be used at the district scale where several small GCHP systems are planned to be installed in a given geological region. Both methods, which have been developed in the Appalachians and the St. Lawrence Lowlands geological provinces of Canada, are presented with respect to the scale at which they provide extension of subsurface thermal conductivity assessments.

**BUILDING SITE SCALE METHOD**

The method to extend TRT assessments at the building site scale can is to infer subsurface thermal conductivity changes in a field where many GHEs could be installed. A single TRT performed in a first GHE and temperature profiles measured in additional boreholes offer the observations to be reproduced with inverse numerical simulations aiming at identifying the terrestrial heat flow and, then, the subsurface thermal conductivity beyond the first GHE at the additional boreholes.

As a first step, the subsurface thermal conductivity is evaluated with a conventional TRT in a pilot GHE installed before the complete GHE field. A temperature profile undisturbed by the TRT and in equilibrium with the subsurface is additionally measured in the GHE. This can be done with a wired probe lowered in a pipe of the GHE. Upon lowering the probe, the water level in the GHE pipe rises and the depth-temperature measurements have to be corrected to adequately evaluate the vertical geothermal gradient along the borehole. This is achieved by subtracting the water level rise due to the probe and the wire volume to the depth measured with the pressure transducer.

Transient numerical simulations of heat transfer in the subsurface are performed to reproduce the temperature profile of the pilot GHE affected by climate warming at the surface and to identify the terrestrial heat flow at the site. In other words, climate changes that occurred over the past centuries are considered as a thermal perturbation in heat tracing simulations to reproduce the temperature signal of the subsurface. The model used for simulations can be unidimensional in the case of a flat surface or bi- or three-dimensional for sites with topographic variations. A 2D model with a surface slope was used for a first example (Figure 1a). Conductive heat transfer was solved numerically with the finite element method using the program Comsol Multiphysics (COMSOL AB 2011):

\[
\frac{\partial}{\partial x} \left( \lambda \frac{\partial T}{\partial x} \right) + \frac{\partial}{\partial z} \left( \lambda \frac{\partial T}{\partial z} \right) = \rho c \frac{\partial T}{\partial t}
\]  

(1)

where the medium thermal conductivity \( \lambda \, [M \cdot L \cdot t^{-3} \cdot T^{-1}] \) is assumed to be isotropic, \( \rho \, [M \cdot L^{-3}] \) is the density and \( c \, [L^2 \cdot t^{-2} \cdot T^{-1}] \) is the specific heat capacity. Heat generation due to the decay of radioactive elements inside the subsurface was neglected. The model boundaries were an imposed temperature at the surface varying with time to reproduce the paleoclimatic changes in ground surface temperature that occurred over the past six centuries (Beltrami et al. 2003), a constant heat flux at the bottom and adiabatic vertical side walls. The temperature values at the upper boundary, defined in the model with a step function (Figure 1b), were selected according to previous studies aiming to reconstruct the ground paleoclimate history in a proximal region with deeper boreholes (Chouinard and Mareschal 2007). The thermal conductivity measured with the TRT in the pilot GHE and the estimated volumetric heat capacity associated to the rock type were imposed to the subsurface for the simulations conducted for a period of 615 years preceding the measurement of the temperature profile. The initial temperature condition was calculated from the equilibrium geothermal gradient according to the surface thermal conductivity and the basal heat flow, which is unknown. A derivative-free solver (Conn et al. 2009) was used to minimize the sum of squared residuals calculated from the differences between measured and simulated temperatures to find the basal heat flow. This information is essential to continue to the next step and infer the subsurface thermal conductivity at the location of other GHEs.
Temperature profiles are measured in other GHEs of the same building site for the second step. This can be done again with a wired probe, in additional pilot GHEs or when the complete GHE field of a GCHP system is installed to verify if there are important subsurface thermal conductivity changes that could affect the initial design plans.

Similar numerical simulations are carried out to reproduce the temperature profiles in equilibrium with the subsurface. The model remains unchanged except that the basal heat flow identified in the first step is imposed at the bottom boundary and the subsurface thermal conductivity is considered as the unknown to be identified by the solver. The objective function of the solver is to minimize the sum of squared residuals between observed and simulated temperatures to find the subsurface thermal conductivity at the location of additional GHEs. The temperature profiles are sufficient to extent the subsurface thermal conductivity assessment of the first GHE without repeating TRTs in each GHE. The method is limited to building sites or regions with similar terrestrial heat flow and surface land use history affecting the ground surface temperature evolution.

**URBAN DISTRICT SCALE METHOD**

The method to extend TRT assessments with geostatistical simulations is to map the subsurface thermal conductivity distribution in an area where GCHP installations are anticipated, likely an urban district with many buildings having the potential to host geothermal systems. The subsurface thermal conductivity is initially determined in situ with TRTs at building sites using ongoing installations and combined with laboratory analysis of thermal conductivity of rock samples collected in surface outcrops to finally simulate the thermal conductivity distribution of the host rock at a shallow depth with stochastic methods constrained from the surface geological map.

In the following example, TRTs were performed with a heating cable inserted into GHEs of residential systems to minimize disturbance to the occupants (Raymond et al. 2011b, 2010). The GHE were approximately 45 m (147.6 ft) depth, which allowed using a continuous heating cable with a power source fewer than 1200 W (4095 Btu hr⁻¹). Heat was injected during 50-55 h followed by the monitoring of the thermal recovery period during 60-72 h. The temperature signals measured by fifteen submersible temperature loggers located along the heating cable inside a GHE pipe were analyzed with the infinite line-source equation to find the subsurface thermal conductivity at
depth based on recovery measurements using the slope method (Pehme et al. 2007):

$$m \approx \frac{\Delta T}{\Delta \ln(t/t_c)} \approx -\frac{q}{4\pi \lambda}$$  \hspace{1cm} (1)

where $q$ [M L t$^{-3}$] is the heat injection rate per unit length of heating cable and $m$ [T] is the slope determined from the late temperature increments $\Delta T$ plotted as function of the normalized logarithmic time $t/t_c$. The analysis of recovery data, where $t_c$ is the time after the heat injection stopped, helps to decrease the effect of power fluctuations and cable movements (Raymond et al. 2011b). The time normalization originates from the application of the superposition principle to the infinite line-source equation to reproduce the thermal recovery. The fifteen thermal conductivity evaluations obtained for each GHE with temperature signals at depth were averaged to find the global thermal conductivity of the host rock over 45 m depth, removing upper values inferred in the overburden. Such in situ measurements provide data points to interpolate with geostatistical simulations in a further step.

Laboratory analyses of rock samples collected in outcrops are additionally performed to complement the in situ data set. The rock samples are collected in representative outcrops of the study area. The modified transient plane source (TPS) method (Harris et al. 2014) was used in the laboratory for the case presented below. Samples were cut and saturated with water to apply the heat source on a flat surface that is heated to determine the thermal conductivity of the rock. The laboratory analyses are fast to achieve when compared to in situ assessments, which allowed increasing the amount of data points available for geostatistical simulations.

The interpolation of host rock thermal conductivity was achieved with sequential Gaussian simulations (SGS; Goovaerts 1997). A grid is initially drawn over the study area and cells with known thermal conductivity from in situ and laboratory measurements are considered static. A new cell is then selected at random. The Gaussian probability density function at the new cell is obtained by kriging from the measured values and the previously simulated values along the random path (Figure 2). This feedback loop, which retains previously simulated values as extra data points, is the key to ensure that the simulations are spatially correlated. This process is repeated until all the grid cells are visited once. Multiple simulations are generated by using different random paths and random seeds. Independent realizations of equivalent probability are combined to calculate the mean and the standard deviation defining the uncertainty. The final result obtained is a map showing the distribution of the host rock thermal conductivity and its uncertainty for a shallow depth up to 45 m, which can be used to design new GSHP systems without repeating TRTs.

RESULTS - BUILDING SITE SCALE METHOD

The numerical modeling method to inverse temperature profiles was used at a pilot site in Saint-Lazard-de-
Bellechasse where two conventional TRTs had been performed previously (Raymond et al. 2016). The GHEs had a depth of 139 m (456.0 ft) and were drilled at a distance of 10 m (32.8 ft) from each other in mudslates of the Armagh Formation in the Appalachians (Lebel and Hubert 1995). Results from the first TRT, indicating a subsurface thermal conductivity equal to 3.0 W m⁻¹ K⁻¹ (1.73 Btu hr⁻¹ ft⁻¹ °F⁻¹), was used to find the site basal heat flow and that from the second TRT, indicating a subsurface thermal conductivity equal to 3.5 W m⁻¹ K⁻¹ (2.02 Btu hr⁻¹ ft⁻¹ °F⁻¹), was used to verify the inverse numerical modeling method.

The temperature profile measured in each GHE showed an inverse geothermal gradient near the surface that is characteristic of climate warming (Figure 3). On top of that temperature signal, seasonal temperature variations can be detected in each GHE and a groundwater flow perturbation is present in the second GHE. Numerical simulations did not consider these phenomena and simulations, therefore, aimed at reproducing the bottom ~100 to 120 m of the temperature profiles. Numerical simulations were first conducted to invert the temperature profile of the first GHE to find the site basal heat flow (Figure 3a). The minimum and maximum bounds for the basal heat flow optimization were 20 and 50 mW m⁻² (6.3 and 15.8 × 10⁻³ Btu hr⁻¹ ft⁻²) determined from a heat flow map (Majorowicz and Grasby 2010) and the optimization started at the lower bound. Twenty-five iterations were necessary for the solver using the coordinate search method to converge toward a solution that decreased the sum of the squared residuals from ~13 to 9.3 × 10⁻² for the best fit scenario, indicating a basal heat flow toward 25 mW m⁻² (7.9 × 10⁻³ Btu hr⁻¹ ft⁻²). The initial temperature condition for the best fit scenario was a temperature gradient equal to 8.3 × 10⁻³ °C m⁻¹ (4.6 × 10⁻³ °F ft⁻¹).

Subsequent simulations were computed to invert the temperature profile of the second GHE to find the subsurface thermal conductivity at this location (Figure 3b). The minimum and maximum bounds for the optimization process were a subsurface thermal conductivity equal to 2.8 and 4.2 W m⁻¹ K⁻¹ (1.62 and 2.43 Btu hr⁻¹ ft⁻¹ °F⁻¹) and the optimization started at the lower bound. Twenty-four iterations were necessary for the coordinate search solver to converge toward a solution that decreased the sum of squared residuals from 2.5 × 10⁻¹ to 2.5 × 10⁻², indicating subsurface thermal conductivity converging toward 3.2 W m⁻¹ K⁻¹ (1.85 Btu hr⁻¹ ft⁻¹ °F⁻¹).
initial temperature condition for the best fit scenario was a temperature gradient equal to $7.8 \times 10^{-3}$ °C m$^{-1}$ ($4.3 \times 10^{-3}$ °F ft$^{-1}$). The thermal conductivity value obtained from inverse numerical modeling was within less than 10% to that measured with the TRT, which demonstrates the use of the method. Inversion of temperature profiles is not expected to be as accurate as TRT because of the uncertainty of temperature changes due to paleoclimates and the basal heat flow to impose at the model boundaries but appears to be sufficiently accurate to identify lateral subsurface thermal conductivity changes at the building site scale.

**RESULTS – URBAN DISTRICT SCALE METHOD**

The geostatistical method to interpolate the host rock thermal conductivity was used in a 35 km$^2$ (8649 acres) region to the north of Montreal in the St. Lawrence Lowlands geological province (Figure 4a; Perozzi et al. 2016), constituted of non-deformed Cambro-Ordovician sedimentary sequences (Globensky 1987). The area was divided in 35,000 cells, each covering 100 × 100 m (328 × 328 ft). Four data points from in situ measurements were acquired with TRTs in boreholes that intercept the Trenton, Black River and Chazy geological groups, as well as the Beauharnois Formation, indicating a thermal conductivity ranging from 2.4 to 4.2 W m$^{-1}$ K$^{-1}$ (1.39 to 2.43 Btu hr$^{-1}$ ft$^{-1}$ °F$^{-1}$; Table 1). The highest value was associated to the Beauharnois Formation with greater dolomite content while the Trenton, Black River and Chazy groups are mostly argillaceous limestones with a lower thermal conductivity. Ten outcrops samples were collected in those three groups for additional data points and laboratory analysis indicated a thermal conductivity ranging from 2.1 to 3.5 W m$^{-1}$ K$^{-1}$ (1.21 to 2.02 Btu hr$^{-1}$ ft$^{-1}$ °F$^{-1}$; Table 1). The in situ assessment and the laboratory analysis were complemented by twenty-seven synthetic data points to increase the spatial resolution and for the simulations to better reflect the geological setting. The synthetic data points were determined from the work of Nasr et al. (2015) and Sirois et al. (2015) that defined thermostratigraphic units in the St. Lawrence Lowlands with forty-five laboratory measurements of thermal conductivity, representing the complete sedimentary sequence and covering the entire sedimentary basin. The average thermal conductivity value specified for the synthetic data points of the Trenton, Black River and Chazy groups was 2.67 W m$^{-1}$ K$^{-1}$ (1.54 Btu hr$^{-1}$ ft$^{-1}$ °F$^{-1}$) and that of the Beauharnois Formation was 3.40 W m$^{-1}$ K$^{-1}$ (1.96 Btu hr$^{-1}$ ft$^{-1}$ °F$^{-1}$).

A total of ten independent SGS realizations (Figure 4b) were computed to map the thermal conductivity distribution of the host rock and combined to define an average realization (Figure 4c), with its standard deviation (Figure 4d) to evaluate the uncertainty of the simulated values. The thermal conductivity distribution reflects the geological map with values above 3.0 W m$^{-1}$ K$^{-1}$ (1.73 Btu hr$^{-1}$ ft$^{-1}$ °F$^{-1}$) in the upper left corner of the study associated to the Beauharnois Formation, while values below 3.0 W m$^{-1}$ K$^{-1}$ (1.73 Btu hr$^{-1}$ ft$^{-1}$ °F$^{-1}$) are dominantly associated to the Trenton, Black River and Chazy groups. The standard deviation of the realizations suggests an uncertainty of less than 0.5 W m$^{-1}$ K$^{-1}$ (0.29 Btu hr$^{-1}$ ft$^{-1}$ °F$^{-1}$) for the central region of the study area including most of the measured data points. Again, this method is not expected to be as accurate as TRTs because local subsurface heterogeneity can be difficult to sample, especially in urban areas with few outcrops, such that the map may not picture site scale heterogeneity. Nevertheless, the map is believed to be a useful tool to design small GCHP systems with low uncertainty in GHE length and avoid repeating TRTs at each building site. Screening design calculations can be achieved with the map data for large GCHP systems and a TRT may be performed afterward to validate design and reduce GHE length uncertainty.
Table 1. Thermal conductivity measurements used for geostatistical simulations

<table>
<thead>
<tr>
<th>Latitude UTM NAD 83</th>
<th>Longitude</th>
<th>Thermal conductivity W m⁻¹ K⁻¹ / Btu hr⁻¹ ft⁻¹ °F⁻¹</th>
<th>Method</th>
<th>Thermostratigraphic unit</th>
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<tbody>
<tr>
<td>45.519249</td>
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<td>45.547637</td>
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</table>

Figure 4  Sequential Gaussian simulations of the host rock thermal conductivity north of Montreal (Perozzi et al. 2016).

DISCUSSION AND CONCLUSIONS

Since the development of the first thermal response test (TRT) concept (Mogensen 1983) and the successful deployment of mobile apparatus (Austin III 1998; Gehlin 1998), research to evaluate the subsurface thermal conductivity in the context of ground source heat pump (GCHP) design has mostly focused on improving field and analytical aspects of the TRT method. Field improvements included the development of TRT units with power regulations to decrease fluctuations in heat injection rate (Witte et al. 2002), downhole temperature measurements with optical fibers (Acuña and Palm 2013; Acuña et al. 2011; Fujii et al. 2009, 2006) and wireless sensors (Martos et al. 2011; Rohner et al. 2005), as well as the use of a heating cable to avoid water circulation and reduce the power requirement (Raymond et al. 2015, 2010; Raymond and Lamarche 2014b). The test analysis has been improved to take
into account groundwater flow (Raymond et al. 2011c; Signorelli et al. 2007; Wagner et al. 2013), thermal recovery (Raymond et al. 2011a, 2011b), variable heat injection rates (Beier and Smith 2003), parameter estimation (Choi and Ooka 2015; Wagner and Clauser 2005) and uncertainty (Witte 2013). The research presented in this manuscript is to address an additional problematic related to the spatial limitation of TRT. It is recognized that heat injection tests carried in a borehole have a limited radius of influence and that the subsurface is non-uniform. New methods to extend TRT assessments beyond a single ground heat exchanger (GHE) can, therefore, benefit to complex GCHP design. The concepts presented are indirect methods to evaluate changes in subsurface thermal conductivity at the building site and the urban district scale. This work is some of the first attempts to inverse temperature profiles to infer the subsurface thermal conductivity beyond a first TRT and map the distribution of the subsurface thermal conductivity with geostatistical simulations.

The first example was to verify the inverse numerical modeling approach at a pilot site in the Appalachians. Results obtained with preliminary simulations presented here highlight the potential of the method that could be improved to simulate groundwater flow and better reproduce the observed temperature signal affected by advective heat transfer. The second example was achieved in a populated region of the St. Lawrence Lowlands to the north of Montreal, where the installation of residential GSHP systems is planned. It was possible to perform TRTs at ongoing residential sites and extent the subsurface thermal conductivity assessments beyond the sites for the upcoming installations. The test costs are distributed over several installations, opening new markets for TRTs, such as the residential geothermal sector. More field data is needed to continue this study by validating the thermal conductivity distribution map with additional TRT to be performed at upcoming installations. Additional data could further be used with different geostatistical simulation scenarios to recommend a minimum density of field data providing reliable prediction of the host rock thermal conductivity.

Both the inverse numerical modeling and the geostatistical methods were used independently to address a common problematic in the current study but could be combined in future work. At the building site scale, when many temperature profiles are collected to infer the subsurface thermal conductivity, geostatistical simulations can be used to infer the thermal conductivity between the extremities of a GHE field. At the urban district scale, the inversion of temperature profiles can provide a low-cost method to evaluate the subsurface thermal conductivity and add static data points to improve geostatistical simulations. While the TRT field and analytical methods are getting more complete and advance, such ideas can help to reduce the spatial limitation of the TRT, one of the next challenges to address with scientific research.

**ACKNOWLEDGMENTS**

The Banting Postdoctoral Fellowship program and the Natural Sciences and Engineering Research Council of Canada are acknowledged for funding this research. The participation of Versaprofiles and Marmott Energies are also recognized for a great contribution to this project.

**NOMENCLATURE**

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Definition</th>
<th>Subscript</th>
</tr>
</thead>
<tbody>
<tr>
<td>c</td>
<td>specific heat capacity [L^2 M T^{-1} t^{-2}]</td>
<td>c = cooling</td>
</tr>
<tr>
<td>m</td>
<td>slope [T]</td>
<td></td>
</tr>
<tr>
<td>q</td>
<td>heat transfer rate per unit length [ML^{-1} T^{-1}]</td>
<td></td>
</tr>
<tr>
<td>λ</td>
<td>thermal conductivity [ML^{-1} T^{-1}]</td>
<td></td>
</tr>
<tr>
<td>ρ</td>
<td>density [ML^{-3}]</td>
<td></td>
</tr>
</tbody>
</table>

[L M T t] are used to denote units of length, mass, temperature and time, respectively.
REFERENCES


A Three Dimensional Numerical Simulation of Double-U Pile Heat Exchangers

Zhonghao Wang  Pingfang Hu  Lu Xing  Na Zhu  Fei Lei

ABSTRACT
The Ground Source Heat Pump system with pile heat exchangers has been used in engineering in recent years. In this paper, a three-dimensional numerical model of double-U pile heat exchangers is established which was verified through the comparison with ground thermal response test results. The influencing factors for the heat exchange performance of double-U pile heat exchangers are analyzed including pile depth, inlet temperature, initial ground temperature and ground thermal conductivity. An improved method is proposed on the basis of the cylindrical source model, combining with the numerical model. The improved method which is simpler than general analytical model can be used to obtain the ground thermal properties and thermal resistance of energy pile.

INTRODUCTION
Nowadays, ground source heat pump technology is the most effective way in the utilization of shallow geothermal energy, and it has been widely used in the world. The ground source heat pump system with pile heat exchangers has become the research hot spot as it has advantages in making full use of land area, reducing the initial investment and enhancing heat transfer performance of the heat exchangers.

The transient heat transfer period in energy pile is much longer than that in vertical borehole because the diameter of the pile is much larger and the depth is shorter than those of vertical borehole. Thus the error would be large if the conventional model is still used which assumes the heat transfer process in the borehole is a steady one (Hu, et al. 2014). Recently, some methods have been developed to study the thermal performance of pile heat exchangers. Zarrella and Carli (2013) and Zarrella, et al (2013) introduced the CaRM numerical model for spiral coil and U type pile heat exchangers and compared the thermal performance between them. Furthermore, Park, et al (2013) developed a numerical model for pile heat exchangers with ABAQUS software, and validated its reliability with a thermal response test. Li and Lai (2012) analyzed and compared the spiral-line model and the cylindrical-surface model, and found that both model can be used to describe the heat transfer of energy piles.

In this paper, a three-dimensional numerical model of pile heat exchangers is established based on COMSOL Multiphysics. This model can accurately analyze the heat conduction between the pile and the soil, and the coupled heat transfer of the pipe wall. It is verified in comparison with result of ground thermal response test. The numerical model of double-U pile heat exchangers is established, and the influencing factors for the heat exchange performance of double-U pile ground heat exchangers are analyzed based on the model. An improved method is proposed on the basis of the cylindrical source model, combining with the numerical model. The inversion result was compared with that of line source model and composite cylindrical source model.

Pingfang Hu (pingfanghu21@163.com) is a professor in building energy engineering and Zhonghao Wang is a graduate student at Huazhong University of Science and Technology.
ANALYTICAL MODEL

As the soil acts as the heat source of the GSHP systems, it is necessary to obtain its thermophysical parameters in the design process. The parameters have significant effect on calculation of number and depth of boreholes, and the initial investment of the systems. Studies show that about 10% error of thermal conductivity will cause 4.5%~5.8% error of pile heat exchanger length (Kavanaugh 1998). At the same time, the error of the thermal properties can also lead to the unbalance of the designed load and the actual load, which will then affect the operational performance of the systems. Therefore, the rational heat transfer model needs to be selected to accurately calculate the thermal physical parameters of the soil.

Infinite line source model and cylindrical source model

The heat transfer model for heat exchanger can be divided into the internal and the external heat transfer model. As for the common borehole heat exchangers, the geometry size and heat capacity inside the borehole are quite small, and generally the temperature can achieve a relatively stable stage in several hours. Thus, the heat exchange inside the borehole can be approximately considered as steady heat transfer process. The arithmetic average of the inlet and outlet fluid temperature can be regarded as average temperature of circulating fluid in pipe for both temperature results are close for regular operating conditions.

The internal heat transfer model mainly includes one-dimensional, two-dimensional and quasi three dimensional models. One-dimensional model ignores the influence of the pipe spacing and some other factors, and the calculating error may be large. And the calculating process of the quasi three dimensional heat transfer model is more complex compared with other two models. Thus, the two-dimensional heat transfer model is commonly used in practical engineering (Loveridge and Powrie 2014).

The two-dimensional model neglects the heat transfer in the axial direction, and analyzes the effect of the geometric configuration on the cross section. The thermal resistance between the fluid and the borehole wall can be obtained through the model. The borehole thermal resistance for vertical double U type heat exchangers can be obtained through equation (1) (Hellstrom 1991).

\[
R_b = \frac{1}{4} \left( \frac{1}{2\pi \lambda_s} \ln \left( \frac{d^4_s}{4d_s d_i^3} \right) + \frac{\lambda_y}{\lambda_y + \lambda_s} \ln \left( \frac{d^4_y}{d^4_y - d^4_s} \right) + \frac{1}{2\pi \lambda_p} \ln \left( \frac{d^4_o}{d^4_i} \right) \right)
\]  

(1)

As for the external heat transfer process, the infinite line source model and cylindrical source model are commonly used, and the average fluid temperature can be expressed in equation (2) and equation (3) (Ingersoll and Plass 1948).

\[
T_r = T_0 + q_i \left[ R_b + \frac{1}{4\pi \lambda_s} \cdot B \left( \frac{d^2_o \rho_s c_s}{16 \lambda_s^2 \tau} \right) \right]
\]

(2)

\[
T_r = T_0 + q_i \left[ \frac{G(F_o, 1)}{\lambda_s} + R_b \right]
\]

(3)

where \( F_o = \frac{4\lambda_s \tau}{\rho_s c_s d_s^2} \).

Composite cylindrical source model

Due to the particularity of pile heat exchangers, such as the large heat capacity of pile foundation, more time will be required to reach the steady state compared with the common vertical borehole heat exchangers. If the heat
transfer inside the borehole is treated as steady heat exchange process, the error would be large. The large pile
diameter and the small depth limit the utilization of composite line source model. A composite cylindrical source
model was established based on the composite line source model and the cylindrical source model (Hu, et al. 2014).

According to the composite cylindrical source model, the borehole thermal resistance varies with time, and it can
be expressed with G function and Fourier number (Hu, et al. 2014).

\[
R_e(\tau) = \frac{G(F0r,1)}{\lambda_x} - \frac{G(F0r',1)}{\lambda_x}
\]

where \( F0' = \frac{a' \tau}{r_w^2} = \frac{b'_w \tau}{r_w^2 \rho_g C_g}, \)
\( F0'' = \frac{a' \tau}{r_b^2} = \frac{b'_b \tau}{r_b^2 \rho_g C_g}. \)

Then the average fluid temperature can be expressed in the equation (5):

\[
T_f(\tau) = T_s + q_l \left( \frac{G(F0,1)}{\lambda_x} + \frac{G(F0',1)}{\lambda_x} - \frac{G(F0'',1)}{\lambda_x} \right)
\]

The model can be used in analysis of performance and TRT for energy pile.

**EXPERIMENTAL TEST**

An experimental test is conducted in Wuhan, China. As the underground water level is relatively low, there will
be no obvious groundwater flow in the process, and then the influence of groundwater seepage can be ignored. The
length of tested pile is 45 m, and its diameter is 600 mm. The double U HDPE pipe is used in the test, and its outer
diameter and inner diameter is 25 mm and 20 mm respectively. The heat exchanger is in the pile (see Fig.1). The
thermal conductivity of pipe wall is 0.51 W/(m•K). The backfill material is cement concrete. The average initial
temperature of the soil is 19.8 °C, the average flow rate of circulating water is 1.3 m³/h. The test is performed for
about 48h with 3.5 kW heating power. The temperatures are measured at 1-min intervals. At first the heat transfer
between the fluid and the soil is not sufficient so the inlet and outlet fluid temperature difference is relatively small.
After an initial transient process the heat transfer is close to stable state then the temperature difference tends to be
nearly constant (see Fig.2).

**NUMERICAL MODEL**

The three dimensional numerical model for pile heat exchangers is established based on COMSOL Multiphysics
which combines the Finite Element Method (FEM) and the Computational Fluid Dynamics (CFD). The heat
conduction between the pile and the soil, and the coupled heat transfer of the pipe wall can be analyzed accurately by
this numerical model (Go, et al. 2014).
The parameter setting

As the model is built based on experimental test, parameter settings should be in accordance with the test condition. Pile depth is 45 m. Pile diameter, outer and inner diameter of the pipe are 0.6, 0.025 and 0.020 m respectively. Table 1 shows the thermal physical parameters of the soil, backfill material and the pipe. The calculation of thermal physical properties of the soil is based on the composite cylindrical source model which has been verified (Hu, et al. 2014). The backfill material is cement concrete, and the initial soil temperature is 19.8 °C.

<table>
<thead>
<tr>
<th>Table 1. Thermal physical parameters</th>
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<tbody>
<tr>
<td>Thermal Conductivity (W/(m·K))</td>
</tr>
<tr>
<td>Soil</td>
</tr>
<tr>
<td>Cement concrete</td>
</tr>
<tr>
<td>HDPE</td>
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</tbody>
</table>

The inlet fluid temperature is the same as that in the experimental test, and it is defined as the global variable (T_in) which is imported into the model. The average fluid flow rate is 1.3m³/h.

The free tetrahedron mesh is adopted in the model. The three different kinds of meshing number are selected, and the temperature variation of outlet fluid was compared. According to the error of three meshing density, meshing density (25681) was adopted for the model.

Validation of the numerical model

The model can be validated through the comparison of the outlet fluid temperature between numerical simulating results and the measured values. Figure 2 shows the comparison of the outlet fluid temperature for 48 hours. It is obviously shown that the numerical simulating results and the measured values are in good agreement, especially after the outlet fluid temperature reaches the balance. The deviation at the short term may be mainly resulted from the power fluctuation. In the test, the heating power fluctuates sometimes due to the power supply situation on site. The maximum relative error occurs in the initial stage, and it is limited within 3% after 10 hours’ operation. Therefore, it’s proven that the three-dimensional numerical model is reliable.

Figure 2  Outlet fluid temperature.

AFFECTING FACTORS FOR PERFORMANCE OF PILE HEAT EXCHANGERS
The heat exchange performance of pile heat exchangers is related to the depth of pile and some other factors. The three dimensional numerical model can be used for analyzing the effect of various factors on heat transfer performance. It can provide reference for the design and operation of pile heat exchangers.

The software COMSOL is used to establish the double U type heat exchange model. The simulating time is set as 3000 min. Pile diameter, outer and inner diameter of the pipe are 0.6 m, 0.025 m and 0.020 m respectively. The thermal physical parameters of the surrounding soil, the backfill material and the buried pipe are as shown in Table 2. The operation mode is cooling.

<table>
<thead>
<tr>
<th>Table 2. Thermal physical parameters</th>
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<tbody>
<tr>
<td>Soil</td>
</tr>
<tr>
<td>Thermal conductivity: 2.5 W/(m·K)</td>
</tr>
<tr>
<td>Specific heat capacity: 1200 J/(kg·K)</td>
</tr>
<tr>
<td>Density: 2000 kg/m³</td>
</tr>
<tr>
<td>Backfill material</td>
</tr>
<tr>
<td>Thermal conductivity: 2.0 W/(m·K)</td>
</tr>
<tr>
<td>Specific heat capacity: 850 J/(kg·K)</td>
</tr>
<tr>
<td>Density: 2800 kg/m³</td>
</tr>
<tr>
<td>Pipe</td>
</tr>
<tr>
<td>Thermal conductivity: 0.49 W/(m·K)</td>
</tr>
</tbody>
</table>

**Pile depth**

According to Figure 3, the heat exchange rate per unit length decreases while the pile depth increases. Although the longer pile has bigger exchanging area and the total exchange rate as well, the outlet fluid temperature is reduced which leads to the enhancement of thermal effect between inlet fluid and outlet fluid.

**Soil thermal conductivity**

As shown in Figure 4, the heat exchange rate per unit length increases as the soil thermal conductivity increases. And the maximum variation ratio of the heat exchange rate can reach about 10%.

**Inlet fluid temperature**

As shown in Figure 5, the heat exchange rate per unit length increases as the inlet fluid temperature increases, and their relationship tends to be linear. When the pile depth takes 35 m, the linear relationship can be expresses as the following equation: \( q_l = 3.7T_{in} - 72.7 \). Thus, if the inlet fluid temperature rises by 1°C, the heat exchange rate per unit length is supposed to increase about 3.7 W/m.

Increment of the inlet fluid temperature will lead to the larger temperature difference between the fluid and the soil, the heat exchange rate per unit length will also be increased. However, increasing the inlet fluid temperature will reduce the efficiency of the heat pump.

**Initial soil temperature**

As shown in Figure 6, the heat exchange rate per unit length decreases linearly as the initial soil temperature increases. When the initial soil temperature rises by 1 °C, the heat exchange rate per unit length will decrease in the range from 3.5 W/m to 4.5 W/m.
IMPROVED METHOD

Here we proposed an improved method to obtain the thermal property of soil and thermal resistance of energy pile. The numerical model established in this paper can not only simulate the inlet and outlet fluid temperature, but also the average temperature of the borehole wall at real time, and then the borehole thermal resistance $R_b(\tau)$ under unsteady state can be obtained through the following two equations.

The heat exchange rate per unit length and the borehole thermal resistance can be expressed in the equation (6) and equation (7), respectively.

\[ q_i = \frac{cm(T_{in} - T_{out})}{H} \]  

\[ R_b(\tau) = \frac{T_f - T_b}{q_i} \]  

Then the equation (3) is used to obtain the thermal properties of soil. Equation (3), (6) and (7) constitute the main content of an improved method. The borehole thermal resistance could be obtained by a simple equation (7) instead of a complicated calculation in general analytical method, and the borehole wall temperature $T_b$ is obtained by numerical simulation. Then the soil thermal conductivity can be inversed by equation (3).

Generally the temperature sensors are installed at the inlet and outlet of the pipe for experimental test, and it is definitely not feasible to install the sensors widely to cover the borehole wall in the practical engineering, so the field test can’t provide accurate average temperature of the borehole wall which exposes its disadvantage compared with
numerical simulation. While solving the thermal properties of soil, the analytical model should be adopted with complicated calculations obtaining the borehole thermal resistance, like the equation (1) or equation (4).

Compared with other analytical model in previous sections, the improved method based on the cylindrical model and numerical model can be used to obtain the borehole thermal resistance easily under unsteady state. The method is also applicable for many other types of heat exchangers, including spiral heat exchangers.

According to the above formulas of different models, the line source model, composite cylindrical source model and the improved method are established respectively on the MATLAB platform. These three models are analyzed and compared based on the existing experimental test data. Among all the models, the improved method is only used to invert the thermal conductivity of soil. The results are shown in Table 3.

According to Table 3, the result of line source model is quite different from that of the composite cylindrical source model. The main reason is that the former one linearize the heat transfer between the pile and the soil which may result in a major error, and the internal heat transfer can’t be handled as a steady-state progress. The inversion result of improved method is close to that of composite cylindrical source model. Thus, the improved method can be adopted for the designing of GSHP systems with pile heat exchangers, and its accuracy is high with a relative difference of about 4.6% compared with the result of composite cylindrical source model.

<table>
<thead>
<tr>
<th>Model</th>
<th>Line source model</th>
<th>Composite cylindrical model</th>
<th>Improved method</th>
</tr>
</thead>
<tbody>
<tr>
<td>Thermal conductivity W/(m·K)</td>
<td>1.87</td>
<td>2.17</td>
<td>2.27</td>
</tr>
</tbody>
</table>

Current studies on the pile heat exchanger are mainly based on analytical or dimensional simulation method. This paper developed a method combining a three-dimensional model with analytical model. Thematically speaking, the borehole thermal resistance could be obtained by a simple equation (7) instead of a complicated calculation in general analytical method, and the borehole wall temperature $T_b$ is obtained by numerical simulation. So the calculation of the borehole thermal resistance is simpler and the method is suitable for wider application compared with current analytical model. That is, the method can be used in the calculation of vertical heat exchanger and spiral heat exchanger.

CONCLUSION

A three-dimensional numerical model of pile ground heat exchangers is established which can accurately analyze the heat conduction between the pile and the soil, and the coupled heat transfer of the pipe wall. The model is utilized to study the relationship between heat exchange performance and affecting factors, including pile depth, inlet temperature, ground initial temperature and ground thermal conductivity.

The improvement of this paper is to present a new method, which combines a three-dimensional numerical model and combined with the cylindrical source model. The calculation of the borehole thermal resistance using this method is simpler than analytical model, and the method is suitable for wider application compared with current analytical model. That is, the method can be used in the calculation of vertical heat exchanger and spiral heat exchanger.

ACKNOWLEDGMENTS
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**NOMENCLATURE**

\[ R = \text{Thermal resistance (m·K)/W} \]
\[ d_i = \text{Pipe inner diameter (m)} \]
\[ d_o = \text{Pipe outer diameter (m)} \]
\[ d_b = \text{Borehole diameter (m)} \]
\[ d = \text{Distance between two legs of the U tube (m)} \]
\[ \lambda = \text{Thermal conductivity (W/(m·K))} \]
\[ b = \text{Coefficient of heat exchange between fluid and inside of the tube (W/(m²·K))} \]
\[ T = \text{Temperature (°C)} \]
\[ q_i = \text{Heat exchange rate (W/m)} \]
\[ Fo = \text{Fourier number} \]
\[ \tau = \text{Time (s)} \]
\[ c = \text{Heat capacity (J/(kg K))} \]
\[ m = \text{Mass flow rate (kg/s)} \]
\[ H = \text{Pile depth (m)} \]
\[ r_e = \text{Effective radius of the pipe (m)} \]
\[ r = \text{Radius (m)} \]

**Subscripts**

\[ b = \text{Borehole} \]
\[ g = \text{Backfill material} \]
\[ s = \text{Soil} \]
\[ f = \text{Fluid} \]
\[ p = \text{Pipe} \]
\[ in = \text{Inlet} \]
\[ out = \text{Outlet} \]

**REFERENCES**


Measurement of internal and effective borehole resistances during thermal response tests

Louis Lamarche  Jasmin Raymond
Claude Hugo Koubikana Pambou

ABSTRACT

In a conventional thermal response test (TRT), the main parameter evaluated is the bulk subsurface thermal conductivity surrounding the borehole. It is also possible to evaluate the borehole thermal resistance. Several approaches were proposed in the literature to evaluate the possible combination of these two parameters. For example, it is often suggested to measure the temperature during the injection and the recovery periods, where the thermal conductivity is found with the recovery response whereas the borehole resistance is calculated with injection measurements. For this calculation, some authors suggested to use different means for the borehole temperature considering the asymmetric temperature distribution along the pipe legs that affects the borehole resistance. Some confusion about the borehole resistance that should be obtained may come from the difference between the 2D borehole resistance and the effective (3D) borehole resistance taking into account the internal heat transfer between the pipe legs inside the borehole. In practice, the latter one should be used in a design algorithm since it provides a more representative approach. In many cases, the difference between the two is rather small. However, since the borehole length is becoming an important variable to optimize, this difference in borehole resistance may represent a factor to better assess in the design of future systems. This effective resistance depends on the 2D borehole resistance, the water flow rate, the length of the borehole and the so called "internal resistance". To our knowledge, the in-situ assessment of this internal resistance has never been achieved. In this paper, we present our first investigation of a method that can be used to evaluate both the 2D borehole resistance ($R_b$) and the 2D internal resistance ($R_a$). The method uses the temperature at the bottom of the borehole at the same time as the inlet and outlet temperatures that are measured in a conventional TRT. Interesting results were found by comparison with theoretical resistances calculated with the multipole method.

INTRODUCTION

Conventional thermal response tests (TRTs), successfully implemented in the commercial geothermal sector, is to inject heat in a borehole and measure the temperature response from the heat pulse. Heat is normally generated by an electrical resistance outside the borehole and transported through a heat-transfer fluid, usually water flowing inside the borehole. Heat can also be generated using a heated cable inside the borehole (Raymond et al. 2010). Heat pumps have alternatively been used with heated water circulating in a pilot ground heat exchanger (GHE). The method is mostly used to evaluate the subsurface thermal conductivity when designing ground-coupled heat pump (GCHP) systems but most of the time an evaluation of the borehole resistance is provided during the test. This last parameter is characteristic of the borehole heat transfer performances and can be assessed for quality control purposes.
Several methods have been proposed in the literature in order to evaluate these two parameters (Spitler and Gehlin 2015). The most common approach is to evaluate the thermal conductivity using the slope of the mean temperature increase with respect to the logarithm of the time. This formula comes from the well-known expression of the infinite line source (Carslaw and Jaeger 1959). Indeed, if we assume that the heat is released at the origin of the borehole and that it depends only of the radial conduction, which is a valid approximation for the time scale of a TRT test, the mean fluid temperature is given by:

$$T_f(Fo) - T_o = \frac{q'_{inj}}{4\pi k_s} \int_1^{\infty} \frac{e^{-u}}{u} du = q'_{inj} \left( \frac{R_s}{E_1(1/(4Fo))} + R_b \right)$$

Where $E_1$ is the exponential integral, $q'_{inj}$ is the amount of heat per meter injected during the test, $R_s$ is the borehole resistance and $Fo$, the Fourier number based on the borehole radius. This expression is only valid for Fourier number larger than 5. For these values, it is known that the exponential integral is proportional to the natural logarithm of time. It follows that:

$$k_s = \frac{q'_{inj}}{4\pi m}, \quad R_b = \frac{b - T_o}{q'_{inj}} - \frac{\ln(4\alpha/\eta_b^2) - \gamma}{4\pi k_s}.$$

where $m$ is the slope and $b$ is the intercept of the linear approximation (Fig. 1).

Some authors (Raymond et al. 2011) have suggested separating the TRT in two parts: an injection period that is done the same way as in the classical method followed by a thermal recovery period where no heat is injected. Since the temperatures become independent of the borehole resistance in during the thermal recovery, it is suggested to use this period to evaluate the thermal conductivity and the heat injection period to evaluate the borehole thermal resistance. Others researchers (Austin et al. 2000) used optimization methods to evaluate the unknown parameters using parameter estimation algorithm. Using this approach, other equations than the infinite line source function can
be used for the thermal response factor. In all the previous expressions, $T_f$ represents the mean fluid temperature. Since most of the models neglect axial temperature variations, it is implicitly assumed that this mean temperature does not change with the depth and can be evaluated from the mean temperature at the exit of the borehole:

$$T_f(t) \approx \frac{T_{f, in} + T_{f, out}}{2} \quad (3)$$

For high flow rate, this approximation is generally representative. However, this assumption may not be valid for the case of a small flow rate and/or long boreholes, where the non-linearity of the temperature profile will be accentuated. For this reason, Marcotte and Pasquier (2008), based on numerical simulations, proposed to replace the mean fluid temperature (Eq. 3) by a so-called $p$-linear average defined as:

$$T_f(t) \approx \frac{p \left( |\Delta T_{in}|^{p+1} - |\Delta T_{out}|^{p+1} \right)}{(1 + p) \left( |\Delta T_{in}|^p - |\Delta T_{out}|^p \right)} + T_o \quad (4)$$

Where $\Delta T = T - T_o$. They found that using Eq. 3 in the analysis of a TRT can overestimate the borehole resistance whereas using Eq. 4, with $p \to -1$ gives a better estimate. Beier (2011), using an analytical modeling approach, found that this is indeed the case and that the use of the $p$-linear average, although not exact gives less error for the evaluation of the borehole resistance. Lamarche et al. (2010) confirmed, using numerical simulations, that the borehole resistance deduced when using Eq. 3 is higher than the borehole resistance but closely correspond to the effective borehole resistance. The concept of the effective borehole resistance was introduced by Hellström (1991) and is greater than the borehole resistance because the former takes into account the loss of performance due to the short-circuiting effect between the two segments of the U-tube inside the borehole. Hellström (1991) found two expressions for the effective resistance given by:

$$R_b^e = R_b \eta \coth(\eta) \quad (5)$$

$$\eta = \frac{H}{mC_p \sqrt{R_b R_a}} \quad (6)$$

for the case where the borehole temperature is uniform and another equation for the case when the heat flux is uniform along the borehole. In practice, neither assumption is strictly valid but most of the time it gives a good approximation of the internal heat transfer in real boreholes. Eq. 5 involves the expression $R_a$ which is called the internal borehole resistance and takes into account the short-circuiting effect between the two legs of the U-tube:

![Figure 2 Internal resistance pattern inside a typical borehole](image)
The borehole resistance \( R_b \) is found by taking the resistances \( R_1 \) and \( R_2 \) in parallel and the internal resistance is given by evaluating the equivalent resistance between both legs (Fig. 2). In the symmetrical case (\( R_1 = R_2 \)), it gives:

\[
R_b = \frac{R_1}{2}, \quad R_a = \frac{4R_bR_{12}}{(4R_b + R_{12})}
\] (7)

The concept of effective resistance was extended to double U-tubes arrangements by Zeng et al (2003). In a typical bore field design, just the effective resistance is needed to evaluate the total length of the field. However, if the parameters between the TRT and the real GSHP system vary, like the depth of the borehole, it could be interesting to evaluate both resistances (\( R_a \) and \( R_b \)) in a TRT test to adjust design parameters according to the field response. The following sections give some first results that were found to evaluate these parameters.

**MEASUREMENT ON THE TEMPERATURE AT THE BOTTOM**

In order to have a better understanding of the heat transfer inside the borehole, a measurement of the temperature profile is of great interest. Measurement of the temperature profile during TRTs were performed in by Fujii et al. (2009) and Acuña at al. (2011) using fiber optic sensors. Unfortunately, the apparatus to evaluate temperature with optical fiber is expensive. Lamarche et al. (2010) suggested an approach that can be used to evaluate these resistances assuming a prescribed temperature profile inside the borehole. The profile was first derived by Hellström (1991) assuming a thermal exchange between the fluid and a uniform temperature along the borehole. The expressions of the given profile at the bottom and at the exit are given by (Lamarche et al. 2010):

\[
\theta_{\text{bottom}} = \frac{T_{f,\text{bottom}}(t) - \bar{T}_b(t)}{T_{f}(t) - \bar{T}_b(t)} = \cosh(\eta) - \left( \frac{2\xi \sinh(\eta)}{\cosh(\eta) + \xi \sinh(\eta)} \right) \xi + \xi \sinh(\eta)
\] (8)

\[
\theta_{\text{out}} = \frac{T_{f,\text{out}}(t) - \bar{T}_b(t)}{T_{f}(t) - \bar{T}_b(t)} = \left( \frac{\cosh(\eta) - \xi \sinh(\eta)}{\cosh(\eta) + \xi \sinh(\eta)} \right)
\] (9)

With \( \eta \) is given by (6) and

\[
\xi = \frac{L}{m_p C_p 2R_b \eta} \quad , \quad \zeta = \frac{2R_b}{R_{12}}
\] (10)

Having measured value of \( \theta_{\text{bottom}} \) and \( \theta_{\text{out}} \), Eqs. 8 and 9 can be solved for the two unknown \( R_b \) and \( R_a \). They tested their method using numerical simulations while preliminary results in real TRT tests are presented here.

The test was performed at INRS laboratory facilities located in Québec City. The borehole was drilled in 10 m of unconsolidated till followed by 144 m of shale from the Sainte-Rosalie Formation in the St. Lawrence Lowlands geological province. A single U-pipe with no space clips was installed in the borehole filled with thermally enhanced grout to make the GHE (Table 1). Heat injection during the TRT was achieved for 81 h followed by 75 h of thermal recovery monitoring, where heat injection was stopped but water kept circulating in the GHE. The undisturbed subsurface temperature was measured before the test with a submersible probe lowered in the GHE and was 7.89 °C. Three temperature measurements in the descending pipe leg at depth of 50 m, 100 m and 150 m were achieved with submersible temperature data loggers during the TRT. The average heat injection rate was 62.7 W/m, creating a temperature difference of more than 7 °C between the inlet and outlet of the GHE. At the end of the test the average fluid temperature increased by up to ~20 °C.
Table 1. GHE configuration and TRT parameters

<table>
<thead>
<tr>
<th>$r_b$ (m)</th>
<th>$L$ (m)</th>
<th>$Q$ (l/s)</th>
<th>$x_c$ (m)</th>
<th>$r_{po}$ (m)</th>
<th>$r_{pi}$ (m)</th>
<th>$k_{grout}$ (W/mK)</th>
<th>$k_s$ (W/mK)</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.057</td>
<td>153</td>
<td>0.315</td>
<td>0.025</td>
<td>0.021</td>
<td>0.017</td>
<td>1.73</td>
<td>2.07</td>
</tr>
</tbody>
</table>

In order to solve Eqs. 8 and 9, we need the value of the borehole temperature $T_b(t)$. In practice, this is not easy to measure, so here this temperature is estimated using the infinite line source solution (Eq. 1) using the subsurface thermal conductivity found during the TRT. Even though the fluid and the borehole temperature are time dependent, in theory, Eqs 8 and 9, if valid, should be time independent. To verify that, the measured value of the normalized fluid temperature at the bottom and at the exit were plotted as function of time (Fig. 3).

The value becomes almost constant after approximately 7 h which gives a Fourier number of approximately 6, which is a typical value for the validity of steady-flux regime. In our calculations, the mean measured value of the normalized fluid temperature during the steady-flux regime was used in Eq. 8 and 9 in order to find $R_s$ and $R_b$ (Table 2). Results are compared with the calculated resistances using the multipole method (Claesson and Hellström 2011). The whole normalized temperature profile is calculated using the calculated resistances (Fig 4). On the same figure, the expected profile, assuming linear profile as it is done usually. In order to have a better idea on the validity of the method, measurements of the fluid temperature in the descending tube at 0 m, 50 m, 100 m and 150 m as well as temperature at 0 m in the ascending tube are superimposed to the expected normalized temperature in the steady-flux regime.
Figure 4  Normalized temperature using Hellstrom profile compared to measured temperature.

Table 2

<table>
<thead>
<tr>
<th>Measured values</th>
<th>Calculated values (multipole)</th>
</tr>
</thead>
<tbody>
<tr>
<td>$R_b$ (mK/W)</td>
<td>$R_a$ (mK/W)</td>
</tr>
<tr>
<td>0.099</td>
<td>0.897</td>
</tr>
</tbody>
</table>

Beier Profile

Beier (2011) proposed a modified equation to calculate the temperature profile inside the U-tube by coupling the borehole resistance network to the undisturbed ground temperature. He latter proposed a modified version taking into account the variation of the ground temperature but, in this manuscript, the uniform temperature case with symmetric configuration ($R_1 = R_2$) only was considered. The modified temperature profile is normalized with the ground temperature (Fig. 5):

\[
\theta_{down}(\bar{z},t) = \frac{T_f(t) - T_o}{T_{fi}(t) - T_o} = C_1 \exp(a_1 \bar{z}) + C_2 \exp(a_2 \bar{z})
\]  

(11)

\[
\theta_{up}(\bar{z},t) = \frac{T_f(t) - T_o}{T_{fi}(t) - T_o} = C_3 \exp(a_3 \bar{z}) + C_4 \exp(a_4 \bar{z})
\]  

(12)
The mathematical expressions for $C_1, C_2, C_3, C_4, a_1, a_2, a_3$ and $a_4$ are described by Beier (2011). They depend on two unknowns ($R_1$ and $R_{12}$) and known values of the fluid heat capacity, the borehole length and the ground resistance $R_{s,i} = 2R_s$ with $R_s$ given by Eq. 1. Expressing Eq. 11 and 12 at the bottom and the exit will, in theory, give us the two equations for the two unknowns. One of the advantage of the Beier’s profile is that it is not based on a vertically uniform borehole temperature, an assumption has been the subject of debates (Beier 2011, Marcotte and Pasquier 2009). However, one of the disadvantages is that the normalized profile found with Eqs. 11 and 12 is not time independent, even in the steady-flux regime. The short-circuiting effect in the delta equivalent circuit between both legs of the U-tube will follow two possible paths, a direct one through $R_{12}$ and an indirect one via the borehole wall temperature $T_b$ (Fig. 2). The equivalent resistance found is the internal resistance $R_a$. The resistance $R_{12}$ in Beier’s model should then be compared to $R_a$ in the delta circuit and not to $R_{12}$.

The equations must be solved with temperature measured at a given time to find the borehole resistance network. This practice can introduce errors since measurements are known to vary randomly and averaged values are always a better approach, when possible. In practice, it was observed that using temperature at different time in solving Eq. 11 and 12, gave large variations in values of $R_1$ and $R_{12}$. One could of course average the thermal resistances found. Instead, the approach used here was not to solve Eq. 11 and 12 at a given time but to minimize the least-square error defined by:

$$e = \sum_i \left( (\theta_{\text{down,meas}}(1,t_i) - (C_1 \exp(a_i) + C_2 \exp(a_i)) \right)^2 + \left( (\theta_{\text{up,meas}}(0,t_i) - (C_3 + C_4)) \right)^2$$

using a Nelder-Mead algorithm. From these thermal resistance results (Table 3), the expected normalized temperature profile given by Eq. 11-12 is compared with the measured temperature values (Fig. 6). As noted previously, normalized fluid temperatures are time dependent and the absolute temperatures are given for a specific time.
Figure 6 Absolute temperature using Beier’s profile compared to measured temperature at t = 30 hours.

<table>
<thead>
<tr>
<th>Measured values</th>
<th>Calculated values (multipole)</th>
</tr>
</thead>
<tbody>
<tr>
<td>$R_1$ (mK/W)</td>
<td>$R_{12}$ (mK/W)</td>
</tr>
<tr>
<td>0.2</td>
<td>6.37</td>
</tr>
<tr>
<td>$R_1$ (mK/W)</td>
<td>$R_{12}$ (mK/W)</td>
</tr>
<tr>
<td>0.184</td>
<td>2.55</td>
</tr>
</tbody>
</table>

The expected profile matches the experimental data event though the thermal resistances are different (Fig. 6). It is important to note that using resistance in Table 2 with the Beier’s profile or the resistance given in Table 3 with the Hellström’s profile will give a wrong normalized temperature profile. Comparing Table 2 and 3, we should remember that $R_b$ in the delta model corresponds to $R_{1/2}$ and $R_a$ to $R_{12}$ when the tube placement is symmetric. So, it is observed that the borehole resistance gives very similar final values but the short-circuit resistances show larger variations. It should be remember that during this test the interference was small. Any values of the short-circuit resistances will consequently have a small effect on the final results as long as they are large. Also, it should be remembered that the multi-pole evaluation is based on a symmetric configuration, which is not necessary the case in for real field tests.

**DISCUSSION AND CONCLUSIONS**

TRTs are becoming a mature technology for the evaluation of the subsurface thermal conductivity. However, some questions remain concerning the evaluation of the borehole resistance. In this work, preliminary work to find both, the borehole resistance and the internal resistance, using the bottom fluid temperature is presented. It was found that the resistances are dependent of the assumed temperature profile. It is important to note that the profile used should be compatible with the design algorithm to size the bore field when specifying the measured resistances. In this work, we used the profile suggested by Hellström (1991) and Beier (2011). The Beier’s method does not need the assumption of a uniform temperature profile. However it brings some conceptual questioning. Indeed, the network approach to represent a typical borehole is based on the quasi steady-state regime (steady-flux) where heat is exchanged between temperatures that vary with time, and where heat transfer can always be expressed by temperature differences divided by some thermal resistances. The thermal flow between both pipes in Beier’s network (Fig. 5) will depend on the soil resistances, which are time-dependent. However, this flow is small in a steady-flux regime and $R_{12}$ correspond to the internal resistance $R_a$ in the usual network approach. Both approaches used in this study gave satisfying results even though the internal resistance was different than expected theoretically. The main reason is that
the interference effect was small for the specific test analyzed. In that case, where $\eta$ (Eq. 6) is small, the solution becomes almost independent of $R_a$. In future work, field case with higher internal interference effect will be investigated.

ACKNOWLEDGMENTS

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REFERENCES


Thermal Response of helix ground heat exchangers

Marco Fossa  Benoit Stutz  Antonella Priarone  Antoine Coperey

ABSTRACT
This paper is devoted to the thermal analysis of shallow ground heat exchangers with pipes arranged in a helix configuration. The pipes where the carrier fluid is circulated typically embrace a cylindrical volume that is filled by ground or concrete, the latter being the case of the so called geopiles. Other pipes dispositions include conic helices that can be easily inserted in proper excavations. The analysis of the transient thermal behavior of a helix/ground assembly is here carried out according to different approaches, including the exploitation of superposition techniques, the finite element modelling and experiments in a reduced scale mock up. Different geometrical configurations have been taken into account and also the variability of ground and concrete thermal properties have been considered. A detailed description of the experimental set up is provided and the model results have been processed in order to develop suitable temperature response factors (or g-functions) to be employed for predicting the ground heat exchanger behavior in different operating conditions.

INTRODUCTION
Short vertical heat exchangers are often employed in Ground Coupled Heat Pump (GCHPs) applications since they usually do not need specific deep drilling equipment and can be buried on place during the operations of excavation in the building site. Such shallow heat exchangers (compared to deep borehole heat exchangers, BHE) can be either placed horizontally typically within 1 meter of soil or in trenches of various shapes down to some dozens of meters. Trench heat exchangers are often arranged as helix pipes (Helix Heat Exchanger, HHE) occupying a cylindrical volume in the ground that can be either filled with soil or concrete, hence realizing in the latter configuration a cylindrical geopile. When employing shallow heat exchangers, the external diameter of the whole assembly is comparable to the depth of the heat transfer surface. Simple models that describe linear (infinite line source ILS or finite line source FLS, Ingersoll et al., 1954, Eskilson 1987, Lamarche and Beauchamp 2007) and cylindrical infinite heat sources (ICS, Ingersoll et al., 1954) are not suitable for either describing the performance of the heat exchanger to constant and variable heat loads or to infer the ground properties in preliminary in situ tests devoted to soil conductivity evaluation.

In fact, the axial heat conduction contribution in the filled inner part of HHEs becomes not negligible, especially in the long term operations, and significantly affects the thermal response of the system. Moreover, the actual geometry of the HHE, i.e. the helix pitch, influences the temperature field especially close to the carrier fluid pipes. On this subject different studies have been developed during the last decades, either from the analytical or numerical point of view. One of the first pioneering studies is the numerical simplified model of the spiral heat exchanger by Rabin and Korin (1996) in which the Authors represented the behaviour of helix by means of a series of...
rings, with the same pitch distance. The effects of the thermal properties of the soil, the aspect ratio of the heat exchanger and its pitch distance have been analysed and compared with experimental data from field experiments.

A group of researchers has more recently developed several analytical solutions based on the Green’s function method, with increasing accuracy to represent the thermal response of HHEs into the ground (Man et al. 2010, Cui et al. 2011, Man et al. 2011). Exploiting the possibility of the superposition of basic solutions, they analysed either infinite or finite geometries, the seconds represented by the use of the virtual heat source theory. The starting point was the solution of the infinite and finite “solid” cylindrical model, where the inner space of the helix heat source was filled by ground. In these studies analytical solutions in terms of integrals to be solved have been presented for either ring or helix heat sources.

A few of recent studies combine the pure conduction into the soil surrounding a spiral coil with other mechanisms. Moch et al. (2014) analytically modelled and numerically solved the soil freezing problem around a helix coil, represented as an annular cylinder or a series of rings; the theoretical results have also been compared with experimental data with a satisfactory agreement. Go et al. (2015) analysed the effects of groundwater advection into the ground on the performance of a spiral coil field. Finally, Dehghan et al. (2016) developed fully 3D numerical simulations of spiral coils fields to analyse the effects of the different geometrical parameters including the distance between the heat exchangers, their aspect ratio, and the pitch distance between spires.

Based on the present literature survey it is possible to conclude that additional aspects of the problem deserve further investigation. These aspects include the development of laboratory scale facilities where to infer experimental data on heat exchanger behaviour in controlled conditions. Another issue is related to the development of proper temperature response factors (TRF or g-functions) for the helix source by applying the concepts first developed by Eskilson (1987). Finally the different thermal properties of ground and concrete media (in geopile configurations) have not been taken into consideration and dedicated models need to be developed to this special but not uncommon case.

In this paper the problem of developing temperature response factors for helix shaped ground heat exchangers is tackled according different strategies that include a semi analytical approach based on basic solution spatial superposition and the numerical solution of the 3D conduction equation with FEM codes.

Furthermore an experimental set up for assessing the heat exchanger performance is proposed and a detailed description of the laboratory scale set up for helix heat exchangers is provided.

**TEMPERATURE RESPONSE FACTORS WITH SPATIAL SUPERPOSITION**

The solution of the transient conduction equation is often the typical approach to describe the ground thermal response to a system of (or to a single) ground heat exchanger. One-dimensional (in the radial direction) and two-dimensional (radial and axial) analytical solutions have been proposed for predicting the ground response to a single constant heat pulse. These basic solutions (temperature response factors) can be superposed in time and space (Ingersoll et al. 1954, Eskilson 1987), for describing the transient response of a BHE borefield/ground assembly to any stepwise function describing the variable thermal load to the ground during the seasons (Yavuzturk and Spitler, 1999). Linear heat source solutions (ILS and FLS models) are based on the integration over a line of the Single Point Source solution (SPS, Ingersoll et al., Eq. 1):

\[
T(r, \tau) - T_{g,\infty} = \frac{Q}{4\pi k_{gr} r} \text{erfc} \left( \frac{1}{2\sqrt{Fo_r}} \right)
\]  

(1)

In the above expression, \( T \) is the ground temperature, \( r \) is the radial distance from the point source, \( k_{gr} \) is the ground thermal conductivity, \( T_{g,\infty} \) is the undisturbed (initial) ground temperature, \( Fo_r \) is the radius based Fourier number and finally \( Q \) is the heat transfer rate. By performing an integration along a line of length \( H \), the temperature
field at any radial and axial position around the line source (in infinite medium) can be evaluated as superposed contributions by the multiple point sources.

\[
T(r,z,\tau) - T_{gr,\infty} = -\frac{\dot{Q}}{4\pi k_{gr}} \int_0^h \frac{\text{erfc}\left(\frac{\sqrt{\alpha(z-h)^2 + r^2}}{2\sqrt{\alpha\tau}}\right)}{\sqrt{(z-h)^2 + r^2}} dh
\]  

(2)

Once the temperature field in time is known, a dimensionless temperature difference (g-function) can be obtained based on the concept of temperature response factor and with reference to a given boundary condition (heat rate BC or temperature BC, Priarone and Fossa 2016) and a reference radial distance. Being \( \bar{T} \) the average ground temperature at reference distance from (single or multiple) heat source, the temperature response factor can be evaluated as:

\[
g(Fo) = 2\pi k_{gr} \frac{\bar{T}(\tau) - T_{gr,\infty}}{\dot{Q}_H}
\]  

(3)

When referring to BHEs, the above reference distance is typically the borehole radius. In the present investigation the location at which the average temperature is evaluated is at pipe to solid interface, irrespective of the pipe arrangement in the solid medium.

The approach for calculating the TRF of HHEs is here based on a numerical calculation of the temperature field at radial distance \( r_p \) (Figure 1) from a helix curved line describing the heat exchanger coil. The line of individual sources has been discretized in terms of single SPSs and includes mirror sources for realizing the interface condition of constant temperature at ground top interface. A mesh sensitivity to g-function results have been performed with reference to the ILS and FLS reference solutions, as calculated for example in Fossa (2016).

FINITE ELEMENT ANALYSIS OF HELIX TYPE HEAT EXCHANGERS

A FEM model has been built to further develop the TRF concepts applied to conditions where the HHE inner volume has different thermal properties from the external surrounding soil. The model has been developed within Comsol Multyphysics environment. The model geometry is described in Figure 1. The helix coil is simplified by considering N circular rings with the same pitch distance \( \rho \), with imposed constant heat flux on their surface. Two domains are considered, both represented in axial symmetry coordinates: the inner (made by concrete in case of geopiles) with radius \( r_c \) and depth \( H_c \) and the external ground volume, with radius \( r_g \) and depth \( H_g \). The rings are tubular surfaces of \( r_p \) radius (pipe radius) and distance from the axis equal to \( r_p \). An additional dimension has been introduced for accounting for geopile covering thickness at pipe location. This extra thickness of concrete has been set equal to \( 5r_p \).

Assuming pure conduction in both the concrete and the ground domains, the transient equations to be solved are the Fourier ones, either in terms of ground (g) or concrete (c) properties.

\[
\frac{\partial T}{\partial \tau} = \frac{k}{\rho c} \nabla^2(T)
\]  

(4)

\( T \) is here the temperature, \( t \) is time, \( k \) is thermal conductivity, \( \rho \) is density and \( c \) the specific heat.

The boundary conditions are imposed heat flux on the ring surface (Eq. 5) and imposed temperature \( (T = T_{gr,\infty}) \) on the ground/concrete upper surface and on the external surfaces of the ground domain.
\[-k_c \left( \nabla T \cdot n \right) = \frac{\dot{Q}''_L}{2\pi \cdot r_p} = \frac{\dot{Q}''_H}{4\pi^2 \cdot r_p \cdot \frac{r_b}{p}} \]

\( \dot{Q}''_L \) represents the heat transfer rate for unit length of the spiral pipe, whereas \( \dot{Q}''_H \) the heat transfer rate for unit depth of the pile. By assuming that the distance of the first and the bottom rings from the end of the pile is \( p/2 \), the relationship between the two values is obtained by considering that

\[ \dot{Q}''_L = \frac{\dot{Q}}{2\pi \cdot r_b \cdot N} \quad \text{and} \quad \dot{Q}''_H = \frac{\dot{Q}}{2\pi \cdot \frac{r_b}{p}} \]

Continuity conditions hold at the interface concrete to ground, and the initial condition is \( T = T_{gr,\infty} \) on the whole computational domain. \( T_{gr,\infty} \) is the undisturbed ground temperature.

The selected dimensions of the ground volume are \( r_b = 200 \text{m} \) and \( H = 200 \text{m} \). These values assure an approximation of the ideal case of semi-infinite ground with an error of less than the 1% for the external boundary temperature with respect to the undisturbed ground temperature.

Different meshes have been tested, with increasing number of elements on the pipes surface and on the interface between the concrete and the ground, up to values that ensure the results in term of average temperature on pipe to not change more, with an accuracy of 0.2%. Finally, an unstructured triangular mesh is chosen, with 12 uniformly spaced elements on each pipe boundary and 450 elements on the interface boundary, for an overall number of elements equal to about 56800.

The average temperature on the rings surface has been evaluated by simulations as a function of time to infer the dimensionless thermal response factor:

**EXPERIMENTAL SET UP FOR REDUCED SCALE HELIX HEAT EXCHANGERS**

An experimental device has been designed and realised at Locie (Université Savoie Mont Blanc) to study the unsteady coupled heat and mass transfers (due to groundwater circulation and even freezing) occurring around a reduced scale HHE under controlled conditions. The experimental setup will be beneficial for assessing the limits of the modelling approach, which is based on thermal properties constant in time and no moisture or water effects. Temperature measurements using thermocouples and water content sensors are employed together with geophysical methods including Direct Current resistivity, Induced Polarization and Self-Potential measurements (Herma n et al 2014, Ikard and Revil 2014). The setup consists of a cylindrical tank (1.0 m diameter; 1.2 m height) filled with silica sand (grain size 180 µm) of known thermal properties (Figure 2). The side wall is thermally insulated and the tank bottom part is held at constant temperature thanks to a spiral heat exchanger (#6, Figure 2). A ventilation system controls air temperature and relative humidity on the upper surface of the sand. The presence of a water table is imposed in the set up by acting on a series of valves. Water percolation related to rainfalls is simulated by injecting water at the top of tank. 40 thermocouples (#14, Figure 2) are vertically distributed into the sand volume at three radial profiles (at 0, 5, 10 cm to tank axis) and three vertical positions. Distribution of water content is record by four sensors. Two X-shape profiles of 17 electrodes are set into sand and under the HHE, and four vertical profiles of 28 electrodes are located on tank walls for geophysical measurements in time domain. Set up dimensions have been chosen based on a dimensional analysis using the Buckingham-Pi theorem which reveals scaling laws and remarkable properties of this problem. When soil freezing is neglected, the dimensional analysis show that thermal perturbation of the soil can be characterized in terms of Biot and Fourier numbers together with proper dimensionless geometrical parameters, as defined in Figure 1 and where \( D=2n \).
\[ Bi = A \cdot (F_{0H})^a \cdot \left( \frac{H}{D} \right)^b \cdot \left( \frac{p}{D} \right)^c \cdot \left( \frac{p}{d_p} \right)^d \]  \hspace{1cm} (7) \]

Where Bi is based on thermal power extracted by HHE and the temperature difference between pipe periphery and undisturbed ground temperature.

In this experiment, the reducing scaling factor is 10 for the geometric parameters. The ground properties are conserved as well as the characteristic temperature differences of the problem. The capillary fringe is not scaled (sand grain and pore scales have been not changed). The conservation of the Biot and the Fourier number leads to a reduction by a factor 100 of the time scale (one real year is equal to 3.6 days), and a reduction by a factor 10 of the thermal power extracted by the exchanger. Due to geometrical constraints and based on previous considerations (Moch et al. 2014), it has been decided to scale only the shape parameter H/D.

Carrier fluid temperature is controlled in order to keep constant the pipe outer surface temperature.

RESULTS AND DISCUSSION

Two different models have been developed as described in previous chapters. Validation of both models has been carried out with reference to available literature solutions, namely the ICS and ILS solutions (Ingersoll et al.) and the FLS solution as proposed by Lamarche and Beauchamp.

The HHE geometry is defined in terms of quantities H=15m, r_s=0.3m, r_p=0.016m, p=0.1m.

Figure 3 shows the HHE/ground response in terms of T_{RF} (Eq. 3) for different values of discretization parameter zstep/r_p, where zstep is the distance between single point sources and r_p is the position at which the SPS g-function is evaluated. Abscissa is the H based Fourier number F_{0H}. As can be observed the SPS solution approaches the reference FLS one when the discretization parameter zstep/r_p is of the order of the unit and in all calculations has been set equal to 2, as a compromise choice between accuracy and calculation speed.
FEM model has been tuned in terms of mesh characteristics either with respect to reference analytical solutions (the ICS model, G solution) or with respect to the SPS superposition model itself. Figure 4 shows the FEM g-function solution for the HHE source when both the inner and outer parts of the helix surface have the same thermal properties.

Figure 3. Linear source discretization and sensitivity analysis. Comparison of reference FLS results with SPS superposition solutions. Parameter: SPS dimensionless distance zstep/rp.

Figure 4. FEM model validation with respect to the reference ICS solution. H=15m, rp=0.016m.

Figure 5. Comparison of SPS superposition and FEM model results in terms of HHE g-function.

Figure 6. Effects of different ground and concrete thermal properties on HHE g-function (FEM model).

Figure 7. FEM temperature field around a HHE with different thermal properties in outer (ground) and inner (concrete) volumes.

(a) 1 hour, ln(9FoH) = -9  |  (b) 1 day, ln(9FoH) = -6  |  (c) 20 days, ln(9FoH) = -3
properties (i.e. ground either inside or outside). The average ground temperature $\bar{T}$ is evaluated along the radius $r_b$. As it could be expected, in the early period the HHE source behaves like the ICS since the mutual influence of neighbour coils and even the ground top interface does not yet play its role.

Figure 5 is the comparison between the SPS and FEM models for the same HHE geometry and medium thermal conductivity ($k=k_{gr}=1.5$ W/mK). It can be noticed that the simple 1D solution by SPS superposition is able to describe the HHE response in very good agreement with reference to the FEM solution and hence seems suitable for example for parametric analyses aimed at HHE geometry optimization. Figure 6 refers to FEM different simulations where at given HHE geometry the inner volume of coils has been changed in terms of heat conductivity ($k_c=2.0$ W/mK, $k_{gr}=1.5$ W/mK) while keeping constant both media density and specific heat. The curve named “soil” refers to homogeneous medium ($k=k_{gr}$) inside and outside the helix coil while the curve “soil and concrete” is related to a geometry with a concrete inner core. Being the thermal conductivity values quite similar, the two g-function profiles are close each other and almost superpose in the early period. Later in time and toward the asymptotic region the axial conduction along the pile starts to increase the overall heat transfer thanks to a better conductivity of the inner core. To this aim Figures 7 show snapshots of the temperature field at three different instants from heat generation start. In particular image 7a refers to the early period ($\ln(9FoH)=-9$) when the heat streamlines are oriented radially and the ground medium offers a convenient heat path. Figures 7b and 7c ($\ln(9FoH)$ equal to -6 and -3 respectively) show on the other hand the heat flow through the pipe core, which can take advantage from pile core higher conductivity.

CONCLUSIONS

In this paper a joint research has been carried out for assessing the performance of trench heat exchangers for geothermal heat pump applications. In particular the helix geometry has been taken into consideration either from an experimental or theoretical point of view. An experimental setup has been conceived and realized at Locie lab in France and data collection is expected to start in 2017, thus allowing a complete comparison between model predictions and real condition HHE thermal response, also in terms of presence of groundwater circulation.

Modeling has been addressed to pure conduction conditions and 1D and 2D models have been proposed. In particular the present 1D model is based on a simple approach of solution superposition starting from the single point source (SPS) while FEM calculations have been performed for taking into account the possibility that the inner volume of helix heat exchanger could be filled by concrete, thus realizing a geopile heat exchanger. Validation of both models with respect to literature solutions showed that the SPS is able to fully describe the geometry effects of HHE parameters and it is hence useful for fast geometry optimization analyses. The FEM model on the other hand is able to cope with different thermal properties inside and outside the helix pipe surface. Cross comparison between models provides first information on the reliability of homogeneous property hypothesis and SPS model application. The temperature response factors (here calculated according to different levels of detail) are considered helpful for applying temporal superposition techniques devoted to variable heat rate simulations of concrete geopiles and helix heat exchangers.

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NOMENCLATURE

\[ c = \text{specific heat} \quad \text{(J/kg K)} \]
\[ F_{op} = \text{Fourier number}, F = \left( \frac{\alpha \tau}{r^2} \right) \]
\[ G = \text{ICS temperature response factor} \quad \text{(\textit{-})} \]
\[ H = \text{HHE depth} \quad \text{(m)} \]
\[ k = \text{Thermal conductivity} \quad \text{(W/mK)} \]
\[ \dot{Q} = \text{Heat transfer rate} \quad \text{(W)} \]
\[ \dot{Q}' = \text{Heat transfer rate per unit length} \quad \text{(W/m)} \]
\[ p = \text{coil pitch} \quad \text{(m)} \]
\[ r = \text{radius} \quad \text{(m)} \]
\[ T = \text{Temperature} \quad \text{(^\circ C or K)} \]
\[ \alpha = \text{Thermal diffusivity} \quad \text{(m^2/s)} \]
\[ \rho = \text{density} \quad \text{(kg/m^3)} \]
\[ \tau = \text{Time} \quad \text{(s)} \]

Subscripts

\[ \infty = \text{undisturbed conditions} \]
\[ b = \text{HHE radius} \]
\[ c = \text{concrete} \]
\[ gr = \text{ground} \]
\[ p = \text{pipe} \]

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Heat Extraction Distributed Thermal Response Test: a methodological approach and in-situ experiment

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ABSTRACT

The Thermal Response Test (TRT) is a worldwide adopted in-situ methodology able to estimate the ground thermal conductivity and borehole thermal resistance. During the test the carrier-fluid exchanges a constant heat flux with the ground while circulating in a pilot Borehole Heat Exchanger (BHE). During a Distributed Thermal Response Test (DTRT) the ground thermal conductivity and borehole thermal resistance are determined at different vertical sections along the borehole.

The measured fluid temperature values are analyzed with numerical or analytical approaches based on mathematical models which typically approximate the BHE. Those models are based on some strict assumptions, including pure conduction and constant heat transfer rate.

During a heat extraction TRT the operating conditions to the ground are similar to the “winter mode” conditions of a working BHE system. In such case the estimated thermal behaviour of the borehole can differ from the result obtained by means of a heat injection TRT. This issue is of peculiar interest for water-filled boreholes, where the BHE thermal resistance is related to the water temperature and density gradient in the borehole filling-space. In this operating mode a heat pump is usually employed and the constant heat transfer rate condition required by the models can be difficult to be respected since the efficiency of the cooling-machine is dependent on the inlet carrier-fluid temperature to the evaporator.

In this paper a methodology to perform a heat extraction DTRT with constant heat transfer rate to the ground is presented. The approach described has been applied in a real water-filled borehole installed in Stockholm, Sweden. Data analysis results are presented and the outcomes regarding the evaluation of the local borehole thermal resistance are discussed and compared with those from an earlier heat injection test performed in the same borehole.

INTRODUCTION

Over 1.7 million Heat Pump units have been sold in the European Union in 2014, for an estimated energy production of about 13 TWh. In Sweden, where the largest share for ground source units have been sold, the employment of ground-water filled boreholes is the most common solution (Eurobserv’er 2015).

An accurate borehole system design involves the challenge of properly estimating the thermal properties of both ground and heat exchangers. The undisturbed ground temperature ($T_{\text{gr,} \infty}$), the ground thermal conductivity ($k_{\text{gr}}$) and the borehole thermal resistance ($R_b$) need to be measured or estimated with sufficient accuracy, since they have a direct impact on the efficiency and operation costs of heat pump systems (Marcotte and Pasquier 2008). The total number of heat exchangers and the overall borehole length strongly affect the general system cost. Hence, the correct sizing of the borefield, through a preliminary investigation on the ground properties, is a compulsory task in order to obtain shorter payback periods.

The thermal response test is well known technique for evaluating the ground conductivity and the borehole thermal resistance. Since modern TRT apparatus started to be developed (Austin 1998, Eklöf and Gehlin 1996), the TRT experimental method has been applied successfully with different approaches and settings (Spitler and Gehlin...
The most widely used analysis method is based on the response of an Infinite Line Source model (ILS) (Carslaw and Jaeger 1945, Ingersoll and Plass 1948, de Vries 1952, Mogensen 1983) or an Infinite Cylindrical Source model (ICS), since those are the easiest to be implemented (Gehlin et al. 2003).

The test methodology has a number of important limitations given mainly by the analysis method assumptions and by practical experimental issues. The thermal energy rate is assumed to be constant during the test and the uncontrolled variation of the heat flux to the borehole circulating fluid affects the results. Beier and Smith (2003) propose a deconvolution algorithm to remove the variable heat rate effect from test. An analysis procedure based on the superposition principle of the basic analytical model solution can also be employed when measured data are affected by variable heat rate. On the other hand, modern apparatus, as described for example in Witte et al. (2002) and Rolando (2015), allow to maintain the heat transfer constant during the test by employing a feedback control system. The ground thermal properties are assumed to be homogenous even if usually the geological profile along the borehole heat exchanger includes layers with different characteristics. The undisturbed ground temperature is also considered to be completely homogeneous and the real temperature is not considered. Numerical simulations and analysis have demonstrated that the undisturbed ground temperature gradient can affect the estimation of the ground thermal properties (Signorelli 2004). Fujii et al. (2006) defined a new methodological approach consisting in the measurement of the temperature along the borehole by means of optical fiber cables installed on the external wall of the pipes. This methodology is nowadays known as named Distributed Thermal Response Test (DTRT). Following the same approach, Acuña et al. (2009) introduced the possibility to evaluate the complete borehole thermal resistance by locating the optical fiber inside the pipes. Witte (2005) presented a test protocol based on the application of several energy pulses including heat injection and heat extraction. Witte (2013) provides and extensive discussion on the uncertainty analysis related to TRT parameter estimation. A similar test protocol has also been tested in groundwater-filled BHEs yielding a good result in detecting thermally induced convective flow in the borehole water (Gustafsson 2011).

Despite heat injection is the most common test mode for TRT, heat extraction tests allow to investigate the ground and borehole thermal properties in temperature ranges comparable to the actual temperature range of heat pump systems used for space heating (Witte and van Gelder 2006). In ground-water filled heat exchangers the convection around the borehole affects the results and the borehole thermal resistance will vary more than heat injection tests. Water has the highest density around 4°C and the convective flow is affected by the ground water-fill temperature across this value. If the temperature in the groundwater falls below zero latent heat is also released during the phase change and the heat transfer properties change as ice is formed. Due to the complexity of the heat transfer phenomena, with the available TRT analysis methods involving line source and parameter estimation based on a conductive heat transfer model, the adoption of one \( R_b \) value for heat extraction TRT in groundwater-filled BHEs is not possible (Gustafsson 2011).

As well known, the main issue related to performing a Heat Extraction TRT (HETRT) is due to the chiller (heat pump) performance dependence on the evaporation temperature. This implies that the heat transfer rate extracted from the ground cannot be constant without the employment of a dedicated control. In this paper the description of the experimental methodology adopted for performing a heat extraction test maintaining the heat transfer rate constant is presented including the results from a field experiment where the temperatures along the borehole have been measured allowing the estimation of the borehole thermal resistance at a local level. The estimated values are presented and discussed focusing on the comparison of the results to the values obtained in the same borehole during a heating TRT.
In Northern Europe countries like Sweden and Norway, the filling material commonly used in Borehole Heat Exchangers is the ground water. In such systems, the effect of the convection induced by the temperature and density gradients of the ground surrounding the borehole may not be neglectable. During a heat injection TRT the thermal behaviour of a BHE, usually described by the effective thermal resistance (Mogensen 1983, Javed and Spitler 2016), can be very different when compared to a heat extraction TRT. Furthermore, in northern countries a BHE system is mainly used in heat extraction mode. For these reasons, a Heat Extraction TRT (HETRT) allows to better characterize a BHE in an operating mode closer to the real working conditions of a ground source heat pump system.

A field experiment has been carried out aiming to perform a HETRT maintaining constant the heat extracted from the ground. The BHE has been described in Section 3.2.1 of the Acuña’s (2013) Doctoral Thesis. It consists in 6 ground-water filled boreholes positioned in an L shaped arrangement. Two of these BHEs are U-pipe polyethylene heat exchangers and have been installed together with an optical fiber cable both inside and outside the pipes. The optical fiber cable is installed along the borehole pipes and extended also to the ground-water between the pipes and the borehole wall. By means of a Distributed Temperature Sensing (DTS) device, laser light pulses are sent through the optical fiber. The signal is refracted back from each section of the fiber cable and is sent back to the laser instrument with a frequency scattering (Raman scattering), that depends on the local temperature (Acuña 2013). The depth of the borehole used for this experiment is 260m. Worth noticing, the BHE used in the field experiment described in the following is part of a working system and it was rested for more than two months before the test has been carried out.

Figure 1a shows the layout of the field experiment. The main components of the test rig are: a water to water heat pump unit, a buffer tank, an auxiliary plate heat exchanger, an electronic three-way valve, a PID controller and three circulation pumps. The main idea was to control and maintain constant the borehole inlet and outlet temperature difference in order to keep constant the extracted heat transfer rate. The flow rate into the borehole was assumed to be constant and it has been verified during the data analysis.

The condenser inlet of the heat pump has been connected to the tap water while the outlet was connected to the drain. The evaporator has been connected to the buffer tank through a dedicated piping loop including a circulation pump. An auxiliary piping loop has been provided to connect the buffer tank to an auxiliary plate heat exchanger.
this loop, an electronic three-way valve was set to modulate the flow rate in one side of the plate heat exchanger. Finally, the other side of the auxiliary heat exchanger has been connected to the borehole.

A Proportional Integrative (PI) controller has been tuned in order to control the opening of the three-way valve and maintain the difference between the inlet and outlet borehole temperatures constant. The set point temperature of the storage tank was - 10°C. The heat pump employed was a variable capacity unit and the compressor speed was controlled by means of an inverter. A computer code has been developed to control the heat pump compressor speed through the inverter during the test. A web interface has also been developed to monitor and control the experiment remotely.

Figure 1b shows the complete equipment assembled in the field. All the pipes and heat exchangers have been properly insulated in order to minimize the heat loss during the experiment. A water and ethanol solution has been properly prepared in order to cope with the expected evaporator working temperature. Since the expected evaporation temperature was between -15 and -18 °C (depending on the compressor speed), the buffer tank was filled with an aqueous solution of ethanol (about 35% by weight) in order to have a freezing temperature of about -22°C. The ethanol concentration in the borehole loop was about 16% by weight and it has not been modified.

Data analysis

The optical fiber measurements have been collected by a DTS device having a space resolution of 2m. As explained in the following section, before the constant extraction rate methodology presented above could be employed, the measurement campaign included a number of heat extraction and recovery cycles with different heat extraction rates. In order to analyse the data considering the variable heat rate, the superposition principle has been applied to the solution of the ILS model for each heat pulse considered. For each section, the borehole wall temperature $T_b$ has been calculated through the expression given by Eq. 1.

$$T_b = T_{gr,\infty} + \sum_{i=1}^{n} \left( \frac{\dot{Q}_i - \dot{Q}_{i-1}}{\pi k_{gr}} \right) \frac{E_i(Fo(t_n-t_i))}{\tau}$$

Where $T_{gr,\infty}$ is the undisturbed ground temperature, $\dot{Q}'$ is the heat transfer rate per unit length, $\dot{Q}_i - \dot{Q}_{i-1}$ represents the heat transfer rate pertaining to the $i^{th}$ heat pulse and related to a given section, $n$ is the number of heat pulses considered, $k_{gr}$ is the ground thermal conductivity, $Fo$ is the Fourier number based on the borehole radius and $\tau$ is the time. $E_i$ is the Infinite Line Source model response factor (Ingersoll 1948). The approximation suggested by Abramowitz and Stegun (1964) has been adopted to evaluate the $E_i$ values without involving numerical integration, as it is shown in Eq.2.

$$E_i = -\gamma - c_1 \ln(X) - \sum_{j=2}^{i} c_j X^j$$

Where $\gamma \approx 0.5772$ is the Euler constant $X = 1/(4Fo)$ and $c_j$ are constant coefficients: $c_1 = 0.99999193$, $c_2 = -0.24999055$, $c_3 = 0.05519968$, $c_4 = -0.00976004$, $c_5 = 0.00107857$.

For each section, the local borehole thermal resistance has been calculated at each time step by means of Eq.3.

$$R_b = \frac{T_f - T_b}{\dot{Q}_r}$$

Where $T_f$ is the average temperature of the fluid circulating in the section.

For a generic section between the depths $A$ and $B$, the heat transfer rate is calculated at each time step considering the inlet and outlet fluid temperatures of both downward and upward pipes, as it is shown in Eq.4.

$$\dot{Q} = \dot{m} c \left( (T_{down,B} - T_{down,A}) + (T_{up,A} - T_{up,B}) \right)$$

Where, $\dot{m}$ is the fluid mass flow rate, $T_{down}$ and $T_{up}$ are the fluid temperatures measured in the downward and
upward pipe, respectively. The fluid specific heat, $c$, has been assumed constant and equal to 4.372 kJ/(kg K).

**RESULTS**

An overall test of 440h has been carried out and Figure 2 shows the average fluid temperature evolution related to the 12 sections considered along the borehole. For each section the temperatures are measured inside both downward and upward pipes and the average of all the values is calculated at each time.

![Figure 2](image1.png)

**Figure 2**   Optical fiber temperature measurements: BHE section temperatures over the entire 450h test.

![Figure 3](image2.png)

**Figure 3**   Constant heat transfer rate results from the employment of the presented experimental methodology.
The sample time of the measurements was 5 minutes. Several heat extraction and recovery cycles have been carried out before the methodology described in the previous section could actually be applied in order to perform a final test maintaining constant the heat transfer rate. From the inspection of the figure, during a first stage of approximately 48h the undisturbed ground temperature has been recorded for every sections. After that, for a period of about 70h the fluid has been circulated in the borehole. A series of interrupted heat extraction tests can be observed between 120h and 190h, due to the failure of the TRT control system to which the test rig had been connected to. At 190h a temperature drop has been caused by the temporary change of the TRT layout that has been required to solve the control system issue. In the time windows between 285h and 440h, the constant heat transfer rate DTRT has been finally carried out.

Figure 3 shows the borehole inlet and outlet temperature evolution during the final test and the evolution of the calculated heat transfer rate. As can be observed, the active control technique presented in this paper proved to be able to maintain constant the heat extracted from the ground with a good stability.

**Comparison to heating DTRT**

Based on the last 150 hours of the test presented above, local borehole resistance values have been calculated. The same equipment and a similar methodology as used in Acuña et al. (2009) has been followed, i.e. the same borehole, the same borehole heat exchanger, the circulating fluid and same measurement equipment. The local rock thermal conductivity values found during heat injection tests have also been adopted for the present data analysis. The thermal conductivity range is between 2.60 and 3.62 W/mK in 12 different borehole sections, with an average of 3.10 W/mK. The results are shown in Figure 4.

![Figure 4](image)

**Figure 4** Comparison of measured local borehole resistances during heat injection and extraction DTRTs

The volumetric flow rates were different in the two tests, i.e. 0.36 l/s and 0.53 l/s in the heat extraction and
injection test, respectively. Due to difference in flow and fluid temperature, a difference in the fluid to pipe thermal resistance of about 0.0087 mK/W (0.016 during cooling and 0.0073 mK/W during heating) is expected during these two tests. Heat extraction and injection tests were carried out at temperatures of about 5 and 15°C, respectively.

The average local borehole resistance obtained during heat extraction was 0.102 mK/W and ranged from 0.067 to 0.172 mK/W. The lowest value, in section 9, is presumably affected by a local borehole anomaly. It is known from Acuña (2013) that a borehole anomaly exists in the vicinity of 190m depth, which may be the cause of the differences observed. Discontinuities in this section of the borehole have also been noticed during other tests in this borehole. The description and discussion related to this observed anomaly can be found in Section 3.3 (page 42) of Acuña’s (2013) Doctoral Thesis as well as in Section 2.1 of Acuña’s (2010) Licentiate thesis.

During the heat injection test, the average local borehole resistance was equal to 0.063 mK/W and ranged between 0.054 to 0.078 mK/W.

The average difference between the heat extraction and the heat injection test is 0.039 mK/W.

While observing differences between these two tests, it is important to keep in mind that the error of the measurements from the heat injection test range between 13% and 17%, as described in Acuña (2013). Since the same thermal conductivity values were used as an input to the calculations in the heat extraction test, a similar error level is expected.

Another aspect to be kept in mind while comparing these values is the extracted cooling power of about 3.6kW, compared to 9 kW while injecting power. Given the low power used in the heat extraction test, small radial temperature gradients between the pipes and the borehole wall can be expected in spite of the variation in borehole resistance.

Local borehole resistances resulted to be larger during heat extraction periods. Although this was expected given the lower temperature levels and the employment of a water-filled borehole, it can be of particular interest to go further and observe tendencies along the borehole depth. The difference in borehole resistance influenced by the groundwater temperatures is somewhat already forecasted. Current ongoing work is being dedicated to the temperature dependency of the local resistance values, as referred to groundwater temperatures inside the boreholes. This investigation will be put in perspective with earlier work carried out in Gustafsson (2010) and Spitler et al. (2016), among others.

CONCLUSIONS AND FUTURE WORKS

In this paper a methodology to perform a heat extraction DTRT maintaining constant the extraction rate has been presented and a field experimental set up has been described. The methodology proved to allow to maintain a constant heat rate extracted from the ground with a good stability.

The field measurements have been used to estimate the local values of the borehole thermal resistance in 12 sections considered along the borehole. The results have been compared and discussed with those from a previous heat injection test. Assuming that the local ground thermal conductivities are the same as found during the earlier heat injection test, the local borehole thermal resistances during the heat extraction DTRT are about 40% higher compared to the heat injection DTRT.

Future work is addressed to further analysis of the measurements including the investigation of the borehole thermal resistance values versus the water-fill temperature along the borehole.

NOMENCLATURE

- $F_o$ = Fourier number, $F_o = (\alpha \tau)/r_b^2$
- $k$ = Thermal conductivity (W/mK)
- $\dot{Q}$ = Heat transfer rate (W)
- $\dot{Q}'$ = Heat transfer rate per unit length (W/m)
- $n_b$ = Borehole radius (m)
\[ R_b = \text{Borehole Thermal resistance (mK/W)} \]
\[ T = \text{Temperature (°C)} \]
\[ \alpha = \text{Thermal diffusivity (m}^2\text{/s)} \]
\[ \tau = \text{Time (s)} \]

**Subscripts**

- \( f \) = carrier fluid
- \( b \) = borehole
- \( gr \) = ground
- \( \infty \) = undisturbed conditions

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Transient thermal resistance of borehole heat exchangers for hourly simulations of geothermal heat pumps systems

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ABSTRACT
The correct design of borehole fields requires the correct evaluation of the transient ground thermal response in time, but also the accurate estimation of the borehole (BHE) thermal resistance, especially the grout contribution. Generally, the borehole thermal resistance is considered as steady-state; however, when considering the borefield hourly response to the building variable thermal loads, also the transient behavior of the grout thermal resistance plays an important role, which is quite often neglected. This study analyzes, with a dimensionless approach, the transient grout thermal resistance, with particular attention devoted to the effect of the boundary condition imposed to the internal tubes, namely imposed heat flux, imposed temperature and imposed convective coefficient, the last being the real operating conditions. In addition, the effects of grout to ground thermophysical properties and of shank spacing are analysed. The steady state numerical results are also compared with literature correlations. Finally, numerical evidences are given to demonstrate that the usual approach of calculating the overall BHE resistance just summing the grout resistance, numerical obtained by imposing a temperature on the tube surface, to the convective one can lead to meaningful errors at low Biot numbers.

INTRODUCTION

Ground Coupled Heat Pumps (GCHPs) are a high efficiency solution for building conditioning in the framework of energy saving and environmental protection. In most applications the heat pump is coupled with the ground by means of Borehole Heat Exchangers (BHEs) and it takes advantage of the favourable ground temperatures for high efficiency heating and in cooling heat transfers. The most complete way for analysing the thermal behaviour of the GCHP system as a whole is an hourly approach, able to forecast and optimize the system response to variable thermal loads.

This approach is quite different from the methods commonly used to size and examine the operating behaviour of the BHE field, since the typical approach is to consider the ground response to simplified heat load profiles, typically with a monthly time step. In this framework, it is a common practice to use a two resistances scheme to model the thermal behaviour of a BHE field. In particular, the first resistance depicts the thermal response of the ground, intrinsically variable in time, and is commonly described by proper Temperature Response Factors (TRF), also known as g-functions. On the contrary, the second resistance represents the BHE thermal behaviour (Yavuzturk and Spitler 1999; Zeng, et al. 2003; Marcotte and Pasquier 2008) and is frequently considered constant in time. Lamarche et al. (2010) presented a wide review of methods and correlations to evaluate the borehole thermal resistance, focusing in particular on the grout thermal resistance. More recently Javed and Spitler (2016) compared a
The aim of this work is to analyze the transient behavior of the grout thermal resistance by means of numerical simulations performed in COMSOL Multiphysics environment. The modelled domain is a 2D cross section of the BHE and the modelling is performed by introducing proper dimensionless quantities. Different aspects are considered, including the effects of the thermal conductivity and heat capacity values, in terms of grout to ground ratio: the dimensionless grout thermal resistance appears to be a function of both these ratios but reaches a constant steady-state value for specific threshold values of the radius based Fourier number.

A special attention has been devoted to the investigation of the influence of the boundary condition applied on the pipes side of the BHE: imposed heat flux, imposed temperature and imposed convective coefficient. For a U-pipe BHE, due to its non-axiallysymmetrical geometry, the choice of the proper boundary conditions is revealed to be important, because it significantly influences not only the transient trend of the BHE thermal resistance but also its steady state value.

THEORETICAL BACKGROUND

The borehole thermal resistance includes different contributions, namely the convective resistance of the fluid, the conductive resistance of the pipes and the conductive resistance of the grout. The first two resistances can be easily calculated, whereas the evaluation of the third requests analytical correlations or numerical simulations (Figure 1). In this paper the analysis is devoted exclusively to the grout thermal resistance which is typically the main contribution; the usual approach to obtain the effective borehole thermal resistance is to add the convective and pipe contributions (Eq. 1 and 2, single U case), even if this can result in meaningful errors, as discussed later in this paper.

\[
R_b = R_{fp} + R_{gt}/2
\]

\[
R_{ftp} = \frac{1}{2\pi} \left( \ln\left(\frac{r_{p,\text{out}}}{r_{p,\text{in}}}\right) + \frac{1}{\frac{r_{p,\text{in}} \cdot h}{k_p}} \right)
\]

In the following, the main correlations to evaluate the grout thermal resistance are presented. Considering the borehole radius \( r_b \), the external pipes radius \( r_p \), the half shank spacing \( d \), the ground volume radius \( r_g \), and the grout and ground thermal conductivities \( k_{gt} \) and \( k_g \), respectively, the following dimensionless variables are introduced:

\[
d^* = \frac{d}{r_b}, \quad r_p^* = \frac{r_p}{r_b}, \quad r_g^* = \frac{r_g}{r_b}
\]
Paul (1996) proposed a very simple correlation, in which the dimensionless grout thermal resistance $R_{gt}^*$ depends on the ratio of pipe to borehole radius $r_p^*$, and on the half shank spacing between the pipes $d$ ($\beta_0$ and $\beta_1$ are constants, whose values depend on the distance $d$, see Figure 2):

$$R_{gt}^* = \frac{2\pi \cdot k_{gt}^* \cdot R_{gt}}{\beta_0 (1/r_p^*)^{\beta_1}}$$ (6)

Hellstrom (1991) proposed the so called ‘line source formula’, depending also on the ratio of grout to ground thermal conductivities $k^*$, by means of the parameter $\sigma$:

$$R_{gt}^* = \frac{1}{2} \left[ \ln \left( \frac{1}{r_p^*} \right) + \ln \left( \frac{1}{2 \cdot d^*} \right) + \sigma \cdot \ln \left( \frac{1}{(1/d^*)^4 - 1} \right) \right]$$ (7)

Bennet et al. (1987) proposed a complex algorithm, called ‘multipole method’ for calculating the overall thermal resistance $R_p$. Based on Eq. 1 formulation, the Bennet expression (first-order approximation) can be rearranged as:

$$R_{gt}^* = \frac{1}{2} \left[ \ln \left( \frac{1}{r_p^*} \right) + \ln \left( \frac{1}{2 \cdot d^*} \right) + \sigma \cdot \ln \left( \frac{1}{(1/d^*)^4 - 1} \right) \right] - \frac{r_p^{*2}}{4d^{*2}} \left[ \frac{1}{1 - R_{fp}^*} - \left( \frac{4\sigma}{(1/d^*)^4 - 1} \right)^2 \right] \left[ 1 + \frac{r_p^{*2}}{4d^{*2}} \right]$$ (8)

Neglecting the pipe contribution, the dimensionless convective resistance can be expressed as:

$$R_{fp}^* = 2\pi \cdot k_{gt}^* \cdot R_{fp} \simeq 2\pi \cdot k_{gt}^* \cdot \frac{1}{2\pi \cdot r_p^* \cdot h} = \frac{1}{Bi \cdot r_p^*}$$ (9)

where the Biot number is here defined as $Bi = h \cdot r_p^* / k_{gt}^*$.

Finally, Sharqawy et al. (2009) more recently suggested the following simple correlation:

$$R_{gt}^* = \left[ -1.49 \cdot d^* + 0.656 \cdot \ln \left( \frac{1}{r_p^*} \right) + 0.436 \right]$$ (10)

---

**Figure 2** Present paper test cases for single U pipe heat exchanger.
These correlations are based on the assumption that the borehole thermal resistance can be determined for quasi-steady state conditions and depends mainly on geometrical parameters and on grout and ground thermal conductivities. On the contrary, the first order approximation of the Bennet et al. correlation takes into account, for the evaluation of the grout thermal resistance, also the convective resistance contribution that influences the temperature field in the grout domain.

GOVERNING EQUATIONS AND NUMERICAL MODEL

In this paper single U-pipe BHE is considered, with $r_b = 7.62$ cm and $r_p = 2.1$ cm ($r_p^* = 0.275$) (based on commercially available geothermal probes). Four different shank spacing values are considered, namely $d^* = 0.289; 0.367; 0.5; 0.695$; the first value leads to a configuration like case A of Figure 1, the second and third values to a configuration like case B, and the last one to a configuration like case C.

The model analyzes a 2D cross section of the BHE and the considered domains are the grout and an appropriate surrounding portion of ground. The grout thermal resistance is assumed to be time dependent; thus, the differential equations in the grout and the ground are, respectively:

$$\frac{\partial T}{\partial \tau} = \frac{k_g}{(\rho c)_g} \nabla^2 T$$ (11)

$$\frac{\partial T}{\partial \tau} = \frac{k_g}{(\rho c)_g} \nabla^2 T$$ (12)

Three different boundary conditions have been imposed at the surface between pipes and grout, namely imposed heat flux per unit length $\dot{Q}'$, imposed temperature $T_p$ and convective heat transfer with imposed fluid temperature $T_f$ and convective coefficient $h$:

$$-k_g (\nabla T \cdot n) = \frac{\dot{Q}'}{4\pi \cdot r_p}$$ (13a)

$$T = T_p$$ (13b)

$$-k_g (\nabla T \cdot n) = h(T - T_f)$$ (13c)

The external boundary of the computational domain, which is a circle with radius $r_g$, is considered isothermal

$$T = T_{gr,\infty}$$ (14)

where $T_{gr,\infty}$ is the undisturbed ground temperature.

Continuity conditions hold at the interface grout to ground, and the initial condition is $T = T_{gr,\infty}$ on the whole computational domain.

The study and the equation solution are developed in a dimensionless form, introducing different dimensionless variables, depending on the boundary conditions. Thus, the following dimensionless time $\tau^*$ and temperatures $T^*$ are defined:

$$\tau^* = \frac{k_g \cdot \tau}{\left(\rho c\right)_g r_p^2} = Fo_{gb}$$ (15)

Imposed heat flux,

$$T^* = \frac{T - T_{gr,\infty}}{\dot{Q}'}$$ (16a)

Imposed temperature,

$$T^* = \frac{T - T_{gr,\infty}}{T_p - T_{gr,\infty}}$$ (16b)
Imposed convection,
\[ T^* = \frac{T - T_{gr,sc}}{T_f - T_{gr,sc}} \]  

(16c)

By introducing also the dimensionless quantity:
\[ (\rho c)_T^* = \frac{(\rho c)_g}{(\rho c)_f} \]  

(17)

it is possible to rewrite Eqs. (11), (12), (14) in the following dimensionless form:
\[ \frac{\partial T^*}{\partial \tau^*} = \frac{k^*}{(\rho c)^*} \nabla^2 T^* \]  

(18)

\[ \frac{\partial T^*}{\partial \tau^*} = \nabla^2 T^* \]  

(19)

\[ T^* = 0 \]  

(20)

The dimensionless initial condition is \( T^* = 0 \) everywhere in the domain.

The different boundary conditions become:
\[ -(\nabla T^* \cdot \mathbf{n})|_{y} = \frac{1}{4\pi k^* r_p} \]  

(21a)

\[ T^* = 1 \]  

(21b)

\[ -(\nabla T^* \cdot \mathbf{n})|_{y} = Bi \cdot (T^* - 1) \]  

(21c)

The analysis of the effects of the convective coefficient is carried out for \( Bi \) values equal to 10 and 50.

In order evaluate the effect of the thermophysical properties of grout and ground, different combinations of the following values have been considered: \( k^* = 0.3; 0.6; 1.0; 1.667; \) \( (\rho c)^* = 0.4; 0.7; 1.0 \).

The dimensionless Eqs. (18)-(20), with the continuity conditions at the interface grout-ground, the initial condition and the boundary conditions (21) have been solved by means of COMSOL Multiphysics.

A dimensionless time interval \( 10^{-4} \leq \tau^* \leq 10^5 \) is considered, which is divided into 4500 uniform time steps (each dimensionless time interval is equal to 0.002).

A circular computational domain representing the ground volume is employed, with dimensionless radius \( r_g^* = 1000 \); an extensive check of the adequacy of this size of the computational domain has been already performed by Zanchini and Lazzari (2014), to which the present work refers.

For the analysis of the mesh suitability, the criteria here adopted follows the work by Priarone and Lazzari (2014), even if in the present investigation the elements number is further increased for a more accurate calculation of temperatures and heat fluxes on pipes and BHE boundaries. Finally, an unstructured triangular mesh is chosen, which presents 200 uniformly spaced elements on each pipe boundary and 400 elements on BHE boundary, for an overall number of elements equal to about 24000.

RESULTS AND DISCUSSION

The transient behaviour of the dimensionless grout thermal resistance \( R_{gt}^* \) is presented in Figures 3 and 4 for an intermediate value of the shank spacing \( d^* = 0.5 \).

In particular, Figure 3 allows the effect of the boundary conditions to be appreciated: the trends are different for both the transient zone and the steady state values, with a different \( F_{th}^* \) value at which they reach the asymptote. The grout thermal resistance for imposed heat flux is lower for the small values of \( F_{th}^* \) whereas it reaches a considerably higher value in the steady state zone. The trends of \( R_{gt}^* \) for the imposed temperature and convective boundary conditions seem to have a comparable behaviour, more similar for higher value of Biot number: in fact, for \( Bi = 50 \) (corresponding approximately to a convective coefficient \( h = 2000 \text{ W/m}^2\text{K} \)) the two curves nearly overlap.
On the contrary, Figure 4 shows the effect of the dimensionless heat capacity for unit volume \((\rho c)_*\) on the transient behaviour of the dimensionless grout thermal resistance \(R_{gt}^*\): increasing the \((\rho c)_*\) value the grout thermal resistance is decreased for all the analysed boundary conditions here taken into account. Conversely, the dimensionless heat capacity \((\rho c)_*\) does not affect the steady value of the dimensionless grout thermal resistance \(R_{gt}^*\), as expected from the steady formulation of the conduction equations.

The steady values of \(R_{gt}^*\) calculated by means of correlations (7, 8, 10) are reported in both Figures 3 and 4: Paul correlation provides a value of 0.95, much higher than all the numerical results and it is outside of the selected axis range. A very good agreement exists between numerical results and Bennet et al. estimations (Eq. 8) when in present simulations the convection boundary condition is applied. The value of \(R_{gt}^*\) from Hellstrom correlation is intermediate between the imposed temperature and imposed heat flux numerical values, whereas the Sharqawy value is considerably lower than all the numerical results.

Figure 5 compares the steady state values of \(R_{gt}^*\) obtained by means of numerical simulations with those from literature correlations. In particular, Figure 5(a) analyses the effect of the dimensionless thermal conductivity \(k'_*\), showing a small increase of \(R_{gt}^*\) with it. On the contrary, in Figure 5(b) the dimensionless grout thermal resistance \(R_{gt}^*\) decreases by increasing the dimensionless shank spacing \(d'_*\).

**Figure 3** Transient behaviour of the dimensionless grout thermal resistance \(R_{gt}^*\) for the three boundary conditions and for \(d'_* = 0.5, k'_* = 1, (\rho c)_* = 0.7\). Comparison with literature correlation steady values \((R_{gt}^*_{Paul} = 0.95)\).

**Figure 4** Effect of the dimensionless heat capacity \((\rho c)_*\) on the transient behaviour of the dimensionless grout thermal resistance \(R_{gt}^*\) for the three boundary conditions for \(d'_* = 0.5, k'_* = 0.3, Bi = 10\). Comparison with the literature correlation steady values \((Paul\ 1996\ value\ is\ out\ of\ the\ axis\ range:\ R_{gt}^*_{Paul} = 0.95)\).
One of the most interesting results of this analysis is the different values of the dimensionless grout thermal resistance $R_{gt}^*$ obtained by applying different boundary conditions on pipe side, namely imposed temperature or imposed convective coefficient. In fact, the non-axiallysymmetry of the geometry leads to different temperature fields in the grout, as already stressed by a previous paper of this research group (Fossa and Dalla Pietà, 2011).

Thus, the usual approach of simply adding the convective thermal resistance (often neglecting the pipe’s one) to the grout thermal resistance obtained by imposing a temperature boundary condition, is revealed to be uncorrect and leading to significant errors. On the other hand the comprehensive approach by Bennet et al. is able to efficiently describe the heat transfer characteristics when a convective boundary condition is applied.

Figure 6 presents the results for a double U pipe, comparing the dimensionless grout plus convective thermal resistances $R_{gt+f}^*$ calculated with two different approaches. The value obtained by applying the imposed temperature boundary condition and then adding the convective thermal resistance significantly understimates the value obtained by directly applying the convective boundary condition, with an error that increases for low values of the Biot number. In particular, the average percentage relative error at the steady state asymptote is $\varepsilon = 3.4\%$ for $Bi = 50$ and $\varepsilon = 10.7\%$ for $Bi = 10$. 

**Figure 5**  Steady state values of the dimensionless grout thermal resistance $R_{gt}^*$: (a) Influence of the dimensionless thermal conductivity $k^*$ ($d^* = 0.5$, $Bi = 10$); (b) Influence of the dimensionless shank spacing $d^*$ ($k^* = 0.6$, $Bi = 10$).
CONCLUSIONS

In this paper the grout thermal resistance of a U pipe borehole is analysed by means of numerical simulations, by applying different boundary conditions to the internal pipe surfaces, namely imposed heat flux, imposed temperature and convective conditions with imposed convective coefficient. The study is carried out by using a dimensionless approach and it examines the effect of the ratio between the thermophysical properties of grout and ground, the effect of the shank spacing of the internal pipes, and the effect of the fluidodynamic regime inside the pipes by means of a Biot number parameter.

The transient behaviour of the dimensionless grout thermal resistance reveals different trends for the three boundary conditions especially for the imposed heat flux one, with a steady value higher than the others. The numerical results for convective conditions are similar to the imposed temperature ones, especially increasing the Biot number (turbulent regime inside pipes). Moreover, the dimensionless heat capacity for unit volume plays an important role on the transient behaviour of the dimensionless grout thermal resistance, which increases as the the dimensionless heat capacity is decreased.

The steady state value of the dimensionless grout thermal resistance has been calculated in terms of either the dimensionless thermal conductivity or dimensionless shank spacing and compared with literature correlations. As expected, the resistance slightly increases with grout thermal conductivity and it increases by decreasing the shank spacing. For the single U pipe case, the present numerical results revealed to be in very good agreement with Bennet et al. formula when the convective boundary condition is applied at pipe inner surface.

The inaccuracy associated to calculating the grout thermal resistance with imposed temperature and then simply adding the convective thermal resistance is relevant, with an average percentage relative error with respect to the case of imposed convective boundary coefficient that increases by decreasing the Biot number.

NOMENCLATURE

\[ Bi = \text{Biot number} \]  
\[ d = \text{Half shank spacing (m)} \]  
\[ F_{rb} = \text{Fourier number based on } r_b \]  
\[ b = \text{Convective coefficient (W/m}^2\text{K)} \]  
\[ \kappa = \text{Thermal conductivity (W/m} \cdot \text{K)} \]  
\[ \dot{Q}' = \text{Heat transfer rate per unit length (W/m)} \]  
\[ r = \text{Radius (m)} \]  
\[ R = \text{Thermal resistance (m} \cdot \text{K/W)} \]  
\[ \rho c = \text{Heat capacity per unit volume (J/m}^3\text{K)} \]  
\[ T = \text{Temperature (K)} \]  
\[ \tau = \text{time (s)} \]

Subscripts

\[ b = \text{borehole} \]  
\[ conv = \text{convective} \]  
\[ f = \text{fluid} \]  
\[ g = \text{ground} \]  
\[ gr,\infty = \text{undisturbed ground} \]  
\[ p = \text{pipe} \]
REFERENCES


An Investigation on the Effects of Different Time Resolutions in the Design and Simulation of BHE Fields

Marco Fossa Davide Rolando Antonella Priarone

ABSTRACT

The correct design of a field of Borehole Heat Exchangers (BHE) requires the knowledge of ground thermal properties, heat pump performance and building heating and cooling demand. The sequence of heat pulses from (to) the ground by the heat pump can be described according to different time steps, from hours to months and even years. The monthly time step approach is often the preferred design choice which involves recursive calculations (temporal superposition techniques) and the availability of precalculated temperature response factors (or g-functions) for given BHE field geometries. Such a complex computing task is usually performed thanks to commercial codes in order to fulfil a carrier fluid temperature at the end of a given time horizon, typically 10 or 25 years. In this paper the monthly design approach (EED code and TecGeo proprietary code) is compared with the three thermal pulse approach (modified ASHRAE Method Tp8) and it is demonstrated that for a representative series of case studies the three pulse calculation, easy to be performed at engineering level, is able to provide the correct BHE field overall length with 8% accuracy with respect to the reference monthly calculations.

INTRODUCTION

In ground coupled heat pump (GCHP) applications, the advantages of borehole heat exchangers (BHE) with respect to their shallow counterparts (trench and horizontal ground heat exchangers) are various, including the favourable temperatures of the soil when increasing the distance from the surface, a great freedom in choosing the best disposition of the heat exchangers in the building site, well assessed techniques for drilling the BHE, reliable models for modelling its thermal behaviour and interactions with the surrounding ground for overall system performance simulations. Concerning the theoretical part of the BHE field design process, the best approach for defining the final BHE field geometry for target system performance (in terms of given COP or overall seasonal performance factor parameters) is to employ precalculated temperature response factors (TRF) for performing ground response simulations where the heat transfer rate is variable in time.

The ground volume is very often modelled according to the thermal conduction laws. Starting from the early contributions by Carslaw and Jaeger (1947) and Ingersoll et al. (1954) that provide the ground theoretical response to simple heat sources (Infinite line and cylindrical sources, ILS and ICS, respectively), new models and related TRF have been developed in order to cope with the geometrical complexity of the real BHE fields (Eskilson 1987, Spitler and Yavuzturk 1999, Lamarche and Beauchamp 2007, Claesson and Javed 2011, Cimmino et al. 2013).

The variability of the heat loads at ground level is the consequence of the building heat demand which is
intrinsically transient in time due to the heat pump daily and weekly duty cycles and weather effects. This variability of
the heat transfer rate from (to) the ground can be tackled by applying suitable temporal superposition techniques (e.g.
Spitler and Yavuzturk 1999). As it is well known the time resolution at which the heat load variation can be taken into
account varies typically from hours to months, being the former approach the most reliable according to designers
and researcher in this field. Since hourly simulations are time consuming from a computational point of view and also
because often the input hourly data in terms (building thermal needs) are not available a common approach is to focus
the analysis on a monthly time step, eventually including in the process of temporal superposition single short (hourly
scale) contributions. This criterion is the one adopted in the well known calculation tools GLHEPRO (Spitler 2000)
and EED (Hellstrom and Sanner, 1994). On the other hand since about 20 years a simplified model has been
proposed which is based on the description of the building/ground thermal interactions in terms of only 3
contributions over 10 years (triple step model, Kavanaugh and Rafferty, 1997, Bernier 2006, Ashrae 2015). This
method, known as the Ashrae method, has the fundamental advantage that it can be implemented in simple
spreadsheets and does not need dedicated softwares. Recently the present Authors have developed further schemes
(the Tp8 method, Fossa and Rolando 2016, Fossa 2016) for applying the Ashrae Method to BHE field design that
according to them are either more precise or faster than previous literature methods based on the evaluation of the
Temperature Penalty parameter.

In this paper the Authors present a comprehensive series of comparisons related to monthly (sequence of 121
heat loads) vs triple step simulations devoted to the estimation of the required overall BHE field length. The overall
time horizon is 10 years, taking into consideration different monthly profiles and a variety of BHE field geometries
(e.g. square, in-line configurations). The monthly calculations are performed with either the EED code or the
proprietary code TecGeo (Dalla Pietà and Fossa, 2006).

The aim of the paper is to assess the reliability of the three step approach (in terms of the Tp8 method) with
respect to the monthly approach when the BHE length is estimated based on a constraint related to the minimum
temperature of the carrier fluid during the analysis period. A short discussion is finally provided about the influence of
different TRFs (i.e. based on heat rate or temperature boundary conditions, see Priarone and Fossa 2016) on the final
results of the design process and about the possibility to extend the Tp8 method to longer time horizons (25 years).

THE MODIFIED ASHRAE METHOD ACCORDING TO THE TP8 MODEL

The Tp8 method is a recent model developed by the Authors which exploits the linear properties of the
conduction equation for obtaining reliable formulas able to describe the ground thermal response to BHE systems in
a great variety of BHE geometries.

The model, its accuracy and the comparison with literature similar methods are described in details in three
papers by the Authors (Fossa and Rolando 2015, Fossa and Rolando 2016, Fossa 2016). Here in the following the
main calculation steps of the Tp8 approach are briefly presented.
As it well known the Ashrae method allows the calculation of the BHE field overall required length $L$ in terms of the ground properties (being $k_{gr}$, $\alpha_{gr}$ and $T_{gr,\infty}$ its thermal conductivity, diffusivity, and undisturbed temperature, respectively), the building heat transfer rates $\dot{Q}$ to (from) the ground in three time steps (10 years, 1 month, 6 hours, subscripts $y$, $m$, $h$ respectively), the borehole effective thermal resistance $R_{bhe}$ and the target carried fluid temperature $T_{f,ave}$ at the end of the analysis period ($\tau_N$ equal to 10 years plus 1 month plus 6 hours). The Ashrae method is based on the calculation of the ground thermal resistances $R_y$, $R_m$ and $R_h$ starting from the G solution (also known as the ICS solution, Ingersoll et al. 1954) and introduces a correction term (the Temperature Penalty $T_p$) that represents the

$$L = \frac{\dot{Q} + \dot{Q}_m R_m + \dot{Q}_h (R_h + R_{bhe})}{T_{gr,\infty} - T_{f,ave} (\tau_N) - T_p}$$

(as in the original Ashrae Method 2015 and Bernier 2006)

<table>
<thead>
<tr>
<th>Quantity</th>
<th>Expression</th>
</tr>
</thead>
<tbody>
<tr>
<td>Required length $L$ of the BHE field (10 years)</td>
<td>$L = \frac{\dot{Q} + \dot{Q}<em>m R_m + \dot{Q}<em>h (R_h + R</em>{bhe})}{T</em>{gr,\infty} - T_{f,ave} (\tau_N) - T_p}$</td>
</tr>
</tbody>
</table>

Evaluated according to the G solution (ICS model) values at given time intervals, as in the original Ashrae Method

Temperature Penalty $T_p$ is expressed through an auxiliary temperature variable $\theta_8$ and as a function of the BHE field geometry ($N_1, N_2, N_3, N_4$ parameters)

$$T_p = \frac{\theta_8}{N_{tot}}$$

Auxiliary temperature variable

$$\theta_8 = \frac{\dot{Q}_y E_1(\tau_N, B) + E_1(\tau_N, B\sqrt{2})}{\pi k_{gr} L}$$

Evaluation of the G function

$$G = \sum_{j=0}^{6} c_j \left[ \log_{10}(F_o) \right]^j$$

Evaluation of the Fourier number $F_o^*$ for calculating the $E_1$ function, Eq. (2)

$$F_o^* = \frac{\alpha_{gr} \tau_N}{R^2} \left( \frac{H_{ref}}{H} \right)$$

### Table 1. Ashrae Modified $T_p$8 model: formula set

<table>
<thead>
<tr>
<th>Quantity</th>
<th>Expression</th>
</tr>
</thead>
<tbody>
<tr>
<td>Required length $L$ of the BHE field (10 years)</td>
<td>$L = \frac{\dot{Q} + \dot{Q}<em>m R_m + \dot{Q}<em>h (R_h + R</em>{bhe})}{T</em>{gr,\infty} - T_{f,ave} (\tau_N) - T_p}$</td>
</tr>
<tr>
<td>Ground thermal resistances $R_y$, $R_m$, $R_h$</td>
<td>Evaluated according to the G solution (ICS model) values at given time intervals, as in the original Ashrae Method</td>
</tr>
<tr>
<td>Temperature Penalty $T_p$</td>
<td>$T_p$ is expressed through an auxiliary temperature variable $\theta_8$ and as a function of the BHE field geometry ($N_1, N_2, N_3, N_4$ parameters)</td>
</tr>
<tr>
<td>Auxiliary temperature variable</td>
<td>$\theta_8 = \frac{\dot{Q}<em>y E_1(\tau_N, B) + E_1(\tau_N, B\sqrt{2})}{\pi k</em>{gr} L}$</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Constants $a$, $b$, $c$, $d$</th>
<th></th>
<th></th>
<th></th>
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</thead>
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<tr>
<td>Evaluated according to the G solution (ICS model)</td>
<td>B/H</td>
<td>0.03</td>
<td>0.05</td>
<td>0.075</td>
</tr>
<tr>
<td>Rectangular configurations</td>
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</tr>
<tr>
<td>$a$</td>
<td>5.41</td>
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<td>3.07</td>
<td>2.42</td>
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<tr>
<td>$c$</td>
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<td>0.450</td>
<td>0.450</td>
<td>0.450</td>
</tr>
<tr>
<td>$d$</td>
<td>0</td>
<td>0</td>
<td>0</td>
<td>0</td>
</tr>
<tr>
<td>Evaluation of the G function</td>
<td>(Fossa 2016)</td>
<td>$G = \sum_{j=0}^{6} c_j \left[ \log_{10}(F_o) \right]^j$</td>
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<tr>
<td>$c_0$</td>
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<td>$c_6$</td>
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<td></td>
</tr>
<tr>
<td>Evaluation of the Fourier number $F_o^*$ for calculating the $E_1$ function, Eq. (2)</td>
<td>(Fossa and Rolando, 2016)</td>
<td>$F_o^* = \frac{\alpha_{gr} \tau_N}{R^2} \left( \frac{H_{ref}}{H} \right)$</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>
necessary correction to the simple G solution when it is employed for describing complex borefield geometries in the late period $\tau_N$ (Fossa 2016).

The complete set of formulas for calculating the required length is given in Table 1.

Further symbol meanings are provided: $E_i$ is the exponential integral (Abramovitz and Stegun, 1964), $B$ is the BHE interdistance, $n_i$ is the BHE radius, $H$ is the actual BHE depth and $H_{ref}$ is a reference depth set equal to 100 m, $N_4$, $N_3$, $N_2$ and $N_1$ are the number of boreholes (in the BHE field) surrounded by only 4 other ones, only 3 other ones, and so on, respectively.

**THE PROCEDURE FOR COMPARING THE DESIGN RESULTS IN TERMS OF DIFFERENT TIME RESOLUTIONS**

Temporal superposition is a well known technique for exploiting base solutions of the transient conduction problem for solving GCHP design tasks when the ground/BHE system is subjected to non constant heat load profiles. Even if hourly load simulations are considered the best way to simulate the borefield response and study the GCHP plant performance very often such an analysis is performed based on the knowledge of the building heating and cooling monthly loads. Such thermal loads are usually easily accessible through steady state and even transient simulations of the building (and its thermal plant) interactions with the external environment. When the monthly load approach is applied to the BHE field design problem an additional peak load is usually included in the superposition scheme as discussed in recent studies by the Oklahoma State University research group (e.g. Cullin and Spitler, 2015).

The resulting algorithm is the one embedded in commercial codes like EED or GHLEPRO and in the proprietary code TecGeo developed at the University of Genova Italy:

$$T_{f, ave} (\tau_N) = T_{gr, \infty} + \hat{Q}_{\text{peak}} R_{\text{bhe}} + \frac{1}{2\pi k_{gr}} \sum_{i=1}^{N} (\hat{Q}'_i - \hat{Q}'_{i-1}) g(Fo(\tau_N - \tau_i))$$

$$+ \frac{1}{2\pi k_{gr}} (\hat{Q}_{\text{peak}} - \hat{Q}'_N) g_{\text{peak}} (Fo(\tau_{\text{peak}}))$$

(6)

where the $\hat{Q}'_i$ are the monthly average heat rates (per unit length) to the ground and $g$ is the $g$-function of that BHE field, i.e. the TRF pertaining that borefield assembly in terms of depth $H$, spacing $B$, radius $r_b$ and disposition (e.g. rectangular, in line, and so on). In the above expression $\hat{Q}_{\text{peak}}'$ is the peak load in the worst operating conditions, $\tau_{\text{peak}}$ its duration and $g_{\text{peak}}$ the TRF employed for describing the ground response to this short thermal pulse.

The related heat rates $\hat{Q}_y$ and $\hat{Q}_m$ of the Ashrae method can be calculated from the 12 monthly values $\hat{Q}'_i$ as the average along the year and as the worst month condition, respectively. The peak load $\hat{Q}_{\text{bhe}}$ is just the corresponding quantity $\hat{Q}_{\text{peak}}'$ multiplied by the BHE depth $H$.

In this paper two design approaches are compared related to a time horizon of 10 years.

In the reference procedure 121 monthly heat loads are taken into account and they represent 12 monthly loads (from January to December) that replicates for 10 years. The 121st term is representative of the January contribution of the 11th year, in order to cope with the $\tau_N$ horizon of the Ashrae approach.

The second approach is the T$_{pe}$ Ashrae method (Table 1 equation set) which takes into account the average term $\hat{Q}_y$ and the (worst) month contribution $\hat{Q}_m$ which corresponds in the present analysis to the January monthly heat load. Both procedures include the same peak load. Since the Ashrae method cannot account for multiple peak loads, in monthly simulations a single peak load is superposed at the above worst monthly contribution. On the other hand, while searching for the minimum fluid temperature during the years, the selection of a single peak load applied
A series of test cases have been taken into account in terms of monthly heat load and peak load distributions to cope with the typical heat demand profiles of Italian sites. Heat monthly profiles have been calculated based on local weather conditions while the peak load has been set equal to the January average heat rate divided by a partial load factor (PLF) parameter which has been arbitrary chosen in a range of typical values.

Figure 1a and 1b show the monthly load distributions (energy per month required by the building) for the 5 case cities taken into consideration and they refer to 2 different building sizes, characterized by a heat demand multiplier $M$ equal to 1 and 10, respectively. Figure 1a is a general case where either the heating and cooling loads are taken into consideration while Figure 1b refer to a condition where the GCHP duty is only heating, say to work only in winter mode. This last condition is addressed to emphasize the role of the correct evaluation of the $T_p$ term (Fossa 2016), which is proportional to $\dot{Q}_r$ and hence gains importance when the building yearly load profile is markedly unbalanced (winter only or summer only operating modes).

Figures 2a and 2b show the distribution of the peak loads (heat rate required by the building, heating operation only is case b) for the 5 case cities and considering a PLF parameter assuming the values 0.3, 0.4 and 0.5.

Figures 1 Monthly load profiles taken as input values for BHE field design. Condition (b) refers to heating operations only

Figure 2 Peak loads taken as input values for BHE field design. Three different conditions have been considered as the effect of different partial load factors (PLF) Condition (b) refers to heating operations only
In spite of the fact that the selected input data set for the present design calculations is necessarily arbitrary it can be stated that it is a reasonably comprehensive of many working conditions encountered by GCHP plants in Europe. Further data input are the seasonal performance factor (SPF) which has been set to 4, $R_{bhe}$ set to 0.10 mK/W, $k_{gr}$ set to 2.5 mK/W, $\alpha_{gr}$ set to 11e-7 m²/s and the undisturbed local ground temperature of each case city ($T_{gr,\infty}$ range: 10 to 16°C).

The calculations have been performed for attaining a minimum carrier fluid temperature of 3°C when the BHE field is characterized by interdistances $B$ equal to 5 meters. The BHE field geometry has been selected among the rectangular and in-line configurations (e.g 2x1, 3x2, 4x10, 6x7) in order to fulfill the carrier fluid constraint with BHE depths not exceeding the range 85 to 120 m.

**Figures 3** Calculated overall BHE length $L$ with the reference method (EED) and the Ashrae/TP8 one. $M=1$ (a) and $M=10$ (b)

**Figures 4** Monthly fluid temperature along the years as calculated with different monthly algorithms. City #2, PLF=0.4, $M=10$, winter only loads.

Filled symbols: EED predictions; open symbols: TecGeo predictions

In spite of the fact that the selected input data set for the present design calculations is necessarily arbitrary it can be stated that it is a reasonably comprehensive of many working conditions encountered by GCHP plants in Europe. Further data input are the seasonal performance factor (SPF) which has been set to 4, $R_{bhe}$ set to 0.10 mK/W, $k_{gr}$ set to 2.5 mK/W, $\alpha_{gr}$ set to 11e-7 m²/s and the undisturbed local ground temperature of each case city ($T_{gr,\infty}$ range: 10 to 16°C).

The calculations have been performed for attaining a minimum carrier fluid temperature of 3°C when the BHE field is characterized by interdistances $B$ equal to 5 meters. The BHE field geometry has been selected among the rectangular and in-line configurations (e.g 2x1, 3x2, 4x10, 6x7) in order to fulfill the carrier fluid constraint with BHE depths not exceeding the range 85 to 120 m.
RESULTS

Simulations of the fluid temperature evolution in time have been performed with the EED code while imposing the minimum allowed fluid temperature (3°C) for the given BHE field configuration. The same constraint have been applied to the Tp8 algorithm where the constants a, b, c, d and the BHE types N1, N2, N3, N4 have been calculated as a function of the BHE field configuration, the same of the corresponding EED case. With both procedures the overall BHE lengths (and hence the corresponding BHE depths) have been evaluated.

The peak load duration has been set equal to 6 hours, as suggested in the original Ashrae method.

Figure 3a shows the final overall BHE length as calculated with both the EED code and the modified Ashrae method for the 5 case cities, at different PLF numbers (e.g. at different peak loads for given base load set) when the base load multiplier M is equal to 1 (monthly load input values: left y-axis of figures 1a and 1b). The resulting BHE configurations resulted 2x1, 3x1 and 4x1 for overall lengths not exceeding 500 m. It can be noticed that the monthly design procedure and the three pulse one give practically the same results, with minor discrepancies in the 3% range. This error is related to the intrinsic error in Tp8 method for small BHE fields as a consequence of its best fit constants.

Figure 3b is the M=10 version of the previous one: the building heat loads are greater and the resulting BHE fields are larger (up to 7x10 BHEs). Here the agreement among the reference EED calculation and the Ashrae/Tp8 one is still good but major differences arise, of the order of 8% (in the direction of oversizing the BHE system).

The above discrepancies could be ascribed to the different TRFs employed in EED with respect to those employed during the optimum search of Tp8 constants (Table 1). As deeply discussed in Fossa and Rolando 2015 and Fossa and Priarone 2016, TRFs (g-functions) can be calculated according to different boundary conditions (i.e. at imposed heat rate, as in TecGeo and Tp8, or temperature, as in EED) but in the Fo range pertaining to 10 year horizons they should not differ meaningfully.

Figure 4 finally shows the comparison between the time evolution of the carrier fluid temperature for the city #2, M=10, PLF=0.4 as calculated by EED and by the proprietary code TecGeo. The two codes should implement the same overall algorithm while employing different g-function databases. In particular TecGeo works with the same g-functions employed for best fit optimization of Tp8 method constants. The BHE configuration is this case is 6x7, B is equal to 5 m and H=107 m. As can be observed the temperature profiles are in close agreement and the maximum fluid temperature difference is about 0.6°C at January of the 11th year. The exact agreement in terms of minimum fluid temperature it would be obtained by setting H_{EED}=103 m. On the other hand when TecGeo is employed for the same case for calculating the required length, the comparison with the Tp8 prediction reveals a difference of less 3% (L_{TGC}=107 m, L_{8}=110 m). It is interesting to notice that for the same input data the original Ashrae method (Kavanaugh and Rafferty, 1997) would provide an overall BHE length of only 82 m.

These evidences deserve further investigation and maybe a critical review of EED database of TRF functions.

Finally it is worth stressing that a series of simulations have been carried at different τ_peak values (from 3 to 12 hours) that provided results still in agreement among them (again 8% differences between EED and Tp8 predictions), thus demonstrating once again the reliability of the modified Ashrae method.

CONCLUSIONS

A series of case tests related to the design of BHE fields have been considered in order to compare the results obtained with a monthly heat load approach with those provided by a modified version of the Ashrae method (the Tp8 method). The simulations have been performed with respect to a 10 year scenario (121 months) and considering various BHE field configurations, from small in-line geometries (1x2 to 1x4) to large rectangular dispositions (up to 7x10). The overall BHE length ranged from few hundreds meters to 7 thousands.

The paper demonstrates that the modified Ashrae method is able to provide similar results (differences within 8%) with respect to the reference calculation procedure (EED code) when the design target is the overall BHE length for given minimum allowed fluid temperature.
The differences in the predicted lengths can be partially ascribed to the different sets of g-functions employed by the reference code and the present procedure: further investigations will be devoted to clarify these aspects.

Current work on the $T_{p8}$ method is finally addressed in the direction of extending its applicability to a 25 year horizon through model tuning in terms of constants and superposition scheme.

ACKNOWLEDGMENTS

The Authors acknowledge Dr. Silvia Trevisan for the valuable calculations she performed during the development of her Thesis at the University of Genova.

NOMENCLATURE

\begin{itemize}
  \item $B$ = BHE separating distance (m)
  \item $F_o$ = Fourier number, $F_o= (\alpha \tau)/r_b^2$ (-)
  \item $F_o'$ = Modified Fourier number, $F_o'=[(\alpha \tau)/R^2](H_{ref}/H)$, Eq. 3 and 5 (-)
  \item $E_i$ = exponential integral (-)
  \item $G$ = ICS temperature response factor (-)
  \item $H$ = BHE depth (m)
  \item $k$ = Thermal conductivity (W/mK)
  \item $L$ = BHE overall length (m)
  \item $M$ = heat demand multiplier (-)
  \item $N$ = BHE number (-)
  \item $PLF$ = partial load factor (-)
  \item $Q$ = Heat (J or MWh)
  \item $\dot{Q}$ = Heat transfer rate (W)
  \item $\dot{Q}'$ = Heat transfer rate per unit length (W/m)
  \item $r_b$ = Borehole radius (m)
  \item $R$ = Distance parameter in $T_{p8}$ scheme (Eq. 3). $R$ is equal to B or $B\sqrt{2}$ (m)
  \item $R_i$ = Thermal resistance (mK/W)
  \item $T$ = Temperature ($^\circ$C or K)
  \item $T_p$ = Temperature penalty parameter ($^\circ$C or K)
  \item $\alpha$ = Thermal diffusivity (m$^2$/s)
  \item $\theta_b$ = Auxiliary temperature variable ($^\circ$C)
  \item $\tau$ = Time (s)
\end{itemize}

Subscripts

\begin{itemize}
  \item $\infty$ = undisturbed conditions
  \item $8$ = $T_{p8}$ method
  \item $\text{build}$ = at building side
  \item $\text{bhe}$ = relative to the BHE
  \item $\text{EED}$ = EED code
  \item $\text{fluid, ave}$ = of the carrier fluid, average in and out value
\end{itemize}


REFERENCES


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Heat pump capacity effects on peak electricity consumption and total length of self- and solar-assisted shallow ground heat exchanger networks

Parham Eslami Nejad  Massimo Cimmino  Sophie Hosatte-Ducassy

ABSTRACT

A new “self-assisted” Ground Source Heat Pump (GSHP) system configuration is proposed to address the relatively high peak electricity demand of undersized GSHP systems equipped with auxiliary electric heater. In this configuration, ground heat exchangers (GHE) have two independent circuits: the first circuit is used to inject the extra heat produced by the heat pump into the ground during off-peak operations, while the second circuit is used to extract heat in the winter and reject heat in the summer for space heating and cooling, respectively. This configuration is compared against a “solar-assisted” configuration and a conventional single U-tube configuration. An analytical model for shallow GHE networks is used to evaluate the effects of the heat pump nominal capacity and the borehole total length on the total electricity consumption and peak electricity demand of the three configurations. Results show that the self-assisted configuration reduces the peak electricity demand by 47%, in a case with a 29% undersized GHE network and a 16% undersized heat pump nominal capacity, while it increases the total energy consumption by 4.1%. Using a solar-assisted configuration for the same sizing parameters reduces the peak electricity demand by only 6.3% and the total energy consumption by 3.8%.

INTRODUCTION

Ground source heat pump (GSHP) systems offer significant energy saving potential in cold climates because of their relatively good seasonal performance in both heating and cooling applications. However, due to their high initial costs, the most expensive system components (e.g. heat pumps and GHEs) are often deliberately undersized, leading to higher peak electricity demands in comparison to other market available heating and cooling systems. A so-called solar-assisted alternative, which integrates solar thermal technologies with GSHP systems, has been shown to decrease the negative impacts of undersized heat pump nominal capacity and borehole length in cold climates (Kjellsson, et al. 2010; Chiasson and Yavuzturk 2003; Han et al. 2008). Among several possible approaches, the direct injection of available solar heat into the ground using shared boreholes has been favored in several recent studies due to its simplified system integration (Eslami Nejad and Bernier 2011; Belzile, et al. 2016).

However, solar-assisted geothermal heat pumps suffer from the fact that heat can only be stored into the ground when solar thermal energy is available, which may differ from the time of peak heating loads. Additionnally, large amounts of solar thermal energy are required to achieve savings in terms of GHE size. To diminish these shortcomings, a “self-assisted” GSHP configuration is examined. This configuration uses the heat pump to store surplus heat into the field of GHEs through a secondary fluid loop. The effects of heat pump nominal capacity and total borehole length on

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the electricity consumption and peak electricity demand are evaluated for the case of a single family detached home located in Montreal. The newly proposed self-assisted configuration is compared against the solar-assisted and conventional single U-tube configurations for four different cases: Case 1 with properly sized borehole network and heat pump capacity, Case 2 with reduced heat pump nominal capacity, Case 3 with shortened borehole network, and Case 4 with combined undersized borehole network and heat pump nominal capacity.

**SYSTEM DESCRIPTION**

The seasonal performance of three configurations, self-assisted, solar-assisted and conventional configurations, is assessed under given space heating and cooling load profiles for a 210 m² single family detached home located in Montreal. Figure 1 presents the building load (left), the annual outdoor temperature profile (center) and the total tilted solar radiation on solar collectors (right) taken from a TMY2 (National Solar Radiation Data Base) weather file for Montreal. The peak hourly building load is 14.18 kW and 4.21 kW for heating and cooling respectively. Annual space heating and cooling requirements are 29266 kWh and 1682 kWh respectively. More details about the building model and validation are found in Kegel et al. (2012a, 2012b). The same outdoor temperatures, solar radiation and building loads are used for every year of the simulations.

![Figure 1](image)

*Figure 1  Building loads (left), ambient temperature (center) and incident solar radiation (right)*

In all configurations, the heating and cooling system consists of a geothermal heat pump connected to an array of 25 (5 × 5) ground heat exchangers in a zig-zag pattern, as shown in Figure 2. When used, Circuit 2 (shown in red) has the heat carrier fluid flowing in opposite direction to Circuit 1 (shown in blue). Auxiliary heating is provided by an electric auxiliary heating system when the heat pump capacity is insufficient to cover the building heating loads. Two heat pumps with nominal capacities of 15.2 kW and 12.7 kW are considered. The variation of the heat pump capacities and the coefficient of performance with regards to entering fluid temperature are presented in Figure 3.

![Figure 2](image)

*Figure 2  Sample zig-zag pattern for a field of 3 × 3 boreholes*
Figure 3  Heat pump capacity and COP as a function of entering fluid temperature

For all cases considered, the spacing between boreholes is $B = 2$ m. The boreholes have a radius $r_b = 0.042$ m and are buried at a distance $D = 1$ m below the ground surface. U-tube pipes have an inner radius $r_{p,i} = 0.0164$ m, an outer radius $r_{p,o} = 0.0167$ m and are equally spaced at a distance $D_s = 0.024$ m from the center of the boreholes. The thermal conductivities of the ground, the grout and the pipes are $k_g = 2.65$ W/m-K, $k_b = 1$ W/m-K and $k_p = 0.4$ W/m-K, respectively. The ground thermal diffusivity is $\alpha_s = 0.08$ m$^2$/day. The fluid flow rate is $m_1 = m_2 = 0.5$ kg/s in both circuits, when applicable, with a fluid specific heat of 4000 J/kg-K (propylene-glycol 20%) and a fluid convection coefficient of 1500 W/m$^2$-K. The minimum allowed fluid temperature is $T_{f,\text{min}} = -2^\circ$C in all cases. The borehole length $H$ and the nominal heat pump capacity vary from case to case.

Conventional Configuration

The first system is a conventional ground source heat pump system, which provides a base of comparison for the other two systems proposed in this paper. In this system, the heat pump is connected to single U-tube vertical boreholes (Figure 4a). The heat pump covers both the heating and cooling loads. Auxiliary electric heating is used when the heat pump capacity is insufficient.

Figure 4  Schematic of (a) conventional GSHP, (b) solar assisted GSHP, and (c) self-assisted GSHP systems

At all times when the heat pump is operating, the total heat extraction rate (positive for extraction, negative for injection) from the borehole (via Circuit 1) is given by the heat pump performance data based on the outlet fluid temperature from the borefield:
Solar-Assisted Configuration

The solar-assisted configuration is equipped with double U-tube vertical boreholes (two circuits). Circuit 1 is connected to the heat pump while Circuit 2 is connected to the solar collectors (Figure 4b). Via Circuit 2, solar heat is injected into the ground at all times when it is available. In heating dominated climates, where the ground temperature typically decreases due to unbalanced loads, this helps to increase the ground temperature and store heat in the borefield. Using Circuit 1, the heat pump meets the space heating and cooling loads, as in the conventional GSHP system. Auxiliary heat is used when the heat pump heating capacity is insufficient. Circuit 2 is connected to an array of solar collectors with total area $A_{col} = 5 \text{ m}^2$. The collector efficiency $\eta$ varies with the ambient temperature $T_a$, the outlet fluid temperature from the borefield (and into the solar collectors) $T_{out,2}$, and the incident solar radiation $I$:

$$Q_{circuit,2} = -\eta A_{col} \cdot I$$  \hspace{1cm} (2)

$$\eta = F_R - \alpha \frac{T_{out,2} - T_a}{I}$$  \hspace{1cm} (3)

where $Q_{circuit,2}$ is the total heat extracted from Circuit 2 (negative for heat injection), $F_R = 0.86$ is the collector heat removal factor and $\alpha = 5.33$ is a parameter associated with the heat losses of the collector to the ambient air.

Self-Assisted Configuration

Double U-tube vertical boreholes are also used in the self-assisted configuration. However, Circuit 2 is only used during the colder winter months, at certain selected periods prior to peak heating conditions when extra heat produced by the heat pump is available to be injected into the ground (Figure 4c). The time period of injection before the peak is determined using simulations to minimize (i) the peak electricity demand and (ii) the heat pump energy consumption for the configuration. Heat injection will increase the ground temperature and store heat in the cluster of shallow boreholes, which may lead to an increase in heat pump capacity during peak conditions and, therefore, lower peak electricity demands.

As in the other configurations, Circuit 1 is connected to the heat pump. However, in this case Circuit 2 is connected to the condenser of the heat pump on the building side. Even though in reality the heat pump system would cycle between heat extraction and self-assisted modes, the two processes are treated as simultaneous in the simulations due to the time step of 1 hour. Whenever the self-assisted mode is allowed, Circuit 2 is used to inject heat into the ground if there is more than 20% of the heat pump capacity leftover after satisfying the building loads. The total rates of heat extraction in Circuits 1 and 2 are given by:

$$Q_{circuit,1} = \begin{cases} \min(Q_{Heating}, \frac{CAP_{Heating}}{COP_{Heating}}) \cdot \frac{COP_{Heating}^{-1}}{COP_{Heating}} \cdot (\text{Heating mode}) \\ \min(Q_{Cooling}, \frac{CAP_{Cooling}}{COP_{Cooling}}) \cdot \frac{COP_{Cooling}^{-1}}{COP_{Cooling}} \cdot (\text{Cooling mode}) \end{cases}$$  \hspace{1cm} (1)

where $Q_{circuit,1}$ is the total heat extraction rate from the heat pump circuit, $Q_{Heating}$ and $Q_{Cooling}$ are the building heating and cooling loads, $CAP_{Heating}$ and $CAP_{Cooling}$ are the heat pump heating and cooling capacities and $COP_{Heating}$ and $COP_{Cooling}$ are the heat pump coefficients of performance in heating and cooling modes.

$$Q_{circuit,1} = \begin{cases} \min(Q_{Heating}, \frac{CAP_{Heating}}{COP_{Heating}}) \cdot \frac{COP_{Heating}^{-1}}{COP_{Heating}} \cdot (\text{Heating mode}) \\ \min(Q_{Cooling}, \frac{CAP_{Cooling}}{COP_{Cooling}}) \cdot \frac{COP_{Cooling}^{-1}}{COP_{Cooling}} \cdot (\text{Cooling mode}) \end{cases}$$  \hspace{1cm} (4)
\[ Q_{\text{circuit,2}} = \begin{cases} 0 & \text{if } Q_{\text{heating}} \geq 0.8 \text{CAP}_{\text{heating}} \\ (\text{CAP}_{\text{heating}} - Q_{\text{heating}}) \cdot \frac{\text{COP}_{\text{heating}}^{-1}}{\text{COP}_{\text{heating}}} & \text{if } Q_{\text{heating}} < 0.8 \text{CAP}_{\text{heating}} \end{cases} \] (5)

**SIMULATION MODEL**

The ground-source heat pump (GSHP) system is simulated using the simulation model of Cimmino and Eslami-Nejad (2016). Boreholes and horizontal sections of pipes are modeled as finite line sources with uniform heat extraction rates. The borehole wall and horizontal section of pipe temperatures are obtained from the spatial and temporal superpositions of the finite line source (FLS) solution for all boreholes and horizontal sections of pipes. The load aggregation algorithm of Claesson and Javed (2012) is used for the temporal superposition of heat extraction rates. Outlet fluid temperatures are calculated from the inlet fluid temperatures and borehole wall temperatures based on analytical solutions for double U-tube boreholes (Eslami-Nejad and Bernetier 2011) when both circuits are active, and single U-tube boreholes (Hellström 1991) when only one is active. The heat pump coefficient of performance (COP), capacity and power consumption are loaded from external data files. Ground surface effects are considered by using a correlation (Badache et al. 2016) for the calculation of the ground temperature at \( z = D \) for horizontal sections of pipes and for the calculation of the average ground temperature over \( D \leq z \leq D + H \) for boreholes.

The simulation model of Cimmino and Eslami-Nejad (2016) was modified to account for the minimum operating fluid temperature \( T_{f,\text{min}} \). Auxiliary heating is used instead of the heat pump if the inlet fluid temperature into the borefield would fall below the minimum temperature by using the heat pump. In these cases, the total heat extraction rate from Circuit 1 is set to zero. At each simulation time step, the simulation model follows this sequence:

1. Calculate aggregated heat extraction rates from previous time steps.
2. Evaluate ground temperatures from correlations.
3. Read building heating and cooling loads and meteorological data from external files.
4. Evaluate the total heat extraction rate from Circuit 1 based on the building load and the values of the heat pump COP and the heat pump capacity calculated based on instant value of returning fluid temperature.
5. Evaluate the total heat extraction rate from Circuit 2:
   a. In the solar-assisted configuration, if solar radiation is available (i.e. \( Q_{\text{circuit,2}} < 0 \)), evaluate the total heat extraction rate based on the solar collector efficiency and the current value of returning temperature.
   b. In the self-assisted configuration, the total heat extraction rate is calculated from the remaining heat pump capacity.
6. Calculate borehole wall and fluid temperatures from the FLS model and borehole analytical solutions respectively.
7. Evaluate auxiliary heating power:
   a. If inlet fluid temperature \( T_{f,1,\text{in}} \) into Circuit 1 is above the minimum allowed fluid temperature \( T_{f,\text{min}} \), then auxiliary heating power is the difference between the building load and the heat pump capacity.
   b. If inlet fluid temperature \( T_{f,1,\text{in}} \) into Circuit 1 is below the minimum allowed fluid temperature \( T_{f,\text{min}} \), then auxiliary heating power is equal to the building load and the total heat extraction rate is set to 0.
8. Re-evaluate heat pump COP, capacity and power consumption and re-calculate fluid and borehole wall temperatures and auxiliary heating power until convergence is reached.

**RESULTS**

In this study, the peak electricity demand and the total electricity consumption of the systems are compared for different borehole lengths and two distinct nominal heat pump capacities in hourly simulations over a period of 5 years. Four cases are selected to investigate the effect of heat pump nominal capacity and borehole length. For Case 1 and Case 2, boreholes are sized adequately and the solar-assisted and conventional configurations are compared under
different heat pump sizes. In Case 3, the borehole length is undersized by 19% and the heat pump size is kept unchanged. In Case 4, the borehole length and heat pump nominal capacity are undersized by 29% and 16%, respectively. The self-assisted configuration is investigated only in Cases 3 and 4 since under properly sized borehole cases (Cases 1 and 2), this configuration does not show any benefits.

**Case 1**

In Case 1, the length of each borehole and the nominal capacity of the heat pump are determined so that almost no auxiliary energy input is needed for both the solar-assisted and conventional GSHP systems (Table 1 and Figure 5). Based on the annual simulation, each borehole has a length of 21 m (Table 1) and the heat pump nominal capacity is 15.2 kW. In this case, solar injection in the solar-assisted configuration has a negligible effect on the fifth-year energy consumption (0.35%) despite significant heat injection (~6000 kWh) into the ground (Table 1: Case 1). Since auxiliary heating is concentrated in the first few months, energy consumption will only be shown for the first 3 months of the fifth simulation year in subsequent cases.

**Table 1. Simulation Results of Case 1 and Case 2**

<table>
<thead>
<tr>
<th>Result metric</th>
<th>Units</th>
<th>Case 1</th>
<th>Case 2</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td>Conventional</td>
<td>Solar Assisted</td>
</tr>
<tr>
<td>Borehole length (m)</td>
<td></td>
<td>21</td>
<td>21</td>
</tr>
<tr>
<td>Heat pump capacity (kW)</td>
<td></td>
<td>15.2</td>
<td>15.2</td>
</tr>
<tr>
<td>Minimum returning fluid temperature (°C)</td>
<td></td>
<td>-1.78</td>
<td>-1.02</td>
</tr>
<tr>
<td>Total energy consumption (year 5) (kWh)</td>
<td></td>
<td>10962</td>
<td>10923</td>
</tr>
<tr>
<td>Reduction relative to case 1 (conventional) (%)</td>
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<td>0</td>
<td>0.35</td>
</tr>
<tr>
<td>Auxiliary heating (year 5) (kWh)</td>
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<td>1.18</td>
<td>0</td>
</tr>
<tr>
<td>Peak energy demand (kW)</td>
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<td>5.29</td>
<td>5.07</td>
</tr>
<tr>
<td>Heat injected into ground (year 5) (kWh)</td>
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<td>2173</td>
<td>8248</td>
</tr>
<tr>
<td>Heat extracted from ground (year 5) (kWh)</td>
<td></td>
<td>18792</td>
<td>18834</td>
</tr>
</tbody>
</table>

**Figure 5** Energy consumption of the conventional (left) and solar-assisted (right) configurations during year 5 for Case 1

**Figure 6** Energy consumption of the conventional (left) and solar-assisted (right) configurations during the first 3 months of year 5 for Case 2

**Case 2**

The same borehole length is used in Case 2 with a reduced heat pump nominal capacity (12.7 kW). In this case, the total energy consumption during the fifth year increased by 2.4% and 1.9% for conventional and solar-assisted configurations, respectively, compared to the conventional configuration in Case 1 (Table 1). Peak energy demand increased by almost 1.5 kW compared to Case 1 (Figures 5 and 6). Similar to Case 1, despite significant solar heat injection into the ground, both the total energy use and peak energy demand reductions are very marginal (0.5% and 3.2%, respectively, compared to the conventional configuration of the same case).
Case 3

In Case 3, shorter boreholes (17 m each) are used in combination with an adequately sized heat pump (15.2 kW). For the self-assisted configuration, heat pump capacity (beyond what is required to meet the space heating loads) is injected into the ground for 18 days prior to peak heating conditions on February 15th (Figure 7, right, red line). This time period was selected based on exploratory simulations so that auxiliary heating is used only marginally throughout the heating season. The solar assisted case followed the same control sequence as in the previous cases. Results show that the solar-assisted and self-assisted configurations decreased the peak energy demand by 0.8% and 59%, respectively, compared to the conventional configuration (Table 2: Case 3). Furthermore, the self-assisted configuration requires almost no auxiliary heating. However, it increased the fifth-year energy consumption by 6.4% due to increased heat pump energy use during off-peak operations. The solar-assisted configuration decreased the energy consumption during the fifth year by 1.8% (Table 2: Case 3).

Figure 7  Energy consumption of the conventional (left), solar-assisted (center) and self-assisted (right) configurations during the first 3 months of year 5 for Case 3

Case 4

In Case 4, both the borehole network and the heat pump are selected to be smaller than that of the base case (Case 1). Each borehole is 15 m long (29% reduction relative to the base case) and the heat pump nominal capacity is 12.7 kW. For the self-assisted configuration, excess heat pump capacity (above what is required for building heating loads) is injected into the ground during two periods; (i) for 29 days prior to February 15th and (ii), for 2 days prior to
March 7th (Figure 8, right, red line). As in Case 3, this time period was selected so that auxiliary heating is used only marginally throughout the heating season. A significant reduction of the peak energy demand has been calculated (47%) for the self-assisted configuration, while the peak is reduced by only 6.3% using the solar-assisted configuration (Table 2: Case 4) despite significant solar thermal energy injection into the bore field when compared to the conventional and self-assisted configurations. The solar-assisted configuration decreases the fifth-year energy consumption by 3.8% while the results for the self-assisted configuration show an increase of 4.1% compared to the conventional configuration. The energy consumption increase of the self-assisted configuration is mainly due to more frequent heat pump operations during off-peak hours as the auxiliary heat is consumed equally in both configurations (182 kWh in Table 2: Case 4).

**Table 2. Simulation Results of Case 3 and Case 4**

<table>
<thead>
<tr>
<th></th>
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<th></th>
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<tbody>
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<td>Borehole length</td>
<td>(m)</td>
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<td>17</td>
<td>17</td>
<td>15</td>
<td>15</td>
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<tr>
<td>Reduction relative to case 1</td>
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<td>Heat pump capacity</td>
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<td>15.2</td>
<td>12.7</td>
<td>12.7</td>
<td>12.7</td>
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<tr>
<td>Minimum returning fluid temperature</td>
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<td>-1.98</td>
<td>-2.00</td>
<td>-2.00</td>
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<tr>
<td>Total energy consumption (year 5)</td>
<td>(kWh)</td>
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<td>11727</td>
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<td>Reduction relative to case 1</td>
<td>(%)</td>
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<td>-2.88</td>
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<td>Auxiliary heating (year 5)</td>
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<td>3.56</td>
<td>831</td>
<td>182</td>
<td>182</td>
</tr>
<tr>
<td>Peak energy demand</td>
<td>(kW)</td>
<td>13.70</td>
<td>13.59</td>
<td>5.55</td>
<td>13.64</td>
<td>12.78</td>
<td>7.21</td>
</tr>
<tr>
<td>Heat injected into ground (year 5)</td>
<td>(kWh)</td>
<td>2173</td>
<td>8257</td>
<td>3817</td>
<td>2166</td>
<td>8244</td>
<td>3738</td>
</tr>
<tr>
<td>Heat extracted from ground (year 5)</td>
<td>(kWh)</td>
<td>18573</td>
<td>18780</td>
<td>18779</td>
<td>18005</td>
<td>18461</td>
<td>18432</td>
</tr>
</tbody>
</table>

**CONCLUSION**

In this study, a shallow ground heat exchanger network is connected to a heat pump to satisfy space heating and cooling of a single family residential building located in a cold climate. The effect of the heat pump and the GHE sizes on the total electricity use and peak electricity demand of the system was investigated for three different configurations. A new “self-assisted” configuration is compared to a solar-assisted and a conventional GSHP system.

Under a properly sized shallow borehole network, it is shown that the solar-assisted GSHP system does not noticeably reduce the total energy use or the peak energy demand. However, it improves the system energy consumption for an undersized borehole network. The self-assisted configuration can significantly lower the peak electricity demand and use of auxiliary heating when the borehole network is undersized, but it increases the heat pump energy consumption. In the presented Cases 3 and 4, the peak energy demand was reduced by 59% and 47% while the total energy consumption increased by 6.4% (710 kWh) and 4.1% (480 kWh) when compared to the corresponding conventional system. Case 4 has thus shown that appreciable peak energy demand reduction can be achieved at relatively low energy costs. Overall, significant borehole length and heat pump size reduction can be envisioned using self-assisted configuration. However, the presented self-assisted geothermal heat pump system relies on the prediction of the time of the peak heating load. Proper control strategies, such as model predictive control, should then be envisioned for this system and will be addressed in future work.

**ACKNOWLEDGMENTS**

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REFERENCES


National Solar Radiation Data Base, web address: http://sel.me.wisc.edu/trnsys/weather/tmy2data.htm
A comparison of the energy use for different heat transfer fluids in geothermal systems

Laurent Gagné-Boisvert  
Michel Bernier

ABSTRACT

Geothermal systems that operate under 0 °C must use antifreeze mixtures instead of water to avoid operational problems. This paper examines the energy consumption of the circulating pump and heat pump for various heat transfer fluids used in a residential geothermal system. Propylene glycol, ethanol and methanol solutions at different concentrations are compared. Effects of fluid temperature and viscosity on head losses, borehole thermal resistance and heat pump operation are reviewed. Efficiency curves for currently available circulators are proposed. Annual energy simulations are then performed on a residential GCHP system. Energy consumption (pump and heat pump) is evaluated subhourly based on fluid temperature and properties prevailing during each time step. Results show, as expected, that higher mixture concentrations and higher flow rates lead to higher energy consumption. Methanol with a concentration of 15% and a 1.5 gpm/ton flow rate provides the best energy performances while ethanol at 30% with 3 gpm/ton is the worst choice, requiring 16% more energy and 525% more pumping power than for the methanol case. Laminar flow in boreholes appears to be favorable when compared to turbulent flow which leads to relatively high pumping energy consumption. Shorter boreholes piped in parallel decrease energy consumption as well.

INTRODUCTION

A typical residential ground-coupled heat pump (GCHP) system is presented in Figure 1. When such a system operates under 0 °C, an antifreeze mixture must be used to avoid operational problems. Typically, designers select a solution with a freezing point approximately 3 °C (5 °F) lower than the lowest anticipated temperature (Dow, 2001). Antifreeze solutions affect the performance of the system in many ways. The greater viscosity of these fluids may lead to laminar flow in boreholes with a corresponding increase of the borehole thermal resistance. Pressure drops in the various parts of the system (ΔpPipe, ΔpBore, ΔpHP and ΔpValve in Figure 1) as well as pumping power (WPump) are increased when a fluid other than water is used. Moreover, antifreeze mixtures affect heat transfer in the source-side heat exchanger of the heat pump, which decreases heat pump capacity (QCap) and heat pump input power (WHP) but to a lesser extent. The objective of this paper is to study the total energy consumption (pump and heat pump) for various heat transfer fluids typically used in GCHP systems.

Figure 1  Representation of a residential GCHP system.  
Figure 2  Overall efficiency of available circulators.

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LITERATURE REVIEW

Solutions of methanol, ethanol and propylene glycol are often used in GCHP systems. An ASHRAE sponsored study (Heinonen, 1997) concluded that propylene glycol was the best compromise mainly because of its low environmental risk despite the fact that systems with propylene glycol use more energy as noted by Bernier et al. (2005). The toxicity of the inhibitors added to propylene glycol solutions must however be considered.

Pumping energy, which increases with the use of antifreeze, is sometimes considered negligible when compared to the overall energy consumption of a geothermal system. In reality, it can represent up to 45% of the total energy consumption, even in recent geothermal systems (Liu et al., 2015). Pumping energy is also influenced by the pumping strategy (Kavanaugh and McInerny, 2001). Kavanaugh and Kavanaugh (2012) suggested a maximum ground loop pump power of 10 hp/100 tons (2.1 kW_therm/100 kW_therm).

A decrement factor ($DF$) evaluating heat pump convective heat transfer variations due to the use of antifreeze mixtures was developed by Spitler and Jin (2003). The $DF$ corrects the fluid-side heat transfer coefficient initially calculated for water. In a follow-up study, Khan and Spitler (2004) stated that propylene glycol increases system energy consumption by 6 to 7% compared to ethanol or methanol. In a residential case study, with relatively balanced heating and cooling loads and low antifreeze concentrations, they concluded that typical antifreeze mixtures have similar life-cycle costs while water presents a higher life-cycle cost because a longer borehole is required.

Spitler and Ghelin (2015) challenged the standard industry recommendations to have turbulent flow in the borehole at all times and to maintain head losses in the range of 1 to 3 ft/100 ft of pipe (10 to 29 kPa/100 m). Their work confirmed Kavanaugh's warning (2011) that high fluid velocities may result in high pumping power with little thermal benefit and even less economic advantage over occasional laminar flow. They also confirmed Mescher's guideline expressed in an ASHRAE Webcast (ASHRAE, 2011) stating that a properly designed bore field should have a head loss ($\Delta p_{bore}$) of less than 25 ft (75 kPa) with a maximum total system pressure drop of 50 ft (150 kPa).

CIRCULATOR EFFICIENCY

Circulators are low power pumps typically used to circulate fluid in residential, one-pipe and decentralized GCHP systems. Until recently, circulators had typical efficiency around 20 to 25% (Kavanaugh and Rafferty, 2015). However, the efficiency of circulators has nearly doubled in recent years (Bidstrup, 2012). Following the methodology used by the COSTIC (2003), an in-house analysis performed for the present study examined the efficiency of 86 commercially available circulators from two manufacturers (Grundfos, 2016 and Salmson, 2016). In each case, the best efficiency point (BEP) was used to extract the nominal overall efficiency at a given nominal hydraulic power. The results of this analysis are shown in Figure 2, which shows the circulator wire-to-water efficiency as a function of the hydraulic power in the 0 to 300 W range. Circulators are categorized into three classes (Low, High and Best Efficiency) each with its own regression equation. In this work, circulators with the “High” efficiency are used.

HEAT TRANSFER AND REQUIRED HYDRAULIC POWER

The thermophysical properties used in this work are obtained from the EES software (Klein et al., 2015). For concentration of 30% by weight (m/m), this tool gives the following freezing points: -13 °C for propylene glycol, -20 °C for ethanol, and -27 °C for methanol. The viscosity of antifreeze mixtures is the property having the most notable effect on the energy consumption of geothermal systems. High fluid viscosities lead to low Reynolds numbers and laminar flows in the borehole which tend to decrease the heat transfer coefficients inside borehole pipes. Figure 3a (left) shows the steady-state convective thermal resistance, $R_{conv}$, and steady-state thermal resistance, $R_b$, of a typical borehole using propylene glycol (30% m/m) and different flow rates (characteristics are given in Table 1). $R_b$ is the effective borehole thermal resistance between the fluid and the ground and is the sum of the grout, pipe and convective resistances (Eq. 1). The well-known relations from Hansen and Gnielinski/Petukhov are used to evaluate the convective heat transfer coefficient in borehole pipes for laminar and turbulent flows.
As shown in Figure 3a, \( R_{\text{conv}} \) increases significantly when the flow becomes laminar at a flow rate smaller than 0.32 L/s (for propylene glycol 30% m/m at 0 °C). In turn, this increases the borehole thermal resistance, \( R_b \), from 0.12 mK/W to 0.20 mK/W. Thus, for a given ground load, the fluid temperature for the laminar case must be lower than for the turbulent case in order to increase the temperature difference between the fluid and the ground. However, as shown by the \( P_{\text{hyd}} \) curve on Figure 3a, the required hydraulic power for laminar flow is significantly less than for turbulent flow. The transition to laminar flow also depends on the fluid temperature and pipe diameter. This is shown in Figure 3b where the dip in each curve represents the transition to laminar flow. The linear head loss is about three to four times higher for a 9 gpm flow rate in a 1.25” pipe than for a 4.5 gpm in a 1.25” pipe.

![Figure 3a and b](image)

Table 1. Main Characteristics of the Borehole used in this Study

<table>
<thead>
<tr>
<th>Parameter</th>
<th>S.I. Value</th>
<th>S.I. Unit</th>
<th>I.P. Value</th>
<th>I.P. Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>Depth</td>
<td>150</td>
<td>m</td>
<td>492</td>
<td>ft</td>
</tr>
<tr>
<td>Borehole diameter</td>
<td>0.15</td>
<td>m</td>
<td>6</td>
<td>in</td>
</tr>
<tr>
<td>Inside pipe radius</td>
<td>0.013</td>
<td>m</td>
<td>0.51</td>
<td>in</td>
</tr>
<tr>
<td>Outside pipe radius</td>
<td>0.016</td>
<td>m</td>
<td>0.63</td>
<td>in</td>
</tr>
<tr>
<td>Grout conductivity</td>
<td>1.5</td>
<td>W/m.K</td>
<td>0.87</td>
<td>BTU/hr.ft.°F</td>
</tr>
<tr>
<td>Grout thermal capacity</td>
<td>3000</td>
<td>kJ/m³.K</td>
<td>44.7</td>
<td>BTU/ft³.°F</td>
</tr>
<tr>
<td>Ground thermal conductivity</td>
<td>2.2</td>
<td>W/m.K</td>
<td>1.27</td>
<td>BTU/hr.ft.°F</td>
</tr>
<tr>
<td>Ground thermal diffusivity</td>
<td>0.096</td>
<td>m²/day</td>
<td>0.94</td>
<td>ft²/day</td>
</tr>
<tr>
<td>Pipe conductivity</td>
<td>0.42</td>
<td>W/m.K</td>
<td>0.24</td>
<td>BTU/hr.ft.°F</td>
</tr>
<tr>
<td>Flow rate</td>
<td>0.28/0.57</td>
<td>L/s</td>
<td>4.5/9</td>
<td>gpm</td>
</tr>
<tr>
<td>Borehole resistance ( R_b ) at 0 °C (PG30%)</td>
<td>0.202/0.123</td>
<td>m.K/W</td>
<td>0.350/0.213</td>
<td>hr.ft.°F/BTU</td>
</tr>
</tbody>
</table>

High fluid viscosities also increase pumping power due to increased pipe friction. Head losses \( \Delta p_{\text{f,pg}} \), \( \Delta p_{\text{f,bo}} \), and \( \Delta p_{\text{f,yp}} \) increase as viscosity affects the friction coefficient. With higher head losses, pumping power \( W_{\text{pump}} \) is increased and a larger pump must be used. Increasing flow rate to maintain turbulent flow also increases pumping power. In this work, the Darcy-Weisbach equation is used for pipe head losses and the Churchill equation is used for the friction factor (Eq. 2) as it is suitable for laminar, transient and turbulent flows. A Power Parameter, \( PP \), is also proposed to combine flow rate and pipe diameter (Eq. 3) to allow simple pumping power predictions.

\[
f = 8 \left( \frac{8}{\text{Re}} \right)^{12} + \left( 2.457 \ln \left( \frac{7}{\text{Re}} + 0.27 \frac{e}{D} \right) + \frac{37530}{\text{Re}} \right)^{1/12} \tag{2}
\]
\[ PP = \frac{Flow^3}{D_{pipe}^5} \quad \text{(with Flow in gpm and } D_{pipe} \text{ in inches)} \quad (3) \]

Figure 4a shows the required linear hydraulic power for PG30% at 0 °C as a function of PP. Dots represent detailed calculations while the straight line represents a linear regression through the data. The same exercise is performed for different antifreeze solutions with flow rates varying from 2 to 12 gpm and pipe diameters varying from 0.75 to 1.5”. Linear regressions are then obtained for each fluid (Figure 4b). Each regression presents an absolute RMSE under 0.015 W/m and allows an adequate first estimate of pumping power. The well-known criteria of 3 ft/100 ft of pipe (29 kPa/100 m) for head loss is also overlaid for 4.5 and 9 gpm flow rates (Kavanaugh and Rafferty, 2015). Figure 4b also shows that for a specific flow rate and pipe diameter, pumping power is higher as fluid viscosity increases. Pumping PG30% at 0 °C requires approximately 45% more power than for water.

The use of Figure 4b is best illustrated with an example. A flow of 9 gpm of ethanol (30% m/m) at 0 °C through a 1 meter pipe with a diameter of 0.032 m (1.25”) leads to a value of \( PP = 239 \text{ gpm}^3/\text{in}^5 \) with a corresponding value of 0.18 W of hydraulic power (0.19 W/m is obtained with a detailed calculation). If the system consists of a 100 m borehole with 10 m of connecting pipes to the heat pump (thus a total pipe length of 220 m) then 39.6 W of hydraulic power is required. Pumping this fluid with a 50% efficient circulator would then require 79 W of electrical power, \( W_{Pump} \).

Head losses through the source heat exchanger of the heat pump as well as in connecting hoses and valves are also important. Heat pump heat exchanger head loss is typically given by manufacturers and is function of flow rate and inlet fluid temperature. Equation 4, where \( \Delta p_{HP} \) is in kPa, \( Flow \) in L/s and \( T_{inHP} \) in °C, is a regression based on a heat pump performance map presenting head loss for several flow and temperature combinations (ClimateMaster, 2012). It is valid for flow ranging from 0.284 to 0.568 L/s (4.5 to 9 gpm) and for inlet temperatures ranging from -1.1 to 48.9 °C (30 to 120 °F). Valve and hose head losses, combined in one term, \( \Delta p_{Valve} \), are calculated using flow coefficients used by Kavanaugh and Rafferty (2015): \( C_v \) equals 25 for the valve and 8 for hoses (based on flows in gpm and a 1 psi (6.9 kPa) pressure drop). For 4.5 and 9 gpm flow rates, pressure drops then vary from 0.2 to 0.9 kPa (0.07 to 0.3 ft) for the valve and from 2.2 to 8.7 kPa (0.7 to 2.9 ft) for connecting hoses.

\[
\Delta p_{HP} = 88.0 \times Flow - 0.179 \times T_{inHP} - 13.3 \quad (4)
\]

An antifreeze correction factor for these head losses, \( f_{WPD} \), is proposed (Eq. 6). This factor, based on Blasius’ equation (Eq. 5), corrects manufacturers’ pressure drops, which are based on water. Blasius’ equation is valid for low turbulent Reynolds number (White, 2009), which is the case in typical small diameter hoses and heat exchangers. The value of \( f_{WPD} \) (Eq. 6) is then the ratio of the pressure drop for the antifreeze solution over the one for water. This factor
evaluated for different antifreezes and concentrations at 0 °C was verified against manufacturer’s data (ClimateMaster, 2012). As shown in Table 2, the method proposed here (Eq. 5 and 6) is within 2% of the manufacturer’s data for PG. Similar results were obtained for other fluids. ∆p_HPV and ∆p_valve (valve and hoses), calculated for water, are then multiplied by this factor to obtain the actual antifreeze pressure drop, ∆p_a.

\[ \Delta p = 0.158 \rho L \mu^{0.25} D^{-1.25} V^{1/4} \]  
(5)

\[ f_{WPD} = \frac{\Delta p_a}{\Delta p_w} = \left( \frac{\rho_a}{\rho_w} \right)^{0.75} \left( \frac{\mu_a}{\mu_w} \right)^{0.25} \]  
(6)

<table>
<thead>
<tr>
<th>Prop. Glycol (% m/m)</th>
<th>( f_{WPD} )</th>
<th>Eq. 6</th>
<th>Manufacturer</th>
<th>( f_{cap,heat} )</th>
<th>Eq. 9</th>
<th>Manufacturer</th>
<th>( f_{power,heat} )</th>
<th>Eq. 9</th>
<th>Manufacturer</th>
</tr>
</thead>
<tbody>
<tr>
<td>5%</td>
<td>1.07</td>
<td>1.07</td>
<td>0.991</td>
<td>0.989</td>
<td>0.998</td>
<td>0.997</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>15%</td>
<td>1.20</td>
<td>1.21</td>
<td>0.972</td>
<td>0.968</td>
<td>0.993</td>
<td>0.990</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>25%</td>
<td>1.37</td>
<td>1.36</td>
<td>0.945</td>
<td>0.947</td>
<td>0.987</td>
<td>0.983</td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

HEAT PUMP CAPACITY AND POWER

The use of antifreeze mixtures also affects heat transfer in the source-side heat exchanger of the heat pump, which has a detrimental effect on heat pump performance. Heat pump heating capacity \( Q_{Cap} \) decreases because of reduced convection in the heat exchanger pipes. Viscosity, density, thermal capacity and conductivity variations reduce capacity, by up to 10% for a 30% ethanol solution (ClimateMaster, 2012). Power consumption \( W_{HP} \) is also affected and it can decrease by up to 3% as capacity decreases. However, heat pump energy consumption increases with the use of antifreeze as it must operate for longer periods to meet the load as capacity is reduced.

Capacity and power consumption can be adjusted by correction factors to account for those effects. A capacity correction factor derived from Nguyen’s work is used here (2010). This factor, \( f_{capacity} \) (Eq. 9) predicts heat pump capacity variation based on heat pump constants (\( C_1 \) and \( C_2 \)) and Spitler and Jin’s (2003) Decrement Factor (Eq. 7). As shown by Eq. 8, the two constants are used to evaluate the total heat pump source-side thermal resistance \( R_t \). \( C_1 \) stands for the fluid convective resistance initially calculated for water. It is divided by \( DF \) to correct the fluid-side convective heat transfer coefficient. \( C_2 \) combines the pipe conductive, the refrigerant convective and the fouling thermal resistances, which are not affected by the choice of the antifreeze solution. \( C_1 \) and \( C_2 \) are back calculated using a pair of capacity correction factors from a manufacturer (ClimateMaster, 2012) with \( DF \) calculated for each fluid and each concentration. This leads to a two equations/two unknowns system that can be solved easily. \( C_1 \) and \( C_2 \) equal 1.0 and 16.35, respectively, when calculated using propylene glycol. \( C_1 \) and \( C_2 \) differ slightly when calculated for ethanol and methanol. Fluid specific values of \( C_1 \) and \( C_2 \) are used to predict \( f_{capacity} \) for the simulations performed in this paper.

\[ DF = \frac{h_a}{h_w} = \left( \frac{\mu_a}{\mu_w} \right)^{-0.47} \left( \frac{\rho_a}{\rho_w} \right)^{0.8} \left( \frac{C_{p,a}}{C_{p,w}} \right)^{0.33} \left( \frac{k_a}{k_w} \right)^{0.67} \]  
(7)

\[ R_t = R_{\text{conv,fluid}} + R_{\text{cond,pipe}} + R_{\text{conv,refrigerant}} + R_{\text{fouling}} = R_{\text{conv,fluid}} + \Sigma R = \frac{C_1}{DF} + C_2 \]  
(8)

\[ f_{capacity} = \frac{R_{t,a}}{R_{t,w}} = \frac{C_1 + C_2}{\frac{C_1}{DF} + C_2} \]  
(9)
It also appears that the use of this capacity correction factor proves to be accurate to correct the heat pump power consumption ($f_{\text{capacity}}$ is replaced by $f_{\text{power}}$ in Eq. 9). The same $DF$ is used but $C_1$ and $C_2$ are based on the manufacturer's power correction factors. They equal 1.0 and 75.68, respectively, when calculated using propylene glycol. As shown in Table 2, the predicted capacity and power correction factors are in good agreement with manufacturer's data.

Finally, there are operational benefits resulting from the use of antifreeze mixtures. Indeed, allowing colder fluid temperatures to the heat pumps (-3 °C instead of 0 °C for example) leads to shorter boreholes as the temperature difference between the borehole fluid and the ground is larger. However, this effect might be counterbalanced by increased pumping energy and reduced borehole and heat pump performances. It is thus important to evaluate the overall effects of using a specific antifreeze solution. This is done using annual simulations as shown in the next section.

**ANNUAL SIMULATIONS**

Annual simulations using TRNSYS v17 (Klein et al., 2010) are performed to compare the energy consumption of a GCHP system using different antifreeze mixtures (PG, EA and MA) and two flow rates (4.5 and 9 gpm or 1.5 and 3 gpm/ton). A 1-minute time step is used, which is small enough to capture borehole transient effects. The system under study is the one presented in Figure 1. It consists of a 3-ton (10.5 kW) water-to-air ground-coupled heat pump linked to a 150 m borehole. The GCHP system provides space heating for a single-family house with enough capacity to avoid the need for an auxiliary heater. The building is simulated in a heating dominated climate (Montreal, Canada) with an annual heating requirement of about 20 000 kWh. Typical models found in TRNSYS for the building, thermostat, heat pump and pump are used. Heat pump performance is modeled based on a manufacturer’s steady-state performance data (ClimateMaster, 2012). The TRCM borehole model from Godefroy and Bernier (2014) is used to model the single U-tube borehole. This experimentally validated model (Godefroy et al., 2016) accounts for fluid and grout thermal capacity, which indirectly affects heat pump performance (Gagné-Boisvert and Bernier, 2016). The main characteristics of the borehole are given in Table 1. The pump is a high efficiency wet rotor circulator. When operating, it is assumed that it provides a constant volumetric flow rate year-round. Pump efficiency is evaluated based on the “High” regression curve presented in Figure 2. Thermophysical properties, correction factors, head losses, and borehole thermal resistance are calculated at each time step.

**Comparison results**

Tables 3 and 4 present one-year simulation results while Figure 5 shows the pump power and heat pump inlet temperature evolution over the year. In these tables, pump fraction is the ratio of the pumping energy over the total energy consumption; seasonal performance factor (SPF) is the ratio of the annual heating requirement over the total energy consumption (Nordman and Zottl, 2011); Head$_{\text{max}}$ is the highest total head loss over the year while pump power is the highest required pump power. As illustrated in Figure 5, pumping power decreases by about 10-15% with higher fluid temperature occurring in summer. $R_0$ also varies over the year as fluid temperature fluctuates, as shown by minimum and maximum values in Tables 3 and 4. These tables show the importance of considering the antifreeze properties and how they influence system operation.

**Figure 5a and b** Pump power and heat pump inlet temperature for PG30% at 4.5 (left) and 9 gpm (right) over one year.
Based on that, a common GCHP system using propylene glycol at 30\% would require less energy if working with 4.5 gpm versus 9 gpm, even if the borehole flow is almost always laminar. This confirms Kavanaugh’s recommendation. Thus, designers must be aware that the heat transfer advantage of turbulent flows is small compared to the increased pumping energy consumption in GCHP systems. Figures 6a and 6b present a breakdown of peak head losses to help understand flow rate effects on results. Cases with 4.5 and 9 gpm using propylene glycol 30\% are presented. As expected, the various head losses are about four times higher when the flow rates are higher. Propylene glycol 30\% has the worst energy performance and a bigger and more expensive pump would be required.

### Table 3. Simulation results for 4.5 gpm flow rate

<table>
<thead>
<tr>
<th>Fluid (%)</th>
<th>Total (kWh)</th>
<th>Pump (kWh)</th>
<th>Pump Fraction (%)</th>
<th>SPF</th>
<th>Head(_{\text{max}}) (kPa/ft)</th>
<th>Pump power (W)</th>
<th>(R_b), min/max (W/mK)</th>
<th>(T_{\text{inHP},\text{min}}) (°C/°F)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Prop. Glycol 15%</td>
<td>6 027</td>
<td>126</td>
<td>2.10%</td>
<td>3.36</td>
<td>93.8/31.4</td>
<td>56.9</td>
<td>0.122/0.123</td>
<td>0.4/33</td>
</tr>
<tr>
<td>Prop. Glycol 30%</td>
<td>6 286</td>
<td>125</td>
<td>2.00%</td>
<td>3.22</td>
<td>99.5/33.3</td>
<td>60.0</td>
<td>0.127/0.204</td>
<td>-1.7/29</td>
</tr>
<tr>
<td>Prop. Glycol 40%</td>
<td>6 545</td>
<td>187</td>
<td>2.86%</td>
<td>3.09</td>
<td>155.2/51.9</td>
<td>88.9</td>
<td>0.212/0.213</td>
<td>-1.4/29</td>
</tr>
<tr>
<td>Ethanol 15%</td>
<td>6 114</td>
<td>130</td>
<td>2.13%</td>
<td>3.31</td>
<td>92.5/30.9</td>
<td>56.2</td>
<td>0.123/0.125</td>
<td>0.5/33</td>
</tr>
<tr>
<td>Ethanol 30%</td>
<td>6 368</td>
<td>120</td>
<td>1.88%</td>
<td>3.18</td>
<td>90.5/30.3</td>
<td>55.1</td>
<td>0.127/0.209</td>
<td>-1.3/30</td>
</tr>
<tr>
<td>Methanol 15%</td>
<td>6 014</td>
<td>119</td>
<td>1.98%</td>
<td>3.36</td>
<td>87.3/29.2</td>
<td>53.4</td>
<td>0.122/0.123</td>
<td>0.5/33</td>
</tr>
<tr>
<td>Methanol 30%</td>
<td>6 113</td>
<td>127</td>
<td>2.07%</td>
<td>3.31</td>
<td>89.6/30.0</td>
<td>54.7</td>
<td>0.124/0.125</td>
<td>0.8/33</td>
</tr>
<tr>
<td>Water</td>
<td>5 878</td>
<td>102</td>
<td>1.74%</td>
<td>3.44</td>
<td>75.8/25.4</td>
<td>47.1</td>
<td>0.120/0.121</td>
<td>0.6/33</td>
</tr>
</tbody>
</table>

### Table 4. Simulation results for 9 gpm flow rate

<table>
<thead>
<tr>
<th>Fluid (%)</th>
<th>Total (kWh)</th>
<th>Pump (kWh)</th>
<th>Pump Fraction (%)</th>
<th>SPF</th>
<th>Head(_{\text{max}}) (kPa/ft)</th>
<th>Pump power (W)</th>
<th>(R_b), min/max (W/mK)</th>
<th>(T_{\text{inHP},\text{min}}) (°C/°F)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Prop. Glycol 15%</td>
<td>6 561</td>
<td>653</td>
<td>10.0%</td>
<td>3.08</td>
<td>302/101</td>
<td>296</td>
<td>0.120/0.121</td>
<td>-0.1/32</td>
</tr>
<tr>
<td>Prop. Glycol 30%</td>
<td>6 917</td>
<td>821</td>
<td>11.9%</td>
<td>2.93</td>
<td>374/125</td>
<td>358</td>
<td>0.121/0.123</td>
<td>0.1/32</td>
</tr>
<tr>
<td>Ethanol 15%</td>
<td>6 670</td>
<td>683</td>
<td>10.2%</td>
<td>3.03</td>
<td>313/105</td>
<td>305</td>
<td>0.120/0.121</td>
<td>0.1/32</td>
</tr>
<tr>
<td>Ethanol 30%</td>
<td>6 966</td>
<td>786</td>
<td>11.3%</td>
<td>2.91</td>
<td>347/116</td>
<td>334</td>
<td>0.121/0.123</td>
<td>0.4/33</td>
</tr>
<tr>
<td>Methanol 15%</td>
<td>6 519</td>
<td>616</td>
<td>9.4%</td>
<td>3.10</td>
<td>282/94.3</td>
<td>278</td>
<td>0.119/0.120</td>
<td>0.0/32</td>
</tr>
<tr>
<td>Methanol 30%</td>
<td>6 647</td>
<td>654</td>
<td>9.8%</td>
<td>3.04</td>
<td>292/97.7</td>
<td>287</td>
<td>0.120/0.121</td>
<td>0.3/33</td>
</tr>
<tr>
<td>Water</td>
<td>6 324</td>
<td>539</td>
<td>8.5%</td>
<td>3.20</td>
<td>250/83.6</td>
<td>250</td>
<td>0.119/0.119</td>
<td>0.0/32</td>
</tr>
<tr>
<td>PG 30% Parallel</td>
<td>6 101</td>
<td>265</td>
<td>4.3%</td>
<td>3.32</td>
<td>107.5/36.0</td>
<td>119</td>
<td>0.126/0.204</td>
<td>1.9/34</td>
</tr>
<tr>
<td>MA 15% Parallel</td>
<td>5 865</td>
<td>214</td>
<td>3.7%</td>
<td>3.45</td>
<td>86.5/28.9</td>
<td>97.8</td>
<td>0.122/0.123</td>
<td>2.2/36</td>
</tr>
<tr>
<td>Water Parallel</td>
<td>5 729</td>
<td>187</td>
<td>3.3%</td>
<td>3.53</td>
<td>76.7/25.7</td>
<td>88.0</td>
<td>0.120/0.121</td>
<td>2.3/36</td>
</tr>
</tbody>
</table>

Results show that higher concentrations and higher flow rates increase total energy consumption of the GCHP system. Using methanol with a concentration of 15\% and 4.5 gpm gives the best energy performances with an energy consumption of 6014 kWh, a \textit{SPF} of 3.36 and a maximum pump power of 53.4 W. It is worth mentioning that propylene glycol at 15\% with 4.5 gpm is not recommended for that application considering its -5 °C freezing point. Ethanol 30\% with 9 gpm requires 6966 kWh (+16\%) and 334 W of pump power (+525\%) with a \textit{SPF} of 2.91 (-13\%). This case gives the worst energy performance and a bigger and more expensive pump would be required.

For pumping, flow rates of 4.5 and 9 gpm lead to pump fractions of about 2\% and 10\%, respectively. Based on an average 8 kW peak heating load, a pumping grade of A on the Kavanaugh and Rafferty (2015) scale is achieved for all 4.5 gpm cases. A grade of F is obtained for 9 gpm cases. Higher flow rates and concentrations are responsible for bigger head losses, leading to higher pumping power and energy consumption. It appears that systems with head losses under 100 kPa (4.5 gpm) lead to better overall energy performances than ~300 kPa (9 gpm) cases, which is in line with Mescher’s statement presented earlier. Interestingly, for propylene glycol 30\% and ethanol 30\% with 4.5 gpm, pumping energy is lower than for cases with a 15\% concentration, which is counterintuitive as these solutions are more viscous. This is because those two cases are in laminar flow around 90\% of the operation time, which leads to lower head losses (Figure 3b). The PG40\% at 4.5 gpm case always operates in laminar flow and presents higher head losses than the PG30\% case. Its minimum heat pump inlet temperature is also slightly higher because more energy comes from the heat pump compressor and less from the ground.

It is important to note that these three 4.5 gpm laminar cases, with typical laminar \(R_b\) values and lower loop temperatures, yield lower energy consumption (-10\%) than the same cases with the fully turbulent 9 gpm flow rate. This is mainly due to the higher pumping requirements. Based on that, a common GCHP system using propylene glycol at 30\% would require less energy if working with 4.5 gpm versus 9 gpm, even if the borehole flow is almost always laminar. This confirms Kavanaugh’s recommendation. Thus, designers must be aware that the heat transfer advantage of turbulent flows is small compared to the increased pumping energy consumption in GCHP systems. Figures 6a and 6b present a breakdown of peak head losses to help understand flow rate effects on results. Cases with 4.5 and 9 gpm using propylene glycol 30\% are presented. As expected, the various head losses are about four times higher when the flow...
rate is doubled.

**Figure 6a, b and c** Breakdown of peak head losses for PG30% at 4.5 gpm (left), 9 gpm (center) and in parallel (right).

The same simulations are performed without the correction factors (\(f_{\text{capacity}}, f_{\text{power}}\) and \(f_{\text{WPD}}\)) to assess their importance. In all cases, neglecting correction factors underestimates energy consumption. The extent of the underestimation is higher for higher antifreeze concentrations. Neglecting the capacity and power correction factors underestimates heat pump energy consumption by 2 to 6%, pump consumption by 3 to 9% and overall energy consumption by 2 to 7%. Neglecting the head loss correction factor (heat pump, hoses and valve) underestimates pump energy consumption by 3 to 8% and overall energy consumption by 0.1 to 0.7% as pumping represents 2 to 10% of the total. Neglecting both factors underestimates pump energy consumption by 6 to 15% and overall energy consumption by 2 to 7%. Table 5 presents these differences for propylene glycol 15 and 30% at 9 gpm.

<table>
<thead>
<tr>
<th>Fluid % m/m</th>
<th>Pump Without</th>
<th>Total Without</th>
<th>Pump Without</th>
<th>Total Without</th>
</tr>
</thead>
<tbody>
<tr>
<td>PG15% (kWh)</td>
<td>653</td>
<td>635</td>
<td>615</td>
<td>6561</td>
</tr>
<tr>
<td>Difference (%)</td>
<td>-2.8%</td>
<td>-3.2%</td>
<td>-5.9%</td>
<td>-2.0%</td>
</tr>
<tr>
<td>PG30% (kWh)</td>
<td>821</td>
<td>762</td>
<td>717</td>
<td>6917</td>
</tr>
<tr>
<td>Difference (%)</td>
<td>-7.2%</td>
<td>-5.9%</td>
<td>-12.7%</td>
<td>-5.2%</td>
</tr>
</tbody>
</table>

As a final test, two 75 m boreholes are piped in parallel with a total flow rate of 9 gpm. Head losses are consequently modified (Figure 6c). As shown in Table 4, this leads to the lowest overall energy consumptions and highest SPF. Using MA15% requires 5865 kWh with a corresponding SPF of 3.45. Using shorter boreholes in parallel must then be considered as an effective design in terms of energy performance, even if laminar flow occurs in boreholes. Shorter parallel boreholes also lead to higher values of \(T_{\text{in HP}, \text{min}}\) which may decrease the required antifreeze concentration or boreholes length.

**CONCLUSION**

The main objective of this paper is to compare the energy consumption of a GCHP system using various antifreeze solutions. It reviews the effects of antifreeze on head losses, borehole thermal resistance and heat pump operation. It also proposes efficiency curves for currently available circulators; a graph estimating required hydraulic power for different flows and pipe diameters; and antifreeze correction factors to correct heat pump capacity, power and head loss. Annual energy simulations are then performed on a residential GCHP system. Results show, as expected, that higher concentrations and higher flow rates increase total energy consumption. Methanol with 15% concentration and a flow rate of 1.5 gpm/ton gives the lowest annual energy consumption. Ethanol at 30% and 3 gpm/ton is the worst choice, requiring 16% more energy and 525% more pumping power compared to the methanol case. Laminar flow in boreholes appears to be favorable when compared to turbulent flow, which lead to relatively high pumping energy consumption. Finally, placing shorter boreholes in parallel appears to decrease energy consumption and increase the seasonal performance factors.
ACKNOWLEDGMENTS

The authors would like to express their sincere gratitude to ASHRAE, Hydro-Quebec and the Natural Sciences and Engineering Research Council of Canada (NSERC) who provided scholarships to the first author. This work was also performed with funds provided by NSERC’s Smart Net-Zero Energy Buildings Strategic Research Network.

NOMENCLATURE

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>$C$</td>
<td>Antifreeze concentration (%)</td>
</tr>
<tr>
<td>$C_a$, $C_w$</td>
<td>Heat pump constants</td>
</tr>
<tr>
<td>$C_r$</td>
<td>Flow coefficient (gpm)</td>
</tr>
<tr>
<td>$C_p$</td>
<td>Thermal capacity (kJ/kg.K)</td>
</tr>
<tr>
<td>$D$</td>
<td>Pipe diameter (m)</td>
</tr>
<tr>
<td>$f$</td>
<td>Friction factor (-)</td>
</tr>
<tr>
<td>$h$</td>
<td>Convection coefficient (W/m².K)</td>
</tr>
<tr>
<td>$k$</td>
<td>Thermal conductivity (W/m.K)</td>
</tr>
<tr>
<td>$D_p$</td>
<td>Pressure drop/Head (kPa)</td>
</tr>
<tr>
<td>$\rho$</td>
<td>Density (kg/m³)</td>
</tr>
<tr>
<td>$\mu$</td>
<td>Dynamic viscosity (kg/m.s)</td>
</tr>
<tr>
<td>$\varphi$</td>
<td>Power Parameter (gpm³/in5)</td>
</tr>
<tr>
<td>$\varphi'$</td>
<td>Fluid speed (m/s)</td>
</tr>
<tr>
<td>$\varphi''$</td>
<td>Power (W)</td>
</tr>
<tr>
<td>$\varphi_0$</td>
<td>Mass Conc.</td>
</tr>
</tbody>
</table>

Subscripts

- $a$: Antifreeze mixture
- $cap$: Capacity
- $conv$: Convection
- $HP$: Heat pump
- $min$: Minimum
- $t$: Total
- $w$: Water
- %: Mass Conc.

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Storage of Solar Thermal Energy in Borehole Thermal Energy Storage Systems

John S. McCartney  Tuğçe Başer  Ni Zhan
Ning Lu  Shemin Ge  Kathleen Smits

ABSTRACT
This study focuses on the evaluation of solar thermal energy storage in a medium-scale soil-borehole thermal energy storage (SBTES) system installed in San Diego, CA. The SBTES system consists of an array of thirteen 15 m-deep, closely-spaced borehole heat exchangers installed in conglomerate bedrock. The entire site is above the water table, with relatively dry subsurface conditions. Instrumentation was included into the array to monitor temperature distributions with depth and radial spacing within the array, as well as water content fluctuations near the ground surface. A total of eight evacuated tube solar thermal panels with an absorber area of 4.16 m² were connected in series to supply heat to a temporary heat storage tank. Results from a 4-month transient heat injection period into the SBTES system are presented in this paper. These include data on the characterization of the thermal properties of the SBTES system, the transient heat flux collected from the solar thermal panels, the corresponding transient heat flux into the subsurface, and the changes in ground temperature.

INTRODUCTION
Soil-borehole thermal energy storage (SBTES) systems are used for storing heat collected from renewable sources in the subsurface that it can be used later for space or water heating. Heat sources such as solar thermal panels generate heat during the day with a greater energy generation during summer months, so SBTES systems permit storage of the abundant and free thermal resource (Sibbitt et al. 2007, McCartney et al. 2013). SBTES systems function in a similar way to geothermal heat exchangers, where a carrier fluid is circulated through a closed-loop pipe network installed in vertical boreholes. Different from conventional geothermal heat exchange systems, the boreholes are spaced relatively close together (1-2 m) to concentrate heat. SBTES systems are a convenient alternative to other energy storage systems as they are relatively inexpensive, involve storage of renewable energy (solar thermal energy), and are space efficient as they are underground (Başer and McCartney 2015).

Since the concept of borehole thermal energy storage systems was introduced by Claesson and Hellström (1981), several SBTES systems have been installed in Canada and Europe that benefit from district-scale heat distribution systems. The Drake Landing SBTES system in Canada provides 95% of the heat demand of 52 homes (Sibbitt et al. 2007), despite only having an efficiency of heat extraction over heat injection of less than 25% (Zhang et
al. 2012). This system supplies heat from solar thermal panels installed on garage roofs to an array of 144 boreholes in a 35 m-deep, 35-m wide grid. Another successful SBTES system was installed in 2007 in Braedstrup, Denmark (Bjoern 2013). This system supplies heat from 18,000 m² of solar thermal panels to an array of 50 boreholes having a depth of 47-50 m installed across a 15 m-wide area. This system provides 20% of the heat to 14000 homes. The most recent SBTES was installed in 2008 in Crailsheim, Germany involving of a series of 55 m-deep boreholes to form a 39000 m³ subsurface storage volume. This system stores heat from 7410 m² of flat plate solar thermal collectors to provide heat for a school and 230 dwellings (Nussbicker-Lux 2012).

Despite the successful use of SBTES systems in practice, there are still opportunities for geotechnical engineers to help understand and improve the efficiency considering the hydrogeological setting of these systems. This is the focus of a collaborative NSF project between the University of California San Diego (UCSD), University of Colorado Boulder, and Colorado School of Mines. In particular, this study seeks to understand the benefits of installing SBTES systems in the vadose zone, the layer of partially-saturated soil or rock near the ground surface extending up to depths on the order of 10 of meters in some locations. The unsaturated porous material in the vadose zone has a lower apparent thermal conductivity that limits the loss of heat from the subsurface heat storage system, yet still permits large amounts of heat to be stored due to the combination of a reasonably high volumetric heat capacity and a large storage volume. In addition to coupling between thermal and hydraulic properties of unsaturated soils the modes of heat transfer in unsaturated soils may include a combination of conduction and convection due to the flow of pore water in liquid and vapor forms under thermal gradients.

This project focuses on the development of new coupled constitutive equations for the thermo-hydraulic properties of soils (Lu and Dong 2015) as well as numerical efforts to understand the impacts of borehole array geometry, ground properties, heat injection magnitudes, and heat injection duration on the heat storage (Başer et al. 2015), characterize the behavior of borehole system behavior (Başer et al. 2016a), understand the impacts of different modes of heat transfer (Başer et al. 2016b, 2016c), and understand the role of incorporating a thermal insulation layer (Başer et al. 2016d). However, data from a field-scale installation is still necessary to understand the variable boundary conditions for heat injection, to provide a baseline data set for calibration of numerical simulations, and to assess different injection schemes to improve the efficiency of heat storage. These are the objectives of a new field installation at UCSD described in this update, which has recently gone into operation.

**SBTES SYSTEM AT UCSD**

**Site Description**

The full-scale SBTES system was installed at the Englekirk Structural Engineering Center (ESEC) on the UCSD campus. The subsurface at the site consists of 1 m of silty sand underlain by conglomerate bedrock, with a groundwater table more than 30 m deep. The thermal conductivity of the unsaturated colluvium measured using the TRIM method (Lu and Dong 2015) and was found to be 0.5 W/mK for the average degree of saturation with depth. The conglomerate has a relatively high hydraulic conductivity of approximately 10-5 m/s, indicating that infiltration and thermally induced water flow may affect heat transfer processes. The SBTES system includes a network of 25 mm-diameter high density polyethylene (HDPE) tubing installed within 15 boreholes in a hexagonal array with a spacing of 1.5 m as shown in Figure 1. Although the number of boreholes is much smaller than those incorporated into the district-scale heating systems mentioned above, it provides an important data point in the scaling of SBTES systems for different sizes of communities.
Three heat exchange loops (A, B, C) were installed in the array so that heat could be injected first into the center of the array then split into three directions to ensure uniform distribution of heat to the different parts of the array (as opposed to connecting all of the heat exchangers in series) and to lead to a radial stratification of heat from the center. Thermistor strings having 6 thermistors equally spaced along a single cable were installed in four boreholes within the array (labeled 1 to 4) as well as in a reference borehole approximately 9 m outside of the array. These sensors permit evaluation of the transient heat transfer within the array as well as the undisturbed ground temperature fluctuations with depth outside of the array over time. Other instrumentation include dielectric sensors in the soil near the ground surface to monitor infiltration and evaporation, thermocouples to monitor the temperature of the carrier fluid entering and exiting the boreholes, flow meters to measure the fluid flow in each loop, and a temperature-humidity sensor to monitor ambient air conditions.

**Construction Process**

The slurry method was used to drill the 100 mm-diameter boreholes as shown in Figure 2(a). Care was taken during drilling due to the relatively small borehole spacing of 1.5 m. Heat exchange tubing and thermistor cables were inserted into the open boreholes and a mixture of sand-bentonite grout placed with a tremie into the boreholes. The soil was then excavated in a hexagonal shape around the array to a depth of 1 m, as shown in Figure 2(b). A 2 mm-thick hydraulic barrier was placed over the array after placement of a thin layer of site soil for leveling, as shown in Figure 2(c). To prevent heat loss to the atmosphere a layer of expanded polystyrene (EPS) insulation was placed on top of the array as shown in Figure 2(d) and the excavation was backfilled with compacted site soil.

Eight evacuated tube solar thermal collectors having a total area of 33 m² were used to collect heat from the sun, and are shown in Figure 3(a). A 2400 liter water-filled tank is used to store heat from the solar panels on a short-term basis to extend the time for injection of the heat collected from the solar thermal panels into the subsurface. The tank included two copper heat exchangers, one connected to the solar thermal connectors and the other connected to the borehole array. The heat transfer through the various components of the system is controlled using the manifold and solar pump station shown in Figure 3(b).
Figure 2 Pictures of the SBTES installation at UCSD: (a) Drilling; (b) Borehole heat exchangers connected after excavation of 1 m of surface soil; (c) Hydraulic barrier; (d) Thermal insulation layer

Figure 3 SBTES system components at the ground surface: (a) Solar thermal panels; (b) Geothermal manifold
PRELIMINARY DATA ANALYSIS

A preliminary operation stage of the system was started on April 29th, 2016, with the goal of characterizing the typical heat transfer from the various components as well as the thermal characterization of the subsurface. The operations are still preliminary because a control system has not yet been implemented to ensure that heat is only collected from the solar thermal panels during the day. Specifically, the circulation pumps in the geothermal and solar loops are operated continuously. Accordingly, fluid is still circulated through the solar panels at night, which may result in a slight loss in heat from the system. Nonetheless, because the nights in San Diego are very mild a significant amount of heat loss has not been detected. Further, the optimal configuration of the solar thermal panels is still being investigated. The heat transfer in the different components of the system can be calculated from the entering and exiting carrier fluid temperatures, the carrier fluid flow rate, and the carrier fluid (water) properties, as follows:

\[
\dot{Q} = \dot{V}_f \rho_f C_f (T_{in} - T_{out})
\]

where \(\dot{V}_f\) is the measured volumetric flow rate of the heat exchanger fluid (water), \(\rho_f\) is the density of water (1000 kgm\(^{-3}\)), \(C_f\) is the specific heat capacity of water (4183 Jkg\(^{-1}\)K\(^{-1}\)), and \(T_{in}\) and \(T_{out}\) are the measured temperatures of the water entering and exiting the heat exchanger loops, respectively. The heat transfer rate collected from the solar thermal panels and injected into the borehole array is shown in Figure 4(a). The heat transfer rate from the solar thermal panels fluctuates in a similar manner to the air temperature shown in Figure 4(b). The average heat transfer rate collected from the solar thermal panels is 5010 W, with a standard deviation of 5996 W. Much higher values than the average value were noted in the peak sunlight times in the afternoon, and the lowest value was still greater than zero. These peak values correspond to a peak fluid temperature of nearly 90 °C on most days, which is why solar thermal panels are used frequently for hot water generation in homes.

![Figure 4](image_url)  
*Figure 4*  
(a) Heat transfer from the solar thermal panels and into the borehole array; (b) Outside air temperature

The average heat transfer rate into the borehole array is 3182 W with a standard deviation of 2073 W. Due to the short term storage tank, the heat transfer rate into the borehole array exceeded that collected from the solar thermal panels for several hours. As there is 196 m of geothermal heat exchanger boreholes in the SBTES system, the average heat transfer rate per unit meter is 16 W/m, while the maximum value is 44 W/m. The typical heat transfer...
rate per unit meter observed in the district-scale SBTES systems mentioned above is 30 W/m, so the range of heat transfer rates encountered at this site is reasonable. The cumulative thermal energy collected and injected during this time period is shown in Figure 5(a). The efficiency of heat injection, defined as the heat injected over that collected, is shown in Figure 5(b) and has stabilized at approximately 0.7. This may be improved by adding a control system for the solar thermal panels so that they only circulate fluid during the day, reconfiguring the panels (connection in parallel instead of in series), and changing the fluid flow rates in the solar thermal panels and in the geothermal heat exchangers. The analyses of Başer and McCartney (2015) indicate that some of the heat injected into the array will be lost due to diffusion to the surrounding subsurface, so the amount of heat injected will not equal the heat stored. The heat loss is currently being evaluated using numerical simulations of this array.

![Figure 5](image_url)  
Figure 5  (a) Comparison of thermal energy collected from the solar thermal panels and injected in the borehole array; (b) Efficiency of heat injection

The analyses presented so far in Figures 4 and 5 are relevant to characterizing the boundary conditions for heat transfer into SBTES systems, and are important in the simulation of the SBTES systems using heat transfer software. Some of the preliminary ground temperatures at a depth of 9.29 m at different locations from the center of the array are shown in Figure 6. The temperatures at the locations of the heat exchangers increase nonlinearly with time, with daily fluctuations in temperature. The temperatures of the soil at the locations between the heat exchangers increase without daily fluctuations. The temperature of the soil in the reference borehole has actually slightly decreased at this depth over the time period of the evaluation. Overall, the duration of time required to heat the soil is slow, but this is similar to observations from Drake Landing, which required several years to become fully charged (Zhang et al. 2012).

The ground temperatures with depth shown in Figure 7(a) indicate that the ground temperature is slowly increasing above the ambient ground temperature shown in Figure 7(b). The higher temperatures near the surface are likely due to surface effects, while the lower temperatures at the base are due to downward heat loss. Although the average change in the temperature of the ground is only 5 °C, this represents a significant amount of thermal energy over the volume of the array. A portion of this energy will be lost to the surrounding subsurface, but as the size of the array is large, it represents a significant amount of energy.
Figure 6  Ground temperatures at different radial locations from the center of the array at a depth of 9.29 m.

Figure 7  Ground temperatures: (a) Borehole 2 within the array; (b) Reference borehole outside of the array.

CONCLUSION

This paper shows some of the preliminary data from a full-scale soil-borehole thermal energy storage (SBTES) system constructed at the University of California San Diego. Data on the solar thermal energy collectors and borehole heat exchangers indicates that 70% of the solar thermal energy collected has been injected into the subsurface. This number is lower than desired but can be improved by refining the details of the heat injection scheme. The ground temperatures over a 4 month period have increased by approximately 10 °C. This increase in temperature may appear to be small, but it represents a significant thermal energy magnitude of 30 GJ transferred into a large volume of soil. The results from this site will be further analyzed using numerical simulations to evaluate the heat losses from the array, as well as strategies to optimize the heat injection scheme to optimize the heat injection.
ACKNOWLEDGEMENTS

Funding from NSF 1230237 is appreciated. The opinions are those of the authors alone.

NOMENCLATURE

\[ \dot{Q} = \text{Heat transfer rate (W)} \]
\[ \dot{V} = \text{Volumetric flow rate of fluid} \]
\[ \rho = \text{Density (kgm}^{-3}\text{)} \]
\[ C = \text{Specific heat capacity (Jkg}^{-1}\text{K}^{-1}\text{)} \]
\[ T = \text{temperature of fluid (K)} \]

Subscripts

\[ f = \text{fluid} \]
\[ in = \text{inlet fluid} \]
\[ out = \text{outlet fluid} \]

REFERENCES


Properties of different ethyl alcohol based secondary fluids used for GSHP in Europe and USA

Monika Ignatowicz, Åke Melinder, Björn Palm

ABSTRACT

The extensive use of ground source heat pumps (GSHP) for heating and cooling purposes has made Sweden the European leader in geothermal energy utilization, in terms of the installed capacity, as well as extracted thermal energy. The commercially available ethyl alcohol based fluids in Sweden are distributed as 88 - 95 wt-% ethyl alcohol concentrate, including up to 12 wt-% of denaturing agents in form of propyl alcohol (8 to 10 wt-%) and n-butylic alcohol (2 wt-%). In Switzerland the commercial ethyl alcohol products contain 2 vol-% methyl ethyl ketone and 0.5 vol-% methyl isobutyl ketone, whereas in Finland commercial products contain up to 1.8 vol-% methyl ethyl ketone and 2.7 vol-% methyl isobutyl ketone. In North America the most common denaturing agents for ethyl alcohol based secondary fluid are methyl alcohol (methanol) (3.76 - 10 wt-%) and pine needle oil (up to 0.5 vol-%). The chemical character of these denaturing agents can in different ways affect the thermophysical properties. Thus, the aim of this paper was to investigate the thermophysical properties of different commercially available alcohol blends in Europe and United States. The results showed that the commercial product commonly used in Sweden (EA18 + PA1.6 + BA0.4) has the best thermophysical properties among different ethyl alcohol based products found in Europe when taking into consideration all thermophysical properties. Pure MA20 poses better thermophysical properties than EA18+MA2 and the lowest viscosity among all investigated alcohol blends. MA20 has as well good properties but special care needs to be taken due to high toxicity of methyl alcohol. Moreover, EA18+MA2 does not have good thermophysical properties compared to other ethyl alcohol blends and products containing small amounts of propyl and butyl alcohol or ketones are more recommended instead.

INTRODUCTION

Attnics et al., 2016 has reported that the total number of ground source heat pump (GSHP) installed in Europe exceeds 1.7 million units. Sweden among other European countries is the leader in geothermal energy utilization in terms of the installed capacity and extracted thermal energy (Gehlin et al., 2016). It is estimated that there are about 500 000 small and 500 large ground source heat pumps (Acuña et al., 2015). According to the statistics there are around two million single-family houses in Sweden and around 20 - 25 % of these houses are heated by GSHPs. The dominant type of GSHP systems are shallow low temperature systems ranging from 5 to 10 kW that provide about 23 TWh of heating and cooling. The total installed heating and cooling capacity in Sweden is estimated to be 6.8 GW. The typical Swedish setup consists of one or several vertical borehole heat exchangers (BHE) having a depth between 120 and 300 m (Gehlin et al., 2016). In many countries the space around the U-pipe in the borehole is filled with grout to prevent water and contaminants migration along the vertical borehole (Gustafsson et al., 2010). Swedish guidelines allow for ungrouted, groundwater filled BHE. Moreover, the market for larger shallow GSHP systems for both residential and non-residential buildings has been expanding over the last years (Gehlin et al., 2016). Ethyl alcohol based secondary fluid is one of the most common secondary fluids in Sweden for GSHP application recommended by the Geological Survey of Sweden (SGU) and the Swedish Environmental Protection Agency due to
relatively good thermophysical properties and low toxicity (SGU, 2008). Due to the flammability risks ethyl alcohol based secondary fluids are usually not exceeding 30 wt-\%, corresponding to the freezing point of \(-20.5\) °C. European Union regulations strictly define the types and concentrations of denaturing agents added to prevent from drinking ethyl alcohol. The most common type of denaturing agents in Europe are: propyl alcohol (2-propanol, isopropanol, PA), n-butyl alcohol (n-butanol, BA), methyl ethyl ketone (2-butane, MEK) and methyl isobutyl ketone (4-methylpentan-2-one, MIBK) (EUR-Lex, 2013). There are only two approved denaturing agents for ethyl alcohol based secondary fluid in Sweden: propyl alcohol and n-butyl alcohol due to their low toxicity compared to ketones. Both propyl and butyl alcohols occur in nature as the fermentation products and their biodegradation time is up to 28 days. The commercially available ethyl alcohol based secondary fluids in Sweden are normally distributed as 88 - 95 wt-\% ethyl alcohol concentrate, including up to 12 wt-\% of denaturing agents but no corrosion inhibitors. The most common type of ethyl alcohol product in Sweden contains 8 wt-\% propyl alcohol and 2 wt-\% n-butyl alcohol. Another less used product on the Swedish market contains 12 wt-\% of denaturing agents (10 wt-\% propyl alcohol and 2 wt-\% n-butyl alcohol) (Helachem, 2014). In other European countries, like Switzerland and Finland, commercial products containing a mixture of two ketones are used for GSHP application. In Switzerland the commercial ethyl alcohol products contain 2 vol-\% methyl ethyl ketone and 0.5 vol-\% methyl isobutyl ketone (Alcosuisse, 2014), whereas in Finland commercial products contain up to 1.8 vol-\% methyl ethyl ketone and 2.7 vol-\% methyl isobutyl ketone (Alta Plc. 2012). Instead, in North America the most common denaturing agents for ethyl alcohol based secondary fluid are methyl alcohol (methanol) (3.76 - 10 wt-\%) (Lyondell, 2003) and pine needle oil (up to 0.5 vol-\%) (Government of Canada, 2016).

The United States continues to lead worldwide in terms of installed geothermal power capacity. GSHPs are being installed at 8 % annual growth rate with 1.4 million units in operation having a typical size of 12 kW. Approximately 60 % of GSHP units are installed in residential buildings and the remaining 40 % in commercial buildings. The current trend is that most of the new units are being installed in the commercial buildings. It is estimated that about 90 % of GSHP units are installed closed loop systems (ground-coupled) and the remaining open loop systems (water-source). Within the residential sector, approximately 70 % of the closed loops systems are horizontal BHEs since they are cheaper to install. In the commercial sector the most dominant type (90 % of GSHP installations) are vertical BHEs due to rather limited space. Geothermal energy is currently supplying 21 TWh per year of heating in the United States. The corresponding installed capacity is 17.5 GW. It should be noted that most GSHPs are sized for the cooling loads and are generally oversized in terms of heating capacity (Boyd et al., 2015). The most common type of secondary fluids used for GSHP application are propylene glycol (up to 30 wt-\%, corresponding to the freezing point of \(-13\) °C), methyl alcohol (up to 20 wt-\%, corresponding to the freezing point of \(-15\) °C) and ethyl alcohol (up to 24 wt-\%, corresponding to the freezing point of \(-14\) °C) (IGSHPA, 2011). Recently, ethyl alcohol based secondary fluids are becoming more and more popular type of the secondary fluid for GSHP application (Spitler, 2016). The most common denaturing agents for ethyl alcohol based secondary fluids is methyl alcohol (methanol) (3.76 - 10 wt-\%) (Lyondell, 2003) that is considered very toxic for human even in small concentrations. Nevertheless, other European commercial products containing ketones or propyl and butyl alcohols can be found on American market as well.

Ignatowicz et al., 2014; Ignatowicz et al., 2015; and Ignatowicz et al., 2016 showed that presence of propyl alcohol in ethyl alcohol solution improves the thermophysical properties such as specific heat capacity, thermal conductivity and dynamic viscosity, when added in small concentrations. The chemical character of various denaturing agents and concentrations can in different way affect the thermophysical properties. Moreover no data regarding the methylated ethyl alcohol as secondary fluid has been found. Thus, a comparative study is made to evaluate the properties of different ethyl alcohol and methyl alcohol based commercial products in Europe and the United States.

METHODOLOGY

Five different ethyl alcohol water based solutions with different denaturing agents used in Europe and the United States for GSHP application were studied. The total alcohol concentration in all samples was set to be 20 wt-\%. Note that samples containing ketones have higher total concentrations compared to samples with different alcohol based denaturing agents. The measured thermophysical properties (freezing point, density, dynamic viscosity, thermal conductivity and specific heat capacity) for different solutions were compared with three reference fluids (deionized...
water, pure 20 wt.% ethyl alcohol solution, (EA20) and pure 20 wt.% methyl alcohol solution, (MA20) to evaluate the measurement errors. Note that commercial glycol products used in Europe and the United States are not investigated in this study due to differences in thermophysical properties. Table 1 summarizes the chemical composition of different ethyl alcohols solutions.

<table>
<thead>
<tr>
<th>Sample</th>
<th>Ethyl alcohol (wt-%)</th>
<th>Denaturing agent 1</th>
<th>Denaturing agent 2</th>
</tr>
</thead>
<tbody>
<tr>
<td>EA20</td>
<td>20.0 (24.47 vol-%)</td>
<td>0</td>
<td>0</td>
</tr>
<tr>
<td>EA18+PA1.6+BA0.4</td>
<td>18.0 (22.07 vol-%)</td>
<td>propyl alcohol (1.6 wt-%)</td>
<td>n-butyl alcohol (0.4 wt-%)</td>
</tr>
<tr>
<td>EA17.5+PA2+BA0.5</td>
<td>17.5 (21.47 vol-%)</td>
<td>propyl alcohol (2 wt-%)</td>
<td>n-butyl alcohol (0.5 wt-%)</td>
</tr>
<tr>
<td>EA20+MEK1.8+MIBK2.7</td>
<td>20.0 (24.47 vol-%)</td>
<td>methyl ethyl ketone (1.8 vol-%)</td>
<td>methyl isobutyl ketone (2.7 vol-%)</td>
</tr>
<tr>
<td>EA20+MEK2+MIBK0.5</td>
<td>20.0 (24.47 vol-%)</td>
<td>methyl ethyl ketone (2 vol-%)</td>
<td>methyl isobutyl ketone (0.5 vol-%)</td>
</tr>
<tr>
<td>EA18+MA2</td>
<td>18.0 (22.07 vol-%)</td>
<td>methyl alcohol (2 wt-%)</td>
<td>0</td>
</tr>
<tr>
<td>MA20</td>
<td>0.0</td>
<td>methyl alcohol (20 wt-%)</td>
<td>0</td>
</tr>
</tbody>
</table>

**Freezing Point**

The freezing point temperature was measured using microDSC7 evo from Setaram Instrumentation. Differential Scanning Calorimetry (DSC) method is a thermoanalytical technique in which the difference in the amount of heat required to increase the temperature of sample and reference is measured simultaneously. Both sample and reference are maintained at the same temperature throughout the experiment. First the water sample was tested in order to define the testing parameters, using continuous standard zone mode at four different heating and cooling scanning rates: 0.025; 0.05; 0.1 and 0.15 K min⁻¹. The difference in results for the three first scanning rates was only 0.01 K, thus, the scanning rate of 0.1 K min⁻¹ was chosen. The sample volume was always kept the same (750μl) and each test was repeated twice. The accuracy of temperature measurements for the instrument according to the manufacturer is set to be ± 0.1 K. The experimental results were compared with reference values for pure water, 20 wt.% ethyl alcohol (EA20) and 20 wt-% methyl alcohol (MA20) solutions (Lide, 1996-97).

**Density**

The density measurements were performed using pycnometers. The pycnometer is a glass bottle with a stopper having a capillary tube through it. By knowing the total volume and by measuring the mass of empty as well as of full pycnometer with Mettler Toledo high accuracy analytical balance (accuracy of +/- 0.0001 g), it was possible to determine the density of solutions at 20 °C. The accuracy of density measurement at 20 °C using calibrated pycnometer (volume 25.131 cm³) is of ± 0.2 %. The experimental results were compared with reference values for pure water, 20 wt-% ethyl alcohol (EA20), and 20 wt-% methyl alcohol (MA20) solutions found in NIST database, (Lide, 1996-97), (Melinder, 2007) and (Melinder, 2010). Later, all results can be fitted to a function to extrapolate values in the desired range between -13 °C and 30 °C with the help of literature values.

**Dynamic viscosity**

Brookfield rotational viscometer DV-II Pro with special low viscosity adapter (UL-adapter) was used to perform dynamic viscosity measurements in the temperature range between -13 and 30 °C with the instrument accuracy of ± 1 %. The working principle of the rotational viscometer is to drive a spindle immersed in the test fluid through a calibrated spring. The viscous drag of the fluid against the spindle is later measured by the spring deflection. All measurements were done using the same UL-adapter and spindle to reduce the uncertainty of measurements. The dynamic viscosity result was obtained as the slope of shear stress versus shear rate function for the range of torque between 10 and 90 %. The obtained results for water, 20 wt-% ethyl alcohol (EA20), and 20 wt-% methyl alcohol (MA20) were later compared to reference values found in NIST database, (Melinder, 2007) and (Melinder, 2010).
Thermal conductivity

Thermal conductivity measurements were performed using Transient Plane Source (TPS) method by means of Hot Disk Thermal Constants Analyser TPS-2500S having the accuracy of ± 2%. Hot Disk sensor consisted of an electrically conducting pattern in the shape of a double spiral, which had been etched out of a thin metal foil. This spiral was sandwiched between two thin sheets of Kapton insulating material. When performing a thermal conductivity measurement, the plane Kapton sensor was fitted between two pieces of the sample holder filled with tested sample. By passing an electrical current high enough to increase the temperature of the sensor between a fraction of a degree up to several degrees, and at the same time recording the resistance (temperature) increase as a function of time, the sensor is used both as a heat source and as a dynamic temperature sensor. Kapton sensor 7577 with radius 2.001 mm was chosen and tests for a given temperature were repeated three times at different measuring time (2 - 3 s) and output power (20 - 30 mW). All samples had the same volume of 10 ml and were tested in the temperature range between -13 and 30 °C. The measurement results for water, 20 wt-% ethyl alcohol (EA20), and 20 wt-% methyl alcohol (MA20) were later compared to reference values found in NIST database, (Melinder, 2007) and (Melinder, 2010).

Specific heat capacity

The specific heat capacity was measured using a microDSC evo7 from Setaram Instrumentation. The tests were performed in cp continuous mode with heating scanning rate of 0.05 K.min⁻¹ in temperature range between -15 and 30 °C. Sample volume was always kept constant (750 μl). The accuracy of specific heat capacity measurements is ± 1%. The obtained results for water, 20 wt-% ethyl alcohol (EA20), and 20 wt-% methyl alcohol (MA20) were later compared to reference values found in NIST database, (Melinder, 2007) and (Melinder, 2010).

RESULTS

The freezing point results for different ethyl alcohol with denaturing agents and methyl alcohol solutions are presented in table 2. As seen, the presence of denaturing agents has affected the freezing point. Note that both samples containing ketones had higher total concentrations compared to samples with different alcohol based denaturing agents. The presence of ketones and methyl alcohol as denaturing agents had the strongest effect on the freezing point and a decrement in the freezing point was observed. EA20+MEK1.8+MIBK2.7 had the highest concentration of ketones and the lowest freezing point of -13.47 °C among all samples. The lowest decrement of about 0.4 K in the freezing point was obtained for EA18+MA2. Both propyl alcohol and n-butyl alcohol had an opposite effect and the increment in freezing point was observed. The highest freezing point of -10.46 °C had been measured for EA17.5+PA2+BA0.5, which was having the highest content of propyl and n-butyl alcohols. These results can be explained by the fact that both propyl and n-butyl alcohol water based solutions have higher freezing points compared to pure ethyl alcohol solutions (Lide, 1996-97). Thus, higher concentration of propyl and n-butyl alcohol in solution results in higher freezing point. Note that no reference data for the different ethyl alcohol solutions with denaturing agents were found and the freezing temperatures were compared with pure EA20 solution. Experimental results for pure methyl alcohol solution were in good agreement with the reference (Lide, 1996-97).

<table>
<thead>
<tr>
<th>Sample</th>
<th>T_f exp (°C)</th>
<th>T_f ref (°C)</th>
<th>Difference (K)</th>
</tr>
</thead>
<tbody>
<tr>
<td>EA20</td>
<td>-10.92</td>
<td>-10.92</td>
<td>0.00</td>
</tr>
<tr>
<td>EA18+PA1.6+BA0.4</td>
<td>-10.58</td>
<td>-10.92</td>
<td>+0.34</td>
</tr>
<tr>
<td>EA17.5+PA2+BA0.5</td>
<td>-10.46</td>
<td>-10.92</td>
<td>+0.46</td>
</tr>
<tr>
<td>EA20+MEK1.8+MIBK2.7</td>
<td>-13.47</td>
<td>-10.92</td>
<td>-2.55</td>
</tr>
<tr>
<td>EA20+MEK2+MIBK0.5</td>
<td>-12.21</td>
<td>-10.92</td>
<td>-1.29</td>
</tr>
<tr>
<td>EA18+MA2</td>
<td>-11.34</td>
<td>-10.92</td>
<td>-0.42</td>
</tr>
<tr>
<td>MA20</td>
<td>-15.06</td>
<td>-15.02</td>
<td>0.04</td>
</tr>
</tbody>
</table>
Figure 1, below presents the results of density measurements at 20 °C for different ethyl alcohol and methyl alcohol based secondary fluids. The difference of around 0.14 % between the experimental and reference value for water was obtained. The experimental result at 20 °C was slightly higher by up to 0.3 % than the reference value for EA20 found in (Lide, 1997-98), (Melinder, 2007) and (Melinder, 2010), which could be related to the testing method. The experimental result for MA20 was lower by up to 0.3 % than reference values (Lide, 1997-98), (Melinder, 2007) and (Melinder, 2010). As seen, any change in the alcohol concentrations in sample can affect slightly the density due to the fact that both propyl alcohol and n-butyl alcohol have higher densities than ethyl alcohol. The densities of both methyl ethyl ketone as well as methyl isobutyl ketone are lower than that the pure ethyl alcohol (Lide, 1997-98). Therefore, both EA20+MEK2+MIBK0.5 and EA20+MEK1.8+MIBK2.7 samples had the lowest densities among all ethyl alcohol based samples. EA18+MA2 had slightly higher density due to presence of methyl alcohol in sample compared to pure EA20.

Figure 2, presents the results of dynamic viscosity measurements. The results obtained for water in full temperature range were higher up to 3 % with reference values (NIST). EA20 and MA20 results were slightly lower than reference values found in (Melinder, 2007) and (Melinder, 2010). As seen, the presence of the denaturing agents in small concentration can significantly decrease the dynamic viscosity in full temperature range. EA18+MA2 had the lowest dynamic viscosity by up to 12 % compared to EA20 due to the fact that methyl alcohol water solutions have lower dynamic viscosity in general. EA18+PA1.6+BA0.4, EA20+MEK2+MIBK0.5 and EA17.5+PA2+BA0.5 had as well lower dynamic viscosity than EA20 at temperature of -8 °C by up to 8.4 %, 7 %, and 3.5 % respectively. Only EA20+MEK1.8+MIBK2.7 had the dynamic viscosity higher by up to 2 % than EA20. Thus, the chemical character and concentration of different denaturing agents influences in different way the dynamic viscosity. Similar observations were reported for different blends of ethyl alcohol with propyl alcohol as well as ethyl alcohol with methyl ethyl ketone (Ignatowicz et al., 2014), (Ignatowicz et al., 2015) and (Ignatowicz et al., 2016). Thus, the concentration of denaturing agents in form of alcohols and ketones should be the lowest in order to decrease the dynamic viscosity of ethyl alcohol based secondary fluid.
Figure 3 presents the results of the thermal conductivity measurements. The difference between the experimental results and NIST reference values for water was less than 0.7 %, which is significantly below the measurement error of instrument set to be ± 2 % (≈ 0.02 W K⁻¹ m⁻¹). It is important to underline that the comparison of different alcohol blends is based on the experimental results obtained for EA20 and MA20 in order to include in the analysis the measurement error. Note that the density and specific heat capacity are input values for post processing of thermal conductivity results since knowledge of the volumetric heat capacity decreases the measurement error below 2 %. Higher values of specific heat capacity for EA20 could explain the steeper slope of curve compared to the reference data found in (Melinder 2007) and (Melinder 2010). As seen, only EA18+PA1.6+BA0.4 had higher thermal conductivity by up to 2 % at temperature of -8 °C than EA20. Moreover, EA17.5+PA2+BA0.5 had lower thermal conductivity values by up to 2.5 % at temperature of -8 °C and 6 % at temperature of 5 °C. Thus, small changes in different alcohol concentration can affect thermal conductivity in a different way. Previous results (Henry and Ignatowicz, 2014) showed that n-butyl alcohol only at small concentrations can increase the thermal conductivity and propyl alcohol at same concentration is giving around 2 % higher values. Thus, small changes in concentrations of three alcohols, especially ethyl and propyl alcohol, can affect the slope of obtained curve and explain the difference in slopes for EA17.5+PA2+BA0.5 and EA18+PA1.6+BA0.4 curves. Moreover, too high concentration of n-butyl alcohol (1.25 wt-%) in water seems to decrease the thermal conductivity by up to 8 % (Bertolini et al., 1990). Higher concentration of n-butyl alcohol makes the curve flatter at very low and high temperatures and its effect is becoming stronger at higher concentrations. The presence of denaturing agents in form of ketones had a negative effect on the thermal conductivity in full temperature range. EA20+MEK2+MIBK0.5 had the thermal conductivity values lower by up to 3 % at temperature of -8 °C and 8 % at temperature of 5 °C.

![Figure 3 Thermal conductivity results.](image1)

![Figure 4 Specific heat capacity results.](image2)
Similar observation was made for EA20+MEK1.8+MIBK2.7 and values lower by up to 7\% at temperature of -8 °C and 13\% at temperature of 5 °C were observed. Thus, the type and concentration of denaturing agent can affect the thermal conductivity and the slope of curve in different ways. MA20 results were lower by up to 1\% at temperature of -13 °C and 3\% at temperature of 30 °C compared to reference values found in (Melinder 2007) and (Melinder 2010), which could be explained again by slightly higher values of volumetric heat capacity used for post processing. EA18+MA2 had lower thermal conductivity by up to 0.2\% at temperature of -10 °C and 15\% at temperature of 30 °C compared to pure EA20. Moreover, EA18+MA2 sample had almost same thermal conductivity results compared to reference EA20 data.

Figure 4, shows the results of the specific heat capacity measurements. The experimental results for water were up to 1.5\% higher than NIST references values whereas the accuracy of instrument is set to be ±1 \% (≈ 10 J kg^-1 K^-1). The standard deviation for five tests for water was 24 J kg^-1 K^-1 and higher measurement error for water could be explained by the small sample volume of 750 μl. Thus, EA20 and MA20 samples were used as the benchmark so that all measurement errors are considered. Recent results for EA20 solutions, showing a different tendency or slope than some literature values, were reported in (Ignatowicz et al., 2014), (Ignatowicz et al., 2015) and (Ignatowicz et al., 2016). EA17.5+PA2+BA0.5 had the highest specific heat capacity by up to 2.5\% than EA20 and by up to 1.5\% higher than EA18+PA1.6+BA0.4. EA20+MEK2+MIBK0.5 and EA20+MEK1.8+MIBK2.7 showed very similar results and gave by up to 1.5\% higher specific heat capacity than EA20. EA18+MA2 had higher specific heat capacity by up to 1.2\% instead. EA18+PA1.6+BA0.4 had higher specific heat capacity only in temperature range between -10 and 0 °C (due to the small concentration of n-butyl alcohol) than both samples containing ketones and EA18+MA2. Moreover, measurement results for MA20 were lower by up to 3\% at temperature of -13 °C and 5\% at temperature of 30 °C compared to reference values found in (Melinder 2007) and (Melinder 2010). The difference in results for MA20 could be explained by the fact that the testing method was different and more sensitive DSC instruments with lower scanning rates are available nowadays. It seems that all ethyl alcohol samples with denaturing agents (ketones as well as different alcohols) seem to have higher specific heat capacity than pure ethyl alcohol solution. This phenomenon could be explained by the types of hydrogen bonds between alcohols and water as well as uniqueness of the binary, tertiary and quaternary systems (Peeters and Leroy, 1994) but research studies still continues to understand more the nature of interactions between alcohol molecules in water. The thermophysical properties of different ethyl alcohol based secondary fluids with denaturing agents and methyl alcohol solution are summarized in table 3 presented below.

**Table 3. Thermophysical properties of solutions with denaturing agents.**

<table>
<thead>
<tr>
<th>Sample</th>
<th>T (°C)</th>
<th>ρ (kg m^-3)</th>
<th>μ (mPas)</th>
<th>k (W m^-1 K^-1)</th>
<th>Cp (J kg^-1 K^-1)</th>
</tr>
</thead>
<tbody>
<tr>
<td>EA20</td>
<td>30</td>
<td>959.75</td>
<td>1.75</td>
<td>0.5362</td>
<td>4450.41</td>
</tr>
<tr>
<td></td>
<td>20</td>
<td>964.75</td>
<td>2.39</td>
<td>0.5039</td>
<td>4454.96</td>
</tr>
<tr>
<td></td>
<td>10</td>
<td>968.45</td>
<td>3.46</td>
<td>0.4793</td>
<td>4432.63</td>
</tr>
<tr>
<td></td>
<td>5</td>
<td>970.05</td>
<td>4.29</td>
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<tr>
<td></td>
<td>0</td>
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<td>0.4525</td>
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</tr>
<tr>
<td></td>
<td>-5</td>
<td>972.70</td>
<td>7.06</td>
<td>0.4369</td>
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<tr>
<td></td>
<td>-8</td>
<td>973.33</td>
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<tr>
<td>EA18+PA1.6+BA0.4</td>
<td>30</td>
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<td>7.76</td>
<td>0.4413</td>
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</tr>
<tr>
<td>Sample (cont.)</td>
<td>T (°C)</td>
<td>ρ (kg m⁻³)</td>
<td>μ (mPa s)</td>
<td>k (W m⁻¹ K⁻¹)</td>
<td>Cp (J kg⁻¹ K⁻¹)</td>
</tr>
<tr>
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<td>--------</td>
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</tr>
<tr>
<td>EA17.5+PA2+BA0.5</td>
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CONCLUSION

This study showed that the chemical character of various denaturing agents and concentrations can in a different way affect the thermophysical properties of commercial products found in Europe and the United States. As seen, the presence of two ketones had a strong effect on the freezing point. Methyl, propyl and n-butyl alcohols used as denaturing agents increased the density while methyl ethyl and methyl isobutyl ketones decreased the density values compared to pure EA20. Moreover, different denaturing agents were positively affecting the dynamic viscosity in most cases and EA18+MA2 had the lowest dynamic viscosity by up to 12 %. EA18+PA1.6+BA0.4, EA20+MEK2+MIBK0.5 and EA17.5+PA2+BA0.5 had as well lower dynamic viscosity than EA20 at the temperature of -8 ºC by up to 8.4 %, 7 %, and 3.5 % respectively. Only EA20+MEK1.8+MIBK2.7 had the dynamic viscosity higher by up to 2 % compared to pure EA20.

As seen, different concentrations of propyl and n-butyl alcohol were affecting in different way the thermal conductivity and EA18+PA1.6+BA0.4 showed higher thermal conductivity than EA20. The presence of denaturing agents in form of ketones decreased the thermal conductivity in full temperature range. Methyl alcohol as denaturing agent was not increasing the thermal conductivity value and EA18+MA2 sample had almost the same thermal conductivity values compared to reference EA20 data.

EA17.5+PA2+BA0.5 had the highest specific heat capacity, by up to 2.5 % than EA20 and by up to 1.5 % than EA18+PA1.6+BA0.4. EA20+MEK2+MIBK0.5 and EA20+MEK1.8+MIBK2.7 showed very similar results and gave by up to 1.5 % higher specific heat capacity. EA18+MA2 had higher specific heat capacity by up to 1.2 % compared to EA20 and EA18+PA1.6+BA0.4 had higher specific heat capacity only in temperature range between -10 and 0 ºC compared to ethyl alcohol samples with ketones and methyl alcohol, which is the most important operational temperature range for Swedish GSHPs.

Summing up, the commercial product commonly used in Sweden (EA18 + PA1.6 + BA0.4) showed the best thermophysical properties among different ethyl alcohol based products found in Europe when taking into consideration all thermophysical properties. Pure MA20 poses better thermophysical properties than EA18+MA2 and the lowest viscosity among all investigated alcohol blends. MA20 has as well good properties but special care needs to be taken due to high toxicity of methyl alcohol. Moreover, EA18+MA2 does not have good thermophysical properties compared to other ethyl alcohol blends and can be considered toxic. Thus, products containing small amounts of propyl and butyl alcohol or ketones are more recommended instead.

ACKNOWLEDGMENTS

The Swedish Energy Agency, Effsys Expand program and all industrial partners are gratefully acknowledged for financing this project.

NOMENCLATURE

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<th>Symbol</th>
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<td>BA</td>
<td>n-butyl alcohol</td>
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<tr>
<td>BHE</td>
<td>Borehole Heat Exchanger</td>
</tr>
<tr>
<td>c_p</td>
<td>specific heat capacity (J·kg⁻¹·K⁻¹)</td>
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<tr>
<td>EA</td>
<td>ethyl alcohol</td>
</tr>
<tr>
<td>GSHP</td>
<td>Ground Source Heat Pump</td>
</tr>
<tr>
<td>k</td>
<td>thermal conductivity (W·m⁻¹·K⁻¹)</td>
</tr>
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<td>MA</td>
<td>methyl alcohol</td>
</tr>
<tr>
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<td>methyl ethyl ketone</td>
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<tr>
<td>MIBK</td>
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</tr>
<tr>
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</tr>
<tr>
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<td>volume concentration (-)</td>
</tr>
<tr>
<td>wt-%</td>
<td>weight concentration (-)</td>
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<tr>
<td>μ</td>
<td>dynamic viscosity (mPa·s)</td>
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<tr>
<td>ρ</td>
<td>density (kg·m⁻³)</td>
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REFERENCES


Spitler J.D., April 2016. Types of commercial products used for GSHPs application in the United States, personal communication.
Direct expansion ground source heat pump using carbon dioxide as refrigerant: Test facility and theoretical model presentation

Parham Eslami Nejad Messaoud Badache Mohamed Ouzzane Zine Aidoun

ABSTRACT
In an attempt to address recent challenges on using natural refrigerants and to develop further knowledge and expertise in the field of direct expansion ground source heat pump (DX-GSHP), an experimental transcritical carbon dioxide (CO₂) test bench was built at CanmetENERGY Research Laboratory. A previously developed theoretical model of the system was modified and validated against a set of experimental results and adopted to investigate the system performance in a wide operating range. A parametric analysis was also performed using the theoretical model for understanding the system and at exploring the performance improvement actions for future installations. Validation results showed that the model predicts the experimentation very well within the uncertainty of the measurement. Furthermore, parametric analysis showed that improper control of some parameters such as gas cooler CO₂ outlet temperature and discharge compressor pressure can degrade the system performance by up to 25% and the heat pump heating capacity by 7.5%.

INTRODUCTION
Although the detrimental environmental impacts of conventional refrigerants have raised global concern, due to the worldwide growing energy demand, high energy efficiency of heat pumps still remains a great incentive for using this technology in residential and commercial buildings. Over the last decade, several studies have been conducted to replace synthetic refrigerants with natural ones. Among the candidates, CO₂ has been attracting more attention as it is environmentally benign and safe. This, together with the established energy efficiency advantages of the GSHP, makes the CO₂ GSHP a promising, environmentally friendly, and energy efficient alternative to other heating equipment. However, the scarcity of related technical knowledge may slow down its development pace.

Many studies have been performed on the air-source and water-source transcritical CO₂ heat pump systems that are not the focus here; in general however, modeling of the DX-GSHP has been rarely studied (Ndiaye 2016). Kruse and Russmann (2005) and Bertsch et al. (2005) proposed a ground heat pipe technology with two-phase CO₂ as a secondary fluid for extracting heat from the ground and transferring it to the GSHP using the thermosyphon principle. Both studies used pipe-in-pipe configurations. They compared the proposed system with a conventional system using a single phase water/brine solution.

Recently Mastrullo et al. (2014) performed modeling of a CO₂-filled U-tube ground heat exchanger (borehole) under thermosyphon principle for the secondary loop GSHP systems. Another study by Eslami Nejad et al. (2014)
focused on the numerical modeling of CO2-filled U-tube vertical boreholes under forced circulation.

Very few works have looked at the whole CO2 GSHP cycle. Austin and Sumathy (2011) simulated a simple CO2 transcritical cycle. However, they did not account for dynamic characteristics of the system. A recent study by Eslami Nejad et al. (2015) developed a quasi-transient CO2 transcritical ground source heat pump model along with numerical and experimental validation of the borehole portion. In the present study, the model is validated using a set of experiments performed at CanmetENERGY Research Laboratory and then, it is modified and used to perform a parametric analysis on several system parameters.

SYSTEM DESCRIPTION

Figure 1 shows the schematic presentation of a single-stage transcritical CO2 DX-GSHP system with hot gas bypass working in heating mode. The system consists of eight main system components including compressor (1-2), gas cooler (2-3), internal heat exchanger (3-4), two different expansion valves (4-5 and 6-7), pressure regulating valve (9-10), receiver (5-6) and boreholes (7-8).

As shown in Figure 1, CO2 (Refrigerant) is flowing through a complete cycle by going directly down to the borehole, changing direction at the bottom (U connection) and coming up to extract heat ($q_b$) from the ground by evaporation. Then it enters the internal heat exchanger (IHE) to exchange heat with the gas at the gas cooler exit in order to be superheated to a certain degree. The gas is then compressed by the compressor to supercritical pressure with a corresponding temperature rise. The high pressure/high temperature vapor enters the gas cooler to heat the water ($q_h$). After the IHE, low temperature/high pressure CO2 gas is throttled to the intermediate pressure level of the cycle. Two-phase CO2 (vapor and liquid) enters the separator and the vapor part is bypassed around the boreholes. Both liquid and vapor parts are throttled to the low pressure level of the cycle. Finally, CO2 with very small vapour quality enters the boreholes and mixes with bypassed vapor at the borehole outlet.

![Figure 1](image)

**Figure 1** (a) System schematic and (b) qualitative temperature-enthalpy curve
THEORETICAL MODEL

The theoretical model comprises transient analytical model of the ground, steady-state numerical heat transfer models of the borehole and gas cooler, and steady-state heat transfer and thermodynamic models of other components such as the expansion valves, pressure regulating valve the compressor, the IHE and the receiver.

Ground heat exchanger and the soil

The numerical steady-state fluid flow and heat transfer model of the borehole developed by Eslami-Nejad et al. (2014) is adopted and combined with the transient classical finite line source (FLS) model of the ground. FLS is used for modeling the heat transfer in the the ground as well as the thermal interaction between boreholes with the heat pulse response superposition in time using a non-history scheme proposed by Lamarche and Beauchamp (2007). For validation of the ground-heat exchanger model, readers are referred to Eslami-Nejad et al. (2015). It is assumed that the grout and ground materials are homogeneous and the heat capacity of the grout is negligible.

Gas cooler

Counter-flow heat exchanger is assumed for the gas cooler. The gas cooler is also discretized along its length into equal control volumes to account for strong temperature-heat-transfer-rate nonlinear behavior in the supercritical region (Chen 2016). Calculation starts from the outlet of the CO₂ side and the inlet of the water side where both temperatures are known. In addition to energy balance between water and CO₂, the LMTD method is used to calculate heat exchange rate at each control volume. Fundamental conservation equation of momentum based on appropriate correlations for CO₂ pressure drop is also applied to each control volume element. The system of equations obtained (energy, heat transfer and momentum) is nonlinear and strongly linked. An iterative method is therefore applied to solve the set of equations (Eslami-Nejad et al. 2015).

Compressor

The volumetric and isentropic efficiency equations, Eq. 1 and Eq. 2 respectively, have been correlated using manufacturer data and used along with other equations to calculate CO₂ total mass flow, compressor work, discharge pressure and temperature.

\[
\eta_v = 1.0649 - 0.1044 \left( \frac{P_2}{P_1} \right) \tag{1}
\]

\[
\eta_{isen} = 0.518 + 0.0666 \left( \frac{P_2}{P_1} \right) - 0.0137 \left( \frac{P_2}{P_1} \right)^2 \tag{2}
\]

Fundamental energy, mass or thermodynamic equations are used for other system components (expansion valves, IHE and separator) to complete the system of equations (Eslami-Nejad et al. 2015).

Solution procedure

At each time step, three main iterative numerical procedures, as well as several internal iterative calculation loops, are used to determine the operating conditions of the system components as well as the ground thermal condition for the next time step. In the last iterative loop, the borehole wall temperature is updated, using the transient heat transfer calculation in the ground. Based on the convergence criteria for each loop, all three main loops
interact iteratively until they all converge. The code is developed in FORTRAN, to which REFPROP Version 9.1 subroutines (Lemmon et al. 2013) are linked, to calculate the thermodynamic properties of CO$_2$ and water. More details about different components have been described in Eslami Nejad et al. (2015)

**EXPERIMENTAL TEST BENCH**

In an attempt to address recent challenges on using natural refrigerant in heat pumps and to develop more expertise on CO$_2$ DX-GSHP, an experimental test bench was built at CanmetENERGY Research Laboratory. The test facility is a transcritical CO$_2$ DX-GSHP with a hot gas bypass working in heating mode. It comprises a semi hermetic compressor (Table 1), a water loop for heat rejection from the gas cooler, two counter-flow plate heat exchangers for the gas cooler ($A_{GC}=0.74m^2$ and $UA_{GC}=0.3kW/K$) and for IHE ($A_{IHE}=0.092m^2$ and $UA_{IHE}=0.1kW/K$), as well as four 30-meter vertical boreholes with single copper U-tube. The boreholes are arranged in a square pattern with a uniform spacing of 6.25 m. Borehole dimensions are listed in Table 2. Insitu thermal properties of the soil obtained from a thermal response test are given in Table 3.

<table>
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<th>Table 2. Borehole dimensions</th>
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The setup is fully equipped with different measuring devices including an electric power meter, pressure and temperature sensors and flow meters to evaluate precisely the overall system performance as well as every specific components of the cycle. Figure 2a presents a schematic of the test bench with all measuring devices at different locations.

The cycle is divided into three pressure zones (Figure 2a): high (red dash line), intermediate (orange long dash line) and low (blue solid line) pressure. The solid orange line in Figure 2a shows the oil flow path from the oil separator to the compressor. The oil mixes with CO$_2$ and therefore it leaves the compressor to the oil separator. System heat transfer losses to the ambient is minimised through sufficient insulation. Figure 2b is the picture of the test bench located in CanmetENERGY laboratory.

The system can be operated in two different modes; automatic and manual. The automatic mode is focused on this paper and it is described in the following section. In the manual mode the opening position of the expansion valves installed before the boreholes can be changed manually.

**Control**

In automatic mode, four control strategies are applied as follows:

- The valve that discharges the vapor from the receiver maintains the intermediate pressure ($P_{receiver}$) at 3750 kPa. This value is the set point given by the operator that corresponds to CO$_2$ saturation temperature of 2.8 °C.
• Optimum high pressure is controlled by the expansion valve installed after the IHE on the high pressure level. Several tests were performed to correlate the following control function giving the optimum pressure ($P_{opt}$) in kPa as a function of gas cooler outlet temperature ($T_{gc,out}$) in °C.

$$P_{opt} (T_{gc,CO2}) = 149.7 \times T_{gc,out} + 2588.4$$

(3)

• Expansion valves’ opening changes automatically to be able to provide superheat temperature set point (1.5 °C) at the borehole exit.

• Two valves installed at the low pressure level before the IHE satisfy the second superheat set point (5.0 °C) at the compressor suction by modulating the flow of CO$_2$ passing through the IHX.

**Measurements and uncertainty**

Uncertainties of measuring equipments are presented in Table 4. The uncertainty of temperature measurement can go up to ±0.8 °C (for measuring high temperatures) due to the use of pipe surface thermocouples. Temperature uncertainties due to heat losses from pipe surface to the ambient and the thermal contact resistance between the thermocouples and the pipe surface is not taken into account.

**Table 4. Measurement uncertainties**

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<tr>
<td>Power input to the compressor</td>
<td>Wattmeter</td>
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VALIDATION

There are minor differences between the theoretical model (based on Figure 1a) and the installation (Figure 2a), including separate expansion valve for each borehole and an oil circuit in the test bench but neither of which was considered by the model. A 17-hour experiment was performed in heating mode for model validation with constant water mass flow rate \( \dot{m}_{water} \) entering the gas cooler at 0.25 kg/s and constant inlet water temperature at 35 °C. As shown in Figure 3, CO₂ pressure value reaches 8300 kPa at the gas cooler inlet. Furthermore, intermediate pressure \( P_{receiver} = 3750 \) kPa is very close to the low pressure \( P_{in\_borehole} = 3540 \) kPa level. Evaporating pressure \( P_{in\_borehole} \) decreases slightly over time from 3540 kPa to 3412 kPa; this is caused by the expansion valves to maintain the superheat set point at the borehole outlet. Water temperature increases by 3.5 °C (from 35 °C to 38.5 °C) taking the heat from CO₂ in the gas cooler \( q_h \) and corresponding to nearly 3.6 kilowatts of heating.

![Figure 3 Cycle pressure levels](image1)

![Figure 4 Gas cooler temperatures](image2)

The model was modified to include the four control strategies implemented into the test bench. Figures 3 and 4 present predicted (line) and measured (symbols) pressure levels of the heat pump and gas cooler water and CO₂ temperatures. The model shows a very good agreement with experimental data considering the measurement uncertainties (Error bars); except, discharge pressure that fluctuations beyond uncertainties mainly due to the superheat control strategy (last point of the control subsection).

PARAMETRIC ANALYSIS

For a better understanding of the system and at exploring the performance improvement actions, a parametric analysis was undertaken using the theoretical model. This analysis focuses on producing domestic hot water using the DX CO₂ GSHP. Eight different cases are considered and compared against the base case. The ninth case is also considered combining the best two individual cases (#5 & #1). All system parameters of the base case which are different from that of the test bench are presented in Table 5. One parameter at a time is changed in each case (#1 to #8) to evaluate the effect of six different parameters including the degree of superheat at the compressor suction \( \Delta T_{sh\_comp} \), inlet water temperature to the gas cooler \( T_{in\_water} \), CO₂ gas cooler outlet temperature \( T_{out\_CO2} \), intermediate pressure \( P_{receiver} \), water mass flow rate \( \dot{m}_{water} \) and IHE efficiency \( \varepsilon_{IHE} \). The optimum pressure
control is not adopted here for the parametric analysis as the CO₂ outlet temperature from the gas cooler is kept constant. Superheat controls at the borehole outlet and at the compressor suction are combined here into one control at the compressor suction and the entire CO₂ flow is going through the IHE. The overall heat transfer coefficient ($\mathcal{U}$) of the gas cooler varies from case to case and with evaporating temperature due to changes in CO₂ mass flow rate and properties ($\mathcal{U} A_{GC}$ changes from 0.12 to 0.3 kW/K).

### Table 5. Different cases for parametric analysis

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<th>Other cases</th>
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<td>#1 $T_{\text{out,CO}_2}$ 25 °C</td>
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<tr>
<td>&amp; OD cm</td>
<td>#2 $T_{\text{in,water}}$ 25 °C</td>
</tr>
<tr>
<td>$k_{\text{grout}}$ W/m/K</td>
<td>#3 $\Delta T_{\text{sh,comp}}$ 10 °C</td>
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<tr>
<td>Length cm</td>
<td>#4 $\Delta T_{\text{sh,comp}}$ 1 °C</td>
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<tr>
<td>$T_{\text{ground}}$ °C</td>
<td>#5 $\dot{\text{m}}_{\text{water}}$ 0.03 kg/s</td>
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<tr>
<td>$\dot{\text{m}}_{\text{water}}$ kg/s</td>
<td>#6 $\dot{\text{m}}_{\text{water}}$ 0.02 kg/s</td>
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<td>$T_{\text{in,water}}$ °C</td>
<td>#7 $P_{\text{receiver}}$ 3578 kPa</td>
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<tr>
<td>$\Delta T_{\text{sh,comp}}$ °C</td>
<td>#8 $\varepsilon_{\text{IHE}}$ 10%</td>
</tr>
<tr>
<td>$P_{\text{receiver}}$ kPa</td>
<td>#9 $\dot{\text{m}}_{\text{water}}$ 0.03 kg/s</td>
</tr>
<tr>
<td>$\varepsilon_{\text{IHE}}$ %</td>
<td>$T_{\text{out,CO}_2}$ 25 °C</td>
</tr>
</tbody>
</table>

Figures 5, 6, 7 and 8 summarize the parametric analysis by demonstrating the effects of several parameters on COP, gas cooler heating capacity, compressor work and discharge pressure over the mean evaporating temperature, respectively.

![Figure 5](image1.png)  
**Figure 5** Coefficient of performance ($\text{COP}_{\text{heating}}$)

![Figure 6](image2.png)  
**Figure 6** Gas cooler heating capacity ($q_h$)

As shown in Figure 5, case #5 ($\dot{\text{m}}_{\text{water}} = 0.03$ kg/s) offers the highest COP among cases #1 to #8 for evaporating temperatures from -6 °C to 0 °C (up to 8% at 0 °C). This is due to the fact that the discharge pressure is
significantly lower (Figure 8) and thus the compressor work (Figure 7). However, Case #5 demonstrates slightly lower gas cooler heating capacity compared to the base case.

Lowering the gas cooler outlet temperature (case #1) also presents a positive effect on COP, particularly at low evaporating temperature (up to 6% at -12 °C). The gas cooler heating capacity also improves despite the increase in the discharge pressure (Figure 8) and compressor work (Figure 7). This is mainly due to the superior gas cooler performance at specific pressure and temperature conditions. On the contrary, by increasing the water temperature entering the gas cooler the COP reduces by up to 6% mainly due to the increase in the discharge pressure. Changing the IHE size and the intermediate pressure (#8 and #7 respectively) does not change the COP and \( q_h \). Under given conditions, more superheat at the compressor suction improves marginally the COP, while less superheat (#4) decreases slightly the COP. Lowering the water mass flow rate in #6 decreases significantly the COP by 15% at 0 °C. In order to satisfy the given \( T_{out,CO2} \) in #6, discharge pressure increases (Figure 8) and ultimately does the compressor work (Figure 7).

A combination of #5 and #1 (best cases) is also presented (#9) at the end of this study to show how proper design and control can promote a good system performance. In this case \( T_{sat,CO2} \) and \( m_{water} \) are changed compared to the base case (Table 5). Results show a COP improvement by up to 25% compared to #6 and 10% compared to the base case. Gas cooler capacity is also improved by 7.5% compared to the base case.

**CONCLUSION**

In this study, an experimental test bench of a transcritical CO\(_2\) DX-GSHP that was built at CanmetENERGY research laboratory was described. Then, a previously developed theoretical model of the system is modified and validated against a set of experimental results. Finally, a parametric analysis was performed using the theoretical model for understanding the system and at exploring the performance improvement actions.

Under using identical components and given conditions, the effect of various operating parameters on COP, gas cooler heating capacity, compressor work and discharge pressure was investigated. Results showed that improper control of some parameters such as gas cooler CO\(_2\) outlet temperature and discharge compressor pressure can
degrade the system performance by up to 25% for hot water production application. Furthermore, right gas cooler selection for given water mass flow rate is very critical to get the best performance and maximum heat capacity.

ACKNOWLEDGMENTS

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REFERENCES


Nanocomposite Materials Used for Ground Heat Exchanger Pipes

Jean-Sébastien Gosselin
Jasmin Raymond
Stéphane Gonthier
Mathieu Brousseau
Jean-François Lavoie

ABSTRACT

This study compares the performance of single U-pipe, double U-pipe, and coaxial ground heat exchangers (GHE) equipped with standard HDPE and thermally enhanced (TE) pipes. Sizing calculations and 10-year hourly simulations were carried out with the GLHEPro software using as input a synthetic thermal load profile of a reference, heating-dominated, medium office building located in the U.S. climate zone 5B enclosing Colorado. Energy consumption by the ground heat and ground loop pumps were then calculated from the simulated outputs. Finally, a life-cycle cost analysis was performed to compare the total costs (construction and operation) net present value of the GHEs equipped with TE pipes with those equipped with standard HDPE pipes. Results showed that the double U-pipe with thermally enhanced pipes was the best option for the conditions considered in the study. Depending on the configuration, the use of TE pipes instead of standard HDPE pipes allowed a reduction of the GHE length between 9 and 14.8% and a reduction of the construction cost between 3.3 and 8.6%. For each configuration tested, the operation costs were similar between the GHEs equipped with HDPE and TE pipes. This study demonstrates that GHEs equipped with TE pipes can be a financially viable and environmentally beneficial solution, especially if secondary benefits are factored in such as saved footprints on available real estate.

INTRODUCTION

Ground-coupled heat pumps are energy efficient and environmentally friendly systems for heating and cooling buildings but are expensive to install. The ground heat exchanger (GHE) is the most expensive component of the system. This is particularly true for vertical GHEs installed in boreholes that tend to be more expensive than horizontal installations in trenches. Technological innovations, such as space clips holding pipes separately and thermally enhanced grout (Allan and Kavanaugh, 1999) can be used to reduce the borehole thermal resistance, which translates in GHE fields with shorter and/or fewer boreholes that are cheaper to install. The thermal conductivity of the pipes is also a factor affecting the borehole thermal resistance. Typically, GHEs are built using high-density polyethylene (HDPE) pipes, which are actually thermal insulators that increase the thermal resistance of boreholes.

In recent years, significant research efforts have been made to develop thermally enhanced (TE) polymers with inorganic nanomaterial fillers due to potential applications in the automotive, aerospace, constructions, and electronic industries. These applications are for example heat sinks in electronic devices, tubing for heat exchangers, enclosures for electronic appliances, casing for small motors, and heat exchangers used in corrosive environments (Gupta et al., 2009). In the geothermal sector, commercial pipes made with HDPE resin loaded with a carbon filler have shown an increase of thermal conductivity by about 75% compared to traditional grades of HDPE used for pipes (Versaprofiles, 2014). Design calculations indicated that these thermally enhanced pipes can reduce the thermal resistance of vertical GHEs by 10 to 50%, depending on the proposed configuration using U-shaped or concentric pipes. This allows a 5 to 25% reduction of the total drilling length required to fulfill the building heating and cooling needs (Raymond and Léger, 2011; Raymond et al., 2011, 2015).

Jean-Sébastien Gosselin (jean-sebastien.gosselin@ete.inrs.ca) is a postdoctoral student and Jasmin Raymond is a professor in geothermal energy at INRS-ETE. Mathieu Brousseau and Jean-François Lavoie are engineers working at Versaprofiles that produce thermally enhanced pipes for ground-coupled heat pump systems. Stéphane Gonthier is the president of Versaprofiles.
While the aforementioned research showed the potential reduction of the boreholes thermal resistance and GHEs length, none have evaluated how much the use of TE pipes, which have currently a higher trade cost than conventional HDPE pipes, would impact the construction and operation costs of GHEs. The objective of this study was to evaluate the economic benefits of TE pipes by comparing sizing calculations and 10-year hourly simulations for various configurations of GHEs (single U-pipe, double U-pipe, and coaxial) equipped with standard HDPE and TE geothermal pipes. In addition to the construction costs, an energy analysis was carried out and allowed to take into account the operation costs of the ground heat and loop circulation pumps linked to the GHE field.

**METHODOLOGY**

Sizing calculations were carried out with the GLHEPro software (Ground Loop Heat Exchanger Professional; Spitler, 2000; Spitler et al., 2016), using as input, a synthetic thermal load profile of a reference, heating-dominated, medium office building located in the U.S. climate zone 5B enclosing Colorado (see Figure 1). Three different configurations of GHEs were considered: single U-pipe, double U-pipe, and coaxial (see Figure 2). For each configuration, a GHE was designed with standard HDPE and TE pipes using the full building loads. The operating conditions of each of the GHEs thus designed were then simulated on an hourly basis with GLHEPro over a 10-year period. The simulation outputs were then used to compute the electrical energy consumption of the ground heat pumps (GHP) and of the ground loop pumps (GLP) used to circulate the fluid. The results obtained from the sizing calculations and energy consumption analysis were then used to perform a life-cycle cost analysis to compare the economic performance of the GHEs equipped with TE pipes with those equipped with standard HDPE pipes.

**GHE Design Specifications**

**Building Load Profile.** Figure 1 presents the thermal load profile of a reference 4982 m² (53 628 ft²), 15 zones, three-story office building located in the U.S. climate zone 5B. This synthetic dataset was created by the Office of Energy Efficiency & Renewable Energy (EERE) with the EnergyPlus simulation software (Crawley et al., 2000) using a typical meteorological year at the Denver location. Heating demand is higher during the months of October to April and from May to June for cooling. Heating is dominant, with annual heating and cooling requirements of, respectively, 106.9 MW·h/yr and 66.1 MW·h/yr, corresponding to a heating on cooling ratio of 1.6.

Sizing calculation of the GHEs was carried out using monthly total and peak loads that were calculated from the hourly load profile using the Peak Load Analysis Tool distributed with GLHEPro. The maximum heating peak load occurred in January with a value estimated at 249.1 kW (850.0 kBTU/h) and a duration of 3 h, while the maximum cooling peak load occurred in June with an estimated value of 104.8 kW (357.6 kBTU/h) and a duration of 7 h.

![Figure 1: Typical one-year building load profile of a reference 4982 m² (53 628 ft²), 15 zones, three-story office building located in the U.S. climate zone 5B enclosing Colorado.](image-url)
**GHE Materials and Configurations.** For each configuration considered (Figure 2), sizing calculations and 10-year hourly simulations were carried out for two different designs: one using thermally enhanced pipes and another using standard HDPE pipes. The boreholes were arranged in a rectangular grid pattern with a separation distance of 6 m (19.7 ft) (see Figure 3) to reduce the thermal interference between individual bores (ASHRAE, 2011). The GHE design strategy consisted in adding or removing boreholes from the borefield until the borehole depth was within 150 ± 10 m (492 ± 33 ft).

![Figure 2: Scale drawing showing the pipe and the drilling dimensions of the three ground heat exchanger configurations considered in this study using, alternately, standard HDPE and thermally enhanced pipes.](image)

All boreholes were assumed to have a diameter of 152.4 mm (6 in) and to be filled with a thermally enhance grouting with an average thermal conductivity of 1.9 W/m·K (1.10 Btu/ft·°F). Thermal conductivity of the subsurface was set to a constant value of 3.0 W/m·K (1.74 Btu/ft·°F), which is a value that is representative of a granite, limestone, or sandstone bedrock material (ASHRAE, 2011). The volumetric heat capacity for the subsurface and grouting were set, respectively, to 2343 kJ/°C·m³ (34.9 Btu/ft³·°F) and 3901 kJ/°C·m³ (58.2 Btu/ft³·°F).

The nominal diameter of the pipes used for the single and double U-pipe configuration was 31.8 mm (1.25 in) and the standard dimension ratio (SDR) was 11 (Figures 2a and 2b). Space clips, attached every 3 m (10 ft) along the U-pipe, were assumed to separate the pipes from each other with a center-to-center distance of 100 mm (3.94 in). This allowed to reduce the borehole thermal resistance and the thermal short-circuiting between the cold and hot leg of the U-pipes, consequently increasing the performance of the GHEs (Hellström, 1991; Claesson and Hellström, 2011). The thermal conductivity of standard HDPE and TE pipes were set, respectively, to a value of 0.4 W/m·K (0.23 Btu/ft·°F) and...
0.7 W/m·K (0.41 Btu/h·ft·°F), as reported in Raymond et al. (2015), while the volumetric heat capacity was set to a value of 1543 kJ/°C·m³ (23.0 Btu/ft³·°F) for both types of pipe material.

The alternative coaxial design considered in this study (Figure 2c) involved the use of two pipes installed in each other. The size of the coaxial configuration was based on Raymond et al. (2015). The inner pipe had a nominal diameter of 50.8 mm (2 in) and a SDR of 11, while the outer pipe had a nominal diameter of 101.6 mm (4 in) and a SDR of 17. Two different cases for the coaxial were considered: one with an outer pipe made of HDPE and another with a TE pipe. In both cases, standard HDPE material was used for the inner pipe in order to minimize the thermal short-circuiting effect between the annulus and the inner pipe. The thermal conductivity and volumetric heat capacity for the HDPE and TE pipes were kept identical to the values given above for the U-pipe configurations.

Ground Heat Pumps Selection. Fifteen commercially available ground heat pumps (ClimateMaster, 2016) with a nominal capacity in heating of 22.6 kW (6.4 tons) were used to meet the maximum peak heating load of 249.1 kW (70.8 tons). The 15 GHPs were assumed to be all identical and connected in parallel to the GHEs (one GHP installed per building zone, see Figure 3). It was also assumed that the entering water temperatures (EWTs), both on the source and load side (building), were the same for all GHPs at all times. Therefore, all the GHPs were assumed to have the same coefficient of performance (COP) and capacity (CAP) when operating. Figure 4 shows the COP and CAP curve fits of the selected GHP as a function of the EWT at a flow rate of 0.9 L/s (14.2 gpm). These curves were estimated from the performance data sheets of the manufacturer.

![Figure 4: Curve fits of the coefficient of performance (COP) and total cooling and heating capacity (CAP) of the selected ground heat pump, estimated from the performance data sheets of the manufacturer, as a function of the entering water temperature at a flow rate of 0.9 L/s (14.2 gpm).](image)

Design and Undisturbed Subsurface Temperatures. The undisturbed subsurface temperature was set to a value of 11 °C (51.8 °F), a value that is representative of locations in the 5B USA climate zone. For instance, the undisturbed ground temperature in the Denver area, Colorado, is about 11 °C (52 °F) according to McQuay International (2002).

Following ASHRAE (2011) guidelines, the minimum EWT in heating mode was set to 0 °C (32 °F), 11 °C (51.8 °F) below the undisturbed ground temperature. In cooling mode, the maximum EWT was set to 28 °C (82.4 °F), 17 °C (62.6 °F) above the undisturbed ground temperature.

Heat Carrier Fluid. According to CAN/CSA-C448 (2013), the heat-transfer fluid shall ensure freeze protection to at least 5 °C (9 °F) below the minimum loop design temperature. Compliance to this requirement was ensured by using a 25 wt % propylene glycol heat carrier solution, which is characterized by a freezing point of −10.44 °C (13.20 °F). A minimum concentration of 25 wt % is recommended to avoid problems of corrosion and bacteria. Table 1 presents the
thermophysical properties of the selected heat carrier fluid at the minimum design EWT of 0 °C (32 °F) in heating, as well as the minimum flow rate required in the U-pipe and coaxial configurations to ensure turbulent flow. Values for pure water are also provided for comparison.

<table>
<thead>
<tr>
<th>Fluid property†</th>
<th>Units</th>
<th>Prop. Glycol</th>
<th>Pure Water</th>
</tr>
</thead>
<tbody>
<tr>
<td>Dynamic viscosity</td>
<td>mPa⋅s (lbm/ft⋅h)</td>
<td>5.91 (14.30)</td>
<td>1.77 (4.28)</td>
</tr>
<tr>
<td>Density</td>
<td>kg/m³ (lbm/ft³)</td>
<td>1030.92 (64.36)</td>
<td>999.84 (62.42)</td>
</tr>
<tr>
<td>Min. turbulent flow rate‡;</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Single/Double U-pipe</td>
<td>L/s (gpm)</td>
<td>0.35 (5.58)</td>
<td>0.11 (1.72)</td>
</tr>
<tr>
<td>Coaxial</td>
<td>L/s (gpm)</td>
<td>1.66 (26.32)</td>
<td>0.51 (8.13)</td>
</tr>
</tbody>
</table>

† Fluid properties were calculated with the GLHEPro software (Spitler et al., 2016).
‡ Flow is assumed turbulent at Re > 2300. See Figure 2 for dimensions of the U-pipe and coaxial GHEs.

Table 1: Heat carrier fluid (25 wt % propylene glycol) and pure water properties at the design minimum EWT of 0 °C (32 °F). The freezing point of 25 wt % propylene glycol is −10.44 °C (13.21 °F).

Loop Pump Design. Each and every U-pipe and coaxial loops were assumed to be connected in parallel (see Figure 3) to the main header and to have the same volumetric flow rate. Similarly, the GHPs were assumed to be all connected in parallel to a one-pipe loop system and to be activated on and off synchronously since they operate under identical conditions.

The one-pipe loop system was assumed to be equipped with a large central pump with on-off control and variable speed drives ensuring a constant nominal flow rate at various head losses (due to fluid temperature variations). The large pump and motor efficiency were assumed constant with mean values of, respectively, 70 and 90 % based on values used by Kavanaugh (2011). The central loop pump was assumed to be cycled off when the GHPs were not activated. A constant hydraulic pressure of 90 kPa (30 ft of water head) was maintained within the loop during these off times by a smaller central pump that operated in parallel at 25 % of the design volumetric flow rate. The smaller central loop pump was assumed to be cycled off when the GHPs were activated. The smaller pump and motor efficiency were also considered to be constant with values of, respectively, 50 and 85 % based on values used by Kavanaugh (2011). Note that the heat rejected to the ground by the pumps was not considered in the sizing calculation and hourly simulation of the GHEs since this component could not be directly accounted for in the GLHEPro software.

Design Volumetric Flow Rate. The total volumetric flow rate  \( \dot{V} \) was designed to ensure a ratio of 0.054 L/s/kW (3 gpm/ton) of heating load during peak conditions. Considering that the maximum heating peak load is 249.1 kW (850.0 kBTU/h), the design volumetric flow rate is estimated at 13.5 L/s (214 gpm) for the entire system or 0.9 L/s (14.2 gpm) per GHP.

For the coaxial configuration, the design volumetric flow rate had to be increased from this calculated value to ensure non-laminar flow (Re > 2300) on each and every loop of the GHE at the design EWT in heating mode. The design volumetric flow rate was calculated in this case by multiplying the minimum turbulent flow rate in heating mode of Table 1 by the number of coaxial boreholes determined during the sizing calculations.

Energy Consumption Analysis

Heat Pump Performance. The GHPs operating time in heating (OT<sub>H</sub>) and cooling mode (OT<sub>C</sub>) within a given 1-hour time interval \( \tau \) was calculated using the curve fits of CAP (Figure 4) and the simulated EWT as:

\[
OT_H[\tau] = \frac{1}{NGHP} \cdot \frac{Q_H[\tau]}{CAP_H(EWT[\tau])}
\]

\[
OT_C[\tau] = \frac{1}{NGHP} \cdot \frac{Q_C[\tau]}{CAP_C(EWT[\tau])}
\]

(1)

where \( NGHP \) is the total number of ground heat pumps and \( Q_H \) and \( Q_C \) are the hourly building heating and cooling loads. The total electrical power (\( W_{GHP} \)) consumed by the 15 GHPs over the 10-year simulation period was then calculated.
using the curve fits of \( \text{COP} \) and \( \text{CAP} \) (Figure 4) and the \( OT \) calculated with Equation 1:

\[
W_{\text{GHP}} = N_{\text{GHP}} \times \sum_{\tau=1}^{24 \times 365 \times 10} \left( \frac{\text{CAP}_{h}(EWT[\tau])}{\text{COP}_{h}(EWT[\tau])} \cdot OT_{h}[\tau] + \frac{\text{CAP}_{c}(EWT[\tau])}{\text{COP}_{c}(EWT[\tau])} \cdot OT_{c}[\tau] \right)
\]  

(2)

**Pumping Energy.** The energy consumed by the ground loop pumps (GLPs) may account for a significant portion of the building annual energy consumption (Bernier, 2001). The electrical consumption of the large and smaller loop pumps (\( W_{\text{GLP}} \)) over the 10-year simulation period was evaluated with the following equation:

\[
W_{\text{GLP}} = \sum_{\tau=1}^{24 \times 365 \times 10} \left( OT_{\text{tot}}[\tau] \cdot \frac{V}{\eta_{\text{large}} \cdot \eta_{\text{motor}} \cdot \Delta p_{\text{on}}[\tau]} + (1 - OT_{\text{tot}}[\tau]) \cdot \frac{0.25 V}{\eta_{\text{large}} \cdot \eta_{\text{motor}} \cdot \Delta p_{\text{off}}}[\tau] \right)
\]  

(3)

where \( \eta_{\text{pump}} \) and \( \eta_{\text{motor}} \) refer, respectively, to the pumping and motor efficiency of the large and small loop pumps, \( V \) is the design volumetric flow rate, \( \Delta p_{\text{on}} \) is the time-varying pressure drop in the loop system while the GHPs are activated, \( \Delta p_{\text{off}} \) is the constant pressure head of 90 kPa (30 ft of water head) delivered by the smaller loop pump when the GHPs are not activated and \( OT_{\text{tot}} \) is the total period of time during which the GHPs are operated within a given 1-hour time interval \( \tau \) and corresponds to the sum of \( OT_{h}[\tau] \) and \( OT_{c}[\tau] \).

Only the pressure drop in the borehole pipes and in the water coil of the GHPs were considered for the calculation of \( \Delta p_{\text{on}} \). The pressure drops in the return and supply header, valves and pipeline fittings were neglected. The water pressure drop (WPD) in the GHPs was determined with a curve fit of the performance data sheets of the selected GHP (ClimateMaster, 2016). For the single and double U-pipe configurations, \( \Delta p_{\text{on}} \) was calculated as the sum of the WPD and the pressure drop in the descending and ascending legs of the U-pipe. For coaxial configurations, \( \Delta p_{\text{on}} \) was calculated as the sum of the WPD and the pressure drop in the annular duct and in the inner pipe conduit.

Assuming non-laminar flow conditions at all times \( (Re > 2300) \), the pressure drop for circular pipe flow was calculated at every time step \( \tau \) with the equations presented in VDI-Gesellschaft (2010):

\[
\Delta p_{\text{pipe}}[\tau] = f_{\text{pipe}} \cdot L_{\text{pipe}} \cdot \frac{\rho[\tau] \cdot w_{\text{pipe}}^2}{2} \quad \text{with} \quad f_{\text{pipe}} = 0.3164 \cdot Re[\tau]^{-1/4}
\]  

(4)

where \( f_{\text{pipe}}, L_{\text{pipe}}, \) and \( d_{i} \) are, respectively, the friction factor, the length, and the inside diameter of the pipe, \( \rho_{\text{pipe}} \) is the average velocity of the fluid in the pipe, \( w_{\text{pipe}} \) is the Reynolds number. The pressure drop for annular flow was calculated at every time step \( \tau \) with the equations presented in Gnielinski (2009):

\[
\Delta p_{\text{ann}}[\tau] = f_{\text{ann}}[\tau] \cdot \frac{L_{\text{ann}}}{d_{h}} \cdot \frac{\rho[\tau] \cdot w_{\text{ann}}^2}{2} \quad \text{with} \quad f_{\text{ann}}[\tau] = \left( 1.8 \log_{10} \left( Re[\tau] \left( \frac{(1 + a^2) \ln a + (1 - a^2)}{(1 - a^2)^2 \ln a} \right) - 1.5 \right) \right)^{-2}
\]  

(5)

where \( f_{\text{ann}}, L_{\text{ann}}, \) and \( d_{h} \) are, respectively, the friction factor, length of the annulus, \( w_{\text{ann}} \) is the average velocity of the fluid in the annulus, \( a \) is the ratio between the outer diameter of the inner tube and the inner diameter of the outer tube, \( d_{h} \) is the hydraulic diameter, which corresponds to the length between the inner diameter of the outer tube and the outer diameter of the inner tube. The Reynolds number \( Re \) is calculated using the hydraulic diameter \( d_{h} \). The fluid properties (density and dynamic viscosity) were estimated in Equations 4 and 5 for every time step \( \tau \) by linear interpolation using property tables that were produced with the GLHEPro software.

**Cost Analysis**

To assess the economic performance of GHEs equipped with thermally enhanced pipes, a life-cycle cost analysis was performed, which consisted in calculating the construction cost of the GHE field and the net present value (NPV) of 10 years of operation of the GCHP system. The construction cost of the GHEs included the cost of drilling, installation labor, heat transfer fluid, grout, spacer clips and pipes. Table 2 presents the material and labor costs that were used to
calculate the borehole construction cost for the various configurations considered in this study. The costs are given for a borehole of 152 m (500 ft) length. All other construction costs were assumed the same from one configuration to another and were excluded from the analysis. This included the cost of the heat and loop pumps, of the horizontal pipes and trenches, and of the geothermal vault.

The operation cost included the NPV of the electrical energy used by the heat and loop pumps over the 10-year simulation period (see Equations 2 and 3). All other sources of operation costs were excluded from this analysis. The NPV of 10 years of operation was evaluated by the following equation (Park, 2010):

\[
\text{Operation Cost Net Present Value} = \sum_{n=1}^{10} \left( \frac{W_{GPH}[n] + W_{GLP}[n]}{(1+i)^n} \right) \cdot EC \cdot (1 + I_E)^n
\]

where \(n\) is the year number, \(I_E\) is the electricity escalation rate assumed to be 3% and \(i\) is the constant discount rate assumed equal to 6% based on values used in Bernier (2006) and Héault et al. (2016), \(EC\) is the average electricity cost in USA assumed equal to 10.60 €/kW·h based on EIA (2016), and \(n\) is the yearly index.

### RESULTS

Results for the sizing calculations, energy consumption, and cost analysis are presented for each GHE configuration in Table 3. Also, percentage reduction of the borehole thermal resistance and total GHE field length, area, and construction costs obtained when using TE pipes instead of HDPE pipes are presented in Table 4 for each configuration considered.

### DISCUSSION

**Sizing calculation.** Depending on the configuration, results in Table 4 show that the use of TE pipes instead of standard HDPE pipes allowed a reduction of the borehole thermal resistance between 22.3 and 24.4% and a reduction of the total GHE length between 9.0 and 14.8%. As shown in Table 3, the double U-pipe configuration equipped with TE pipes has the smallest GHE design length, with a total borehole length that is 31.1% shorter compared to the single U-Pipe GHE equipped with standard HDPE pipes. This important reduction is not only due to the lower thermal resistance of the TE double U-pipe configuration, but also because of its greater heat storage capacity that helps to dampen the impact of peak loads on the required length of the GHE (Raymond et al., 2015). Similarly, coaxial GHEs have 3.8 times more heat storage capacity than the single U-pipe GHEs. This explains why the calculated length for the HDPE and TE coaxial GHEs are smaller than those for single U-pipe by, respectively, 13.2% and 16.2%, even though the coaxial thermal resistance is higher in both cases.

<table>
<thead>
<tr>
<th>Material Costs ($US)</th>
<th>Single U-pipe</th>
<th>Double U-pipe</th>
<th>Coaxial</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>HDPE</td>
<td>TE</td>
<td></td>
</tr>
<tr>
<td>Loop</td>
<td>570</td>
<td>915</td>
<td>1240</td>
</tr>
<tr>
<td>Heat carrier fluid</td>
<td>24</td>
<td>24</td>
<td>47</td>
</tr>
<tr>
<td>Grout</td>
<td>750</td>
<td>750</td>
<td>614</td>
</tr>
<tr>
<td>Spacer clips</td>
<td>147</td>
<td>147</td>
<td>147</td>
</tr>
<tr>
<td><strong>Labor Costs ($US)</strong></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Installation</td>
<td>650</td>
<td>650</td>
<td>650</td>
</tr>
<tr>
<td>Drilling</td>
<td>8000</td>
<td>8000</td>
<td>8000</td>
</tr>
<tr>
<td><strong>Total borehole cost ($US)</strong></td>
<td>10141</td>
<td>10466</td>
<td>10698</td>
</tr>
</tbody>
</table>

Table 2: Material and labor costs details in $US calculated for a single borehole of 152 m (500 ft) for the three GHE configurations (single U-pipe, double U-pipe, and coaxial) and two pipe materials (HDPE and TE) considered in this study.
Table 3: Results summary for the sizing calculation, energy consumption analysis, and cost analysis for the three GHE configurations (single U-pipe, double U-pipe, coaxial) and two types of pipe material tested (HDPE and TE). NPV = Net Present Value; TE = Thermally Enhanced.

<table>
<thead>
<tr>
<th>Units</th>
<th>Single U-pipe</th>
<th>Double U-pipe</th>
<th>Coaxial</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>HDPE</td>
<td>TE</td>
<td>HDPE</td>
</tr>
<tr>
<td>GHE Sizing Calculation</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Design flow rate, $\dot{V}$</td>
<td>13.5</td>
<td>13.5</td>
<td>13.5</td>
</tr>
<tr>
<td>gpm</td>
<td>214</td>
<td>214</td>
<td>214</td>
</tr>
<tr>
<td>Borehole thermal resistance</td>
<td>m-K/W</td>
<td>0.0865</td>
<td>0.0672</td>
</tr>
<tr>
<td>h-ft·°F/Btu</td>
<td>0.1497</td>
<td>0.1163</td>
<td>0.0845</td>
</tr>
<tr>
<td>Number of boreholes</td>
<td>–</td>
<td>12</td>
<td>11</td>
</tr>
<tr>
<td>Borehole length</td>
<td>m</td>
<td>158</td>
<td>152</td>
</tr>
<tr>
<td>ft</td>
<td>518</td>
<td>499</td>
<td>469</td>
</tr>
<tr>
<td>GHE field length</td>
<td>m</td>
<td>1891</td>
<td>1669</td>
</tr>
<tr>
<td>ft</td>
<td>6204</td>
<td>5476</td>
<td>4698</td>
</tr>
<tr>
<td>GHE field area</td>
<td>m²</td>
<td>432</td>
<td>396</td>
</tr>
<tr>
<td>ft²</td>
<td>4650</td>
<td>4263</td>
<td>3875</td>
</tr>
<tr>
<td>Energy Analysis (Yearly Average)</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Heat pump power, $W_{GHP}$</td>
<td>kW·h/yr</td>
<td>42,648</td>
<td>42,712</td>
</tr>
<tr>
<td>kJ/yr</td>
<td>145,521</td>
<td>145,739</td>
<td>146,149</td>
</tr>
<tr>
<td>Loop pumps power, $W_{GLP}$</td>
<td>kW·h/yr</td>
<td>8257</td>
<td>8510</td>
</tr>
<tr>
<td>kJ/yr</td>
<td>28,174</td>
<td>29,037</td>
<td>23,749</td>
</tr>
<tr>
<td>Total Operation Power</td>
<td>kW·h/yr</td>
<td>50,906</td>
<td>51,223</td>
</tr>
<tr>
<td>kJ/yr</td>
<td>173,698</td>
<td>174,780</td>
<td>169,901</td>
</tr>
<tr>
<td>Cost Analysis</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Real estate footprint $†$</td>
<td>$</td>
<td>116,250</td>
<td>106,563</td>
</tr>
<tr>
<td>Average operation cost</td>
<td>$US/yr</td>
<td>5,396</td>
<td>5,429</td>
</tr>
<tr>
<td>10 years operation cost NPV</td>
<td>$US</td>
<td>46,235</td>
<td>46,524</td>
</tr>
<tr>
<td>Borehole construction cost</td>
<td>$US/bore</td>
<td>10,462</td>
<td>10,438</td>
</tr>
<tr>
<td>GHE field construction cost</td>
<td>$US</td>
<td>125,554</td>
<td>114,823</td>
</tr>
</tbody>
</table>

† Based on a land sale price of 260$ per buildable square meter (25 $ per buildable square foot).

Table 4: For each GHE configuration, % reduction of the borehole thermal resistance and total GHE field length, area, and construction costs obtained when using TE pipes instead of HDPE pipes.

<table>
<thead>
<tr>
<th>TE vs HDPE % reduction for the:</th>
<th>Single U-pipe</th>
<th>Double U-pipe</th>
<th>Coaxial</th>
</tr>
</thead>
<tbody>
<tr>
<td>Borehole thermal resistance</td>
<td>22.3</td>
<td>23.8</td>
<td>24.4</td>
</tr>
<tr>
<td>Total GHE field length</td>
<td>11.7</td>
<td>9.0</td>
<td>14.8</td>
</tr>
<tr>
<td>Real estate footprint</td>
<td>8.3</td>
<td>10.0</td>
<td>18.2</td>
</tr>
<tr>
<td>Total GHE field construction cost</td>
<td>8.6</td>
<td>3.3</td>
<td>8.0</td>
</tr>
</tbody>
</table>
Energy analysis. Results from the energy analysis show that the power consumption of the GHPs is very similar for all U-pipe GHEs, ranging from 42,648 kW·h/yr (145,521 Btu/yr) for the HDPE single U-pipe to 42,910 kW·h/yr (146,415 Btu/yr) for the TE double U-pipe. Assuming an electricity cost of 10.60¢/kW·h, this corresponds to a difference of less than 30 $/yr for the operation cost of the GHPs. This shows that the use of TE pipes does not significantly affect the performance of the GHP when sizing calculations are done properly. The GHP energy consumption of the coaxial GCHP systems are the lowest due to the increased flow rate, which helps improving the performance of the GHPs. On the other hand, this is counterbalanced by a higher loop pumping power consumption that is mainly due to the increased pressure drop within the inner pipe of the coaxial GHEs. This could have been mitigated by the use of baffles, mounted within the annulus, to enforce turbulence at lower flow rates (Steins et al., 2012). It is also worth noting that the double U-pipe GHEs require about 16% less pumping energy than the single U-pipe GHEs. This is explained by the fact that, for an equal total flow rate of 13.5 L/s (214 gpm), the mean velocity of the fluid in each loop connected in parallel to the main header is more than twice lower for the double U-Pipe than for the single U-Pipe. As per Equation 4, this results in a significant reduction of the pressure drop and a lower loop pumping power consumption when the large loop pump is operated. For the same reasons, the power consumption for pumping is slightly higher for all the GHEs equipped with TE pipes since the number of boreholes connected in parallel to the main header is less than those equipped with standard HDPE pipes.

Cost analysis. Results for the life-cycle analysis (Table 3) reveal that the 10 years of operation NPV are very similar for all the GHE configurations considered, ranging from 45,225 $ for the HDPE double U-pipe to 46,524 $ for the TE single U-pipe. The TE double U-pipe GHE have the lowest construction cost and appears to be the best option, both technically and financially, for the conditions represented in this work. In contrast, the coaxial GHEs have the greatest construction costs due to the higher trade and installation costs of the coaxial loops (see Table 2). Results in Table 4 show that TE pipes allowed a reduction of the construction costs between 3.3 and 8.6% when respectively compared to GHEs with same configuration and regular HDPE pipes. In all cases, these results suggest that GHEs equipped with TE pipes are an economically viable solution, despite the current typical trade cost of TE pipes being almost two times higher than standard HDPE pipes.

Furthermore, secondary benefits of using TE pipes to reduce the size of the GHE fields must also be factored in. As shown in Table 3, the reduction of the number of boreholes also allows to lower the total land area that is required to build the GHE fields. For larger geothermal project in urban areas where space is limited and where the land sale price per buildable square foot is high, a reduction in the number of boreholes by about 10% could represent significant additional savings on the initial cost of a project. Table 4 shows that in this study, the use of TE pipes instead of standard HDPE pipes allowed an additional saving of 8.3, 10.0 and 18.2% on the real estate value, respectively, for the single U-pipe, double U-pipe, and coaxial configuration. Moreover, less drilling length also means less fuel consumption during construction, smaller carbon footprint, shorter time of installation, less drilling residue, and less horizontal trench work. Besides, the trade cost of TE pipes can be expected to decrease in the future, which will increase even more the benefit of using TE pipes for the construction of GHEs. Of course, as with any cost analysis, the results presented here depends on the assumptions used. Benefit of using TE pipes may be greater or lower depending on drilling, material, and labor costs that can vary greatly from one location to the other in North America.

CONCLUSION

This study compared the performances of single U-pipe, double U-pipe, and coaxial ground heat exchangers (GHE) equipped with standard HDPE and thermally enhanced (TE) pipes. Sizing calculations and 10-year hourly simulations were carried out with the GLHEPro software using as input a synthetic thermal load profile of a reference, heating-dominated, medium office building located in the U.S. climate zone 5B enclosing Colorado. Energy consumption by the ground heat and ground loop pumps were then calculated from the simulated outputs and a life-cycle cost analysis was performed to compare the costs of construction and operation of the GHEs equipped with TE pipes with those equipped with standard HDPE pipes.
Results showed that the double U-pipe equipped with TE pipes was the best configuration for the conditions considered in this study. Depending on the configuration, the use of TE pipes instead of standard HDPE pipes allowed a reduction of the GHE length between 9.0 and 14.8% and a reduction of the construction cost between 3.3 and 8.6%. For each configuration tested, the operation costs were similar between the GHEs equipped with HDPE and TE pipes. This study demonstrates that GHEs equipped with TE pipes can be a financially viable and environmentally beneficial solution, especially if secondary benefits are factored in such as saved footprints on available real estate, a shorter time and smaller cost of installation, smaller carbon footprints, and less drilling residue. Future work on this topic will include the development of a tool to quickly and easily carry out sizing, performance and cost comparison analyses for a wide range of situations to present a more precise and realistic insight into the pros and cons of using TE pipes in the construction of GHE.

ACKNOWLEDGMENTS

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REFERENCES


The coupled heating optimization of hybrid GCHP system with heat compensation unit

Tian You, Xianting Li*, Wei Wu, Wenxing Shi, Baolong Wang

ABSTRACT
The coupled HCUT-GCHP system (integrating the GCHP with the efficient heat compensation unit (HCUT)) can operate in heat compensation mode and coupled heating mode to effectively eliminate the soil thermal imbalance and increase the heating supply at peak heating load. The coupled heating strategy directly influences the soil heat extraction not only at the peak load but also during the whole year. The system model is built in TRNSYS to investigate the optimal strategies of the coupled HCUT-GCHP systems with different boreholes. Results show that the systems with 80% and 100% boreholes can keep soil thermal balance well at different starting temperatures of coupled heating mode. Taking the system with 60% boreholes as an example, when the starting temperature of coupled heating mode increases from 0°C to 9°C, the power consumption for heating increases by 7.32MWh/°C, while the power consumption for heat compensation decreases by 9.40MWh/°C. For the systems with 100%, 80%, 60% and 40% boreholes, the optimal starting temperatures of coupled heating modes are respectively 9°C, 9°C, 6°C and 3°C and their annual system COPs are respectively 2.66, 2.64, 2.63 and 2.48 under the optimal strategies.

1 INTRODUCTION
Annual soil thermal imbalance (Wu et al. 2013, Zhu et al. 2014, You et al. 2016a), heating deficiency at peak heating load (Rad et al. 2013, Gehlin and Spitler 2014) and large borehole cost (Garber et al. 2013) are three important factors influencing the GCHP performance and hamper its application in heating dominant buildings. To deal with these problems, the hybrid GCHP systems integrated with boiler, solar collector are common in the research and the practical projects. However, the low energy efficiency of boiler (Ni et al. 2016) makes the system’s efficiency decrease and the expensive initial cost of solar collector (Bi et al. 2004, Si 2014) make the system’s payback time increase respectively.

Heat compensation unit with thermosyphon (HCUT) is a compact unit combining air source thermosyphon and heat pump together (You et al. 2015), of which the principle is shown in Figure 1. Compared with the conventional air source heat pump, only the three-way valve and solenoid valve are added in HCUT. When the temperature difference between air and water is large enough to driven thermosyphon, the heat from air can be naturally transferred to the water with a very high COP (12.46 under the rated condition). The refrigerant flows along the solid line in Figure 1(a). When the temperature difference is not enough, the compressor works to increase the heat capacity, as the solid line in Figure 1(b). Consequently, HCUT can fully use the heat in the air at different temperatures.

The HCUT-GCHP system (integrating the GCHP with the HCUT) is proposed in the authors’ previous work...
(You et al. 2014, Li et al. 2011). It can eliminate the soil thermal imbalance efficiently and economically. In this system, HCUT can compensate heat into soil efficiently as air source thermosyphon or heat pump in heat compensation mode during non-heating season. Recently, the authors proposed the coupled heating operation of HCUT and GCHP during the heating season to solve the heating deficiency and reduce the number of boreholes (You et al. 2016b). The operation of the coupled heating mode not only influences the accumulated soil heat extraction related to annual soil thermal balance, but also influences the peak soil heat extraction related to the heat supply at peak heating load and the number of boreholes. So, it is essential to optimize the coupled heating strategy of HCUT.

\[
\text{(a) Air source thermosyphon} \quad \text{(b) Air source heat pump} \quad \text{Figure 1 Principle of HCUT}
\]

In this paper, for the coupled HCUT-GCHP system with different reduced boreholes, the coupled heating strategies of HCUT are optimized respectively to improve the systems’ performance and reduce the systems’ cost. The coupled HCUT-GCHP system model is built in TRNSYS to simulate the system applied in a Harbin hotel. The system performance on keeping soil thermal balance, the detailed system energy consumptions under different strategies and the system heating COPs during different periods are investigated. At last, the optimal coupled heating strategy for systems with different boreholes is found out.

2 SYSTEM PRINCIPLE AND OPERATION STRATEGY

2.1 System principle

The system principle of the coupled HCUT-GCHP proposed previously is illustrated in Figure 2. HCUT is an auxiliary unit in the hybrid GCHP, which has a compact structure by sharing the evaporator and condenser of air-source thermosyphon and air-source heat pump and can easily shift between them by a self-control valve. Since HCUT has high performance when outdoor air temperature is high, it can compensate heat from air into soil efficiently in non-heating season to keep annual soil thermal balance of GCHP in heating dominant buildings, which is heat compensation mode in Figure 2(a).

Besides, reducing the soil heat extraction is another way to keep soil thermal balance annually and reduce the number of boreholes. HCUT can operate as air source heat pump during heating period to extract heat from air to reduce the soil heat extraction, which is coupled heating mode in Figure 2(b). In the coupled heating mode, borehole outlet water flows into HCUT and is reheated by the heat from air, then flows into GCHP unit. The coupled heating of HCUT can improve the heating COP and capacity of GCHP unit. However, due to the additional power consumption of HCUT, the system heating COP is not always higher than the normal GCHP heating. But, the reduced soil heat compensation benefits to the whole system COP. Besides, number of boreholes can be reduced due to the heat extraction from air.
2.2 Operation strategy

The coupled heating of HCUT not only benefits the annual soil thermal balance but also reduces the needed number of boreholes. As a consequence, the operation strategy of HCUT in coupled heating mode should be optimized to improve the system efficiency as well as the system cost.

Because the evaporator inlet temperature of GCHP \( (T_{ei}) \) directly influences the heating COP and capacity of GCHP, it’s good for the improvement of heating performance by keeping \( T_{ei} \) high. Thus, the borehole outlet temperature should be elevated before entering GCHP by the HCUT in coupled heating mode. If the acceptable lowest \( T_{ei} \) is set as \( T_{ei0} \), it means \( T_{ei0} \) is also the starting temperature of HCUT in coupled heating mode. When the borehole outlet temperature is above \( T_{ei0} \) in winter, HCUT doesn’t operate and the borehole outlet fluid flows into GCHP evaporator directly. When the borehole outlet temperature is lower than \( T_{ei0} \) in winter, HCUT operates in coupled heating mode. So, when the \( T_{ei0} \) increases, the operating time of coupled heating mode increases correspondingly and less heat is extracted from the soil.

![Figure 2 HCUT’s operating modes of the coupled HCUT-GCHP system](image)

(a) Heat compensation mode  (b) Coupled heating mode

Figure 2 HCUT-GCHP system model based on TRNSYS

3 METHODOLOGY

3.1 System model

The coupled HCUT-GCHP system applied in a hotel building in Harbin is simulated in this paper. The maximum heating and cooling loads are respectively 636kW and 366kW. The accumulated heating load is 1220MWh, far larger than the accumulated cooling load (181MWh).

The system model (You et al. 2014) is built in TRNSYS software (Hellström et al. 1996, Klein et al., 2004.) to simulate the hourly system performance in a long term, as shown in Figure 3. The capacities of main components in system are shown in Table 1. Due to the coupled heating, the system can operate well with 40%~100% boreholes. But, with the number of boreholes reducing from 140 to 56, the capacity of HCUT increases from 250kW to 500kW to keep the heat supply enough at peak heating load.
Table 1. Capacities of main components in HCUT-GCHP systems

<table>
<thead>
<tr>
<th>Items</th>
<th>100% boreholes</th>
<th>80% boreholes</th>
<th>60% boreholes</th>
<th>40% boreholes</th>
</tr>
</thead>
<tbody>
<tr>
<td>Number of boreholes</td>
<td>140</td>
<td>112</td>
<td>84</td>
<td>56</td>
</tr>
<tr>
<td>Rated heat capacity of HCUT (kW)</td>
<td>250</td>
<td>260</td>
<td>350</td>
<td>500</td>
</tr>
<tr>
<td>Rated heat capacity of GCHP unit (kW)</td>
<td>1223</td>
<td>1223</td>
<td>1223</td>
<td>1223</td>
</tr>
</tbody>
</table>

3.2 Case studies

For the coupled HCUT-GCHP systems with different reduced numbers of boreholes, the optimized coupled heating strategies are different to keep the systems at good performance. Several cases with different strategies are studied for systems with different boreholes, as shown in Table 2. The starting temperatures($T_{e0}$) of the coupled heating mode are respectively 0°C, 3°C, 6°C, 9°C, 12°C, 15°C. This is because if $T_{e0}$ is too low, the coupled mode only operates for a short period and the reduction of soil heat extraction is small, and if $T_{e0}$ is high enough, independent GCHP is efficient enough and there is no need to operate coupled heating. For systems with 40% and 60% boreholes, $T_{e0}$ is less than 9°C to avoid the serious soil heat accumulation, which is illustrated in Figure 4.

Table 2. Case studies of systems with different boreholes and operating strategies

<table>
<thead>
<tr>
<th>Numbers of boreholes</th>
<th>Starting temperatures of coupled heating mode($T_{e0}$)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>0°C</td>
</tr>
<tr>
<td>100% boreholes</td>
<td>√</td>
</tr>
<tr>
<td>80% boreholes</td>
<td>√</td>
</tr>
<tr>
<td>60% boreholes</td>
<td>√</td>
</tr>
<tr>
<td>40% boreholes</td>
<td>√</td>
</tr>
</tbody>
</table>

4 RESULTS

To investigate the optimal strategy of coupled HCUT-GCHP with different boreholes, system performance is analyzed in three aspects. Firstly, the system abilities of keeping the annual soil thermal balance and reducing soil heat extraction are illustrated, which are the basic purposes of hybrid HCUT-GCHP. Secondly, taking the system with 60% boreholes as an example, detailed energy consumption and efficiency under different strategy are compared. Thirdly, the different optimal strategies for systems with different boreholes are found out through comparisons.

4.1 Soil thermal balance

The soil thermal imbalance ratios of different systems with different boreholes and different coupled heating strategies are shown in Figure 4. With the increase of soil heat compensation and the reduce of soil heat extraction, the soil thermal imbalance can be effectively eliminated by most coupled HCUT-GCHP, such as systems with 80% or 100% boreholes under different strategies. For the system with 40% and 60% boreholes, the capacity of HCUT should be large to satisfy the heating demand at peaking heating load and the soil heat extraction is greatly reduced. Especially when the starting temperature of coupled heating mode increases, the soil heat injection for cooling becomes even larger than the soil heat extraction for heating. The soil thermal imbalance ratio turn to be far higher than 10%, which means the heat is accumulated in soil, going to the opposite thermal imbalance unfavorable to cooling performance and system energy saving. For the system with 40% boreholes, if the starting temperature of coupled heating mode is higher than 6°C or 9°C, the soil thermal imbalance ratio is about 42% and 74%.
The coupled HCUT-GCHP system can extract heat from air and soil. The percentage of soil heat extraction accounted for the total value from air and soil is demonstrated in Figure 5. When the number of boreholes is reduced or the starting temperature of the coupled heating mode increases, the heat extraction from soil decreases. Taking the system with 60% boreholes as an example, the percentage of soil heat extraction decreases from 44% to 16%, when the starting temperature increases from 0°C to 9°C. For the systems with 40% boreholes, the annual soil heat extraction becomes negative, when the starting temperature is higher than 6°C. It means the HCUT reheats the fluid to a too high temperature, the evaporator outlet fluid temperature is higher than the soil. The borehole is useless under this circumstance, so the strategy is not proper here.

4.2 Efficiencies under different strategies

Taking the system with 60% boreholes as an example, power consumption, hourly system heating COP and average system COP during different period are analyzed to discover the influence of different coupled heating strategies.
COP is always lower than that of normal heating system at the same time, which is shown in Figure 7. But, since less heat is extracted from the soil in coupled heating mode, less heat is needed for compensation. When $T_{e0}$ increases from $0^\circ C$ to $6^\circ C$, the dropping power consumption for heat compensation makes the total system power consumption drop from $565\text{MWh}$ to $533\text{MWh}$. However, if $T_{e0}$ increases from $6^\circ C$ to $9^\circ C$, the total power consumption increases from $533\text{MWh}$ to $544\text{MWh}$. This is because the slightly reduced power consumption for heat compensation is less than the seriously increased power consumption for heating.

The hourly system heating COP and borehole outlet temperature of system with 60% boreholes under $0^\circ C$ and $6^\circ C$ starting temperatures are demonstrated in Figure 7. From the above mentioned system principle, the system with $6^\circ C$ starting temperature has longer coupled heating period than that with $0^\circ C$ starting temperature. For the coupled heating period during which both systems operated, their average system COPs are nearly the same, which is shown in Figure 8. However, for the period that system with $6^\circ C$ starting temperature operates in coupled heating mode and system with $0^\circ C$ starting temperature operates in normal heating mode, the system COPs are quite different as shown in Figure 7.

When the $T_{e0}$ is $6^\circ C$, due to the power consumption of both HCUT and GCHP, the average system heating COP in coupled heating mode is 1.93. At the same time, for the system with $T_{e0}$ of $0^\circ C$, since only GCHP works as normal heating, the average system heating COP is 3.21. Especially when the borehole outlet temperature is high, the system with $0^\circ C$ starting temperature has much higher system heating COP than that with $6^\circ C$ starting temperature.

Average system COPs during different period of coupled HCUT-GCHP under different strategies are shown in Figure 8. When the starting temperature of coupled heating mode increases from $0^\circ C$ to $9^\circ C$, the average system COPs during coupled heating period are nearly the same at 2.63~2.66, while those during the total heating period decreases from 2.91 to 2.45, because the increased coupled heating period of systems with higher $T_{e0}$ has a lower COP, as shown in Figure 7. But the system with high $T_{e0}$ has a low power consumption for heat compensation, as shown in Figure 6. So, the average system COPs during the whole year is the highest (Figure 8), about 2.63 when the starting temperature of coupled heating mode is $6^\circ C$ for system with 60% boreholes.

![Figure 8 Average system COPs during different period under different strategies](image)

![Figure 9 Annual average system COP variations of different systems under different strategies](image)

### 4.3 Optimal coupled heating strategies of different boreholes

The annual average system COP variations of systems with different boreholes under different coupled heating strategies are shown in Figure 9. For the systems with different boreholes, the optimal starting temperature to achieve the highest annual average system COP is different, because the HCUT capacities and the needed heat compensations are different. The optimal starting temperatures of system with 100%, 80%, 60% and 40% boreholes are respectively $9^\circ C$, $9^\circ C$, $6^\circ C$ and $3^\circ C$. The highest annual average COPs of these systems are respectively 2.66, 2.64, 2.63, 2.48.
under the optimal strategies.

It should be noticed that, system with 60% boreholes has a slightly lower system COP, but the number of boreholes is much less, so the cost is less expensive than systems with other numbers of boreholes.

5 CONCLUSION

The performance of HCUT-GCHP in coupled heating mode directly influences the annual soil thermal balance and the system heat supply at peaking load. Therefore, the coupled heating strategy of HCUT-GCHP with different boreholes is optimized in this paper.

For the systems with 80% and 100% boreholes, they keep soil thermal balance well at different starting temperatures of coupled heating mode. However, for the systems with 40% and 60% boreholes, when the starting temperature should be kept at 0°C~3°C and 0°C~6°C respectively.

Taking the system with 60% boreholes as an example, when the starting temperature of coupled heating mode increases from 0°C to 9°C, the power consumption for heating increases by 7.32MWh/°C, while the power consumption for heat compensation decreases by 9.40MWh/°C. Therefore, the system COP reaches highest at 2.63 when the starting temperature is 6°C.

For the systems with 100%, 80%, 60% and 40% boreholes, the optimal starting temperatures of coupled heating mode are respectively 9°C, 9°C, 6°C and 3°C and their annual average system COPs are respectively 2.66, 2.64, 2.63 and 2.48 under the optimal strategies.

ACKNOWLEDGMENTS

The authors gratefully acknowledge the supports of Innovative Research Groups of the National Natural Science Foundation of China (Grant No.51521005) and National Natural Science Foundation of China (Grant No.51638010).

NOMENCLATURE

\[ T_{ei} = \text{Evaporator inlet temperature of GCHP (°C)} \]
\[ T_{e0} = \text{Starting temperature of coupled heating mode (°C)} \]

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Development of Polyolefin Compound and Post-Polymerization Treatments for Ground Heat Exchangers

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ABSTRACT
A ground source heat pump (GSHP) system can be used for both cooling and heating modes simultaneously for commercial, industrial and residential buildings virtually at any location with great flexibility to cover a wide range of demands all around the world. Polyethylene (PE) has been used as the main raw material in production of the Ground Heat Exchangers (GHE). This paper briefly reviews the history of polyethylene and development in polymerization process with emphasis on the third-generation bimodal structure. The characteristics of PE pipes used in GSHP systems are discussed. This paper is devoted to a critical review on the attempts in post-polymerization treatments of the PE, and GHEs to improve the performance of the systems. The experimental and simulated comparisons show that the enhancement of the thermal conductivity of the material can reduce significantly the overall borehole thermal resistance.

INTRODUCTION
Geothermal energy is an energy source known for a long time, but with development of recent materials and techniques has had an improvement due to its renewable nature. Near-surface geothermal energy consists of exploiting the thermal inertia of the ground to gather heat in winter and reject heat to ground in summer. A fluid (e.g. glycol/water mixture) transfers the heat from the ground to the heating system. This geothermal energy method works using a heat pump in which the fluid circulates in the heat system consisting of collector pipes, circulation pumps, plate heat exchangers and refrigerant liquids. The collectors are commonly made of polyethylene (Syndicat, 2012). Studies show that ground coupled heat pump systems are potentially more efficient than traditional air-air heat pump systems (Spitler, Rees, & Yavuzturk, 2000). The most common configuration is the vertical mode, which has the collector pipes in a borehole held either with a backfilling material similar to concrete or without any backfilling material with direct contact to the ground water. The efficiency of the borehole depends on the conductive and convective resistance of the fluid, the pipe, the backfilling material and the ground conditions. One of the important components in the system is the Ground Heat Exchangers (GHE) which are most often made from High Density Polyethylene (HDPE). Many attempts have been made to reduce the thermal resistance of the borehole but only few
have aimed at enhancing the properties of the GHE to diminish borehole resistance. Therefore, this paper briefly reviews the history of polyethylene (PE) and development in polymerization processes with emphasis on the third-generation bimodal architecture. Then, the efforts that have been reported in the literature, to enhance the performance of the GHEs will be discussed and analyzed. Finally, a numerical tool was used to simulate the performance of the GHEs. Dedicated software, Earth Energy Designer (EED) version 3.22 was used to computationally model the evolution of the effective borehole thermal resistance and borehole length (Hellström, & Sanner, 2016).

POLYETHYLENE

PE is the most commonly used polymer around the world in many applications and is the simplest hydrocarbon polymer. It was discovered in 1898 by accident by the German chemist Hans von Pechmann. In 1933, Reginald Gibson and Eric Fawcett discovered the synthesis of industrially practical PE, at the Imperial Chemical Industries in Northwich, England (Visakh & Lüftl, 2016). The most stable staggered conformation is presented in Figure 1. Several types of PE exist as depending on the application. General speaking, there are two macromolecular factors that govern the final properties of a PE compound. The first is the length of the molecule that is proportional to n (Figure 1) and the second is the shape of the molecule for example the presence of side branching. The synthesis of PE has evolved considerably from 1933 to the present with the development of processes, catalysts and additives. These developments led to the synthesis of PE with different lengths of chains, degrees of crystallinity, molecular mass distribution and short or long side branching and, by that technique different properties to accommodate specific needs.

Polymerization process

Polyethylene is produced by reacting gaseous ethylene monomers (CH\textsubscript{2}-CH\textsubscript{2}) to obtain chains of (-CH\textsubscript{2}-CH\textsubscript{2}-\textsubscript{n} with various values of n until over 3,000,000 g/mol (Figure 1). Ethylene is highly reactive and must be of high purity, with very low amounts of moisture, oxygen and other alkenes. This monomer is commonly produced from petroleum but can also be bio-based and made from dehydration of ethanol (for example from sugar beet or sugar cane). Since the polymer forms on the surface of the catalyst in the reactor, the temperature and pressure in the reactor and the type of catalyst govern the final molecular architecture. In fact, the nature of the catalyst and the reactor conditions control the structure as well as, the crystallinity and the mechanical properties of the polymer (Peacock, 2000). Figure 2 shows the three major types of PE. Low density polyethylene (LDPE) is produced by free radical polymerization of ethylene initiated by organic peroxides at high pressure and high temperature. The other types of PE are produced using catalysts under milder conditions.
Figure 2 Schematic microstructure of major types of polyethylene showing main chains, short-chain branches and long-chain branches. Top; Low Density Polyethylene (LDPE) Middle; High Density Polyethylene (HDPE) Bottom; Linear Low Density Polyethylene (LLDPE)

PE grades (regardless of production methods), are usually classified by their density, which also includes the ratio of crystallized (1.00 g/cm³) and amorphous fractions (0.86g/cm³), as illustrated in Figure 3. The scale goes from LDPE with a density of approximately 0.915–0.930 g/cm³ to HDPE with densities around 0.940–0.958 g/cm³ (Malpass, 2010).

Figure 3 Density of the main grades of polyethylene

PIPE APPLICATIONS

The first PE pipes were introduced to the market as early as the 1940’s with the commercialization of the PE materials. Because of the continuous development of PE materials, the efficiency of PE pipes and fittings have been improved considerably. PE pipes for pressure pipe applications are no longer classified by their density, and are now ordered into minimum required strength (MRS) classes based on the international standard ISO 9080. There have been 3 generations of PE pipes. The first PE pipes was produced from LDPE and from Medium Density Polyethylene. These are commonly known as PE 32 and PE 63 in which the digits show the MRS class. The second generation of PE pipe (named PE 80) was made by a unimodal distribution type PE and mostly used for gas distribution. The last and current generation PE pipe is made from a bimodal distribution PE known as PE 100. (Brömstrup, 2009). The polymerization is done using a multi-reactor technology to obtain a tailored material, controlling the molecular weight distribution (Vasile & Pascu, 2005). It is difficult to make a unimodal polymer with a desired set of properties and a shift in the molecular weight or molecular weight distribution (MWD) may improve one property, but may also compromise another. One solution to overcome this problem is to produce PE with a bimodal/multimodal MWD. The goal of bimodal MWD technology is to associate low and high molecular weight components in different concentration by using two series reactors (or multi connected reactors), each of which performs specific sub-process. The low molecular weight fraction is polymerized in the first reactor to bring processability properties and stiffness. This fraction has short un-branched chains with a high crystallinity degree that
provides high density. Then, in the second reactor, a co-monomer is added in the presence of ethylene and hydrogen to produce a co-polymer with low crystallinity, forming in situ blends of the two polymer fractions. The long-branched chains provide low density and stress crack resistance to the polymer.

Bimodal PE is designed to resist slow crack fracture and to withstand any application of high pressure pipe (Brömstrup, 2009). Consequently, these grades permit production of thinner products with the same operating pressure and compare to unimodal PE, production cost is lower. Failure in PE pipes could happen over a long term under low stress (under the yield stress). The resistance to this phenomenon is the environmental stress-cracking resistance or slow crack resistance (SCR). The SCR of bimodal PE is more than three times that of some unimodal PE grades. The high SCR in bimodal PE is due to the chain entanglements obtained with the incorporation of co-monomers in the polymer of the high molecular weight fraction. These entanglement networks affect the overall mechanical behavior under low stress conditions. The failure includes several stages. Microscopic cavities are formed around inhomogeneities such as scratches, pigments, catalyst residues, fillers and then these cavities form the initiation crazing step. It takes the largest fraction of the failure mechanism. Once the deformation zone is formed, crazes propagate by the disentanglement of macromolecules and plastic deformation (yielding) until a crack initiation (Kissin, 2013). Then the crack propagates until material failure. Typical properties of the two grades of PE pipes are shown in Table 1.

<table>
<thead>
<tr>
<th>Characteristic</th>
<th>Standard</th>
<th>PE 80</th>
<th>PE 100</th>
</tr>
</thead>
<tbody>
<tr>
<td>Minimum Required Strength, MRS</td>
<td>EN ISO 9080</td>
<td>8 MPA</td>
<td>10 MPA</td>
</tr>
<tr>
<td>Density at 23°C</td>
<td>ISO 1180</td>
<td>0.94 g/cm³</td>
<td>0.95 g/cm³</td>
</tr>
<tr>
<td>Melt flow rate (MFR)</td>
<td>ISO 1133</td>
<td>±20% RM</td>
<td>±20% RM</td>
</tr>
<tr>
<td>Min. tensile strength</td>
<td>ISO 6259</td>
<td>15 MPa</td>
<td>19 MPa</td>
</tr>
<tr>
<td>Elongation at break</td>
<td>ISO 6259</td>
<td>350 %</td>
<td>350 %</td>
</tr>
<tr>
<td>Oxidation induction time - OIT</td>
<td>ISO 11357-6</td>
<td>&gt;20 min</td>
<td>&gt;20 min</td>
</tr>
<tr>
<td>Hydrostatic strength 20°C, 100 h</td>
<td>EN ISO 1167</td>
<td>10 MPa</td>
<td>12.4 MPa</td>
</tr>
<tr>
<td>Hydrostatic strength 80°C, 165 h</td>
<td>EN ISO 1167</td>
<td>4.6 MPa</td>
<td>5.5 MPa</td>
</tr>
<tr>
<td>Hydrostatic strength 80°C, 1000 h</td>
<td>EN ISO 1167</td>
<td>4 MPa</td>
<td>5 MPa</td>
</tr>
</tbody>
</table>

**POST POLYMERIZATION AND GROUND HEAT EXCHANGERS**

The need for new polymer materials for specific purposes has been grown in recent years because the currently synthesized polymers do not satisfy the increasing demands of many new applications. Though more and more polymers are synthesized they cannot fulfill all the specifications for many fast-growing applications, therefore post polymerization has become an attractive method to fill the gap. Polymer blends and polymer composites are two major approaches for post polymerization modification. Polymer blends are mixtures of structurally different polymers to broaden the range of properties and applications offered by the polymers. In contrast, commercial thermoplastics are commonly compounded with fillers to improve targeted properties, such as rigidity, strength, hardness, dimensional stability, electrical and thermal conductivity or crystallinity. Many particles and fibers have been compounded with polymers, such as metals, carbon fibers, glass fibers, and ceramics, each of which given different qualities and properties for the intended end-use. The most common fillers are minerals such as calcium carbonate, glass spheres, carbon black, clay, mica and tale (Wypych, 2009). Carbon black (CB) has been used as an economical additive in many thermoplastic and thermoset compounds; it is also a very effective additive for the improvement of plastics outdoor stability. It has been shown that for HDPE/CB mixtures, even at a level of 0.05 %-wt CB, the composite has very good UV-resistance and this can be enhanced further by adding up to 5 %-wt CB (White, & Turnbull, 1994). The most commonly used plastics, such as LDPE and HDPE, are considered thermal insulators with low thermal conductivity. There are many new applications such as electronic packaging, pipe networks, heat...
exchangers, and domestic appliances, in which an increase in the heat transfer properties would be an advantage.

Ground source heat pump systems are exceptionally efficient in cooling and heating modes for commercial, industrial and residential buildings all around the world. The amount of heat that can be transferred between the ground and the GHE depends on the thermal conductivity of the ground and on the borehole thermal resistance. The quality of the ground depends on the local geological properties and cannot as such be tuned, but the quality of the boreholes can be engineered to reduce the thermal resistance. The overall borehole thermal resistance can have a significant effect on the system performance and therefore should be kept as low as possible. The overall borehole thermal resistance depends on many factors such as the borehole diameter, the type of backfilling, the convective heat transfer between fluid and GHE, the fluid characteristics, and the position and thermal conductivity of the GHE. The pipe thermal resistance can be expressed under quasi-steady state conditions (Sundén, 2012) by equation 1:

\[ R_{pipe} = \frac{\ln(r_2/r_1)}{2\pi k} \quad (1) \]

Where \( k \) is the pipe thermal conductivity and \( r_2 \) and \( r_1 \) are the pipe outer and inner radius respectively. Figure 4 plots the above relationship for the pipe thermal resistance against the thermal conductivity of the material in the range of polymers-thermal conductivity. As it can be seen from this figure and equation, pipe thermal resistance depends strongly on the thermal conductivity and the standard dimension ratio (SDR, the ratio between pipe diameter and thickness) of the pipe. However, increasing the thermal conductivity of the material above 2 W/m.K, the decrease in pipe thermal resistance is small. This figure also emphasizes that the pipe thermal resistance is a significant portion of the overall borehole resistance so any efforts to decrease the pipe thermal resistance are appreciated. The thermal conductivity of most HDPE grades is about 0.4 W/m.K.

![Figure 4: Pipe thermal resistance vs thermal conductivity of pipe material, top two pipes with SDR 11 and bottom two pipes with SDR 17. The legend shows outside diameter * pipe wall thickness.](image)
To the best of the authors’ knowledge, there are currently two attempts that focus exclusively on enhancing the thermal conductivity of GHE by post-polymerization treatments and compounding. One thermally conductive HDPE material is introduced to the market commercially called Geoperformx (Versaprofiles, 2016), and the other conductive PE composite was patented by MuoviTech Group (Kalantar & Skrifvars, 2013). According to the data sheet and information from Versaprofile, the thermal conductivity of the Geoperformx is about 0.7 W/m.°K. However, in the literature there is no further information about other properties of this material. A thermogravimetric analysis (TGA) was done with a TA instrument Q 500 supplied by Waters LLC, New Castle, IN, USA. Sample of 10 ± 3 mg from the pipe were heated at 10°C/min in a nitrogen purge gas from 25°C to 550°C, the pan was kept isothermal for 3 minutes at this temperature. The sample was then cooled down to 300 °C, meanwhile the gas was switched to oxygen and again raised heating rate 10°C/min to 750 °C. The flow rate for both nitrogen and oxygen gas was 90 ml/min. The weight change as a function of temperature or time was recorded and analyzed. Figure 5 plots weight loss versus temperature and shows that around 8.5 % of conductive particles were added to the PE matrix. Since all the particles have been combusted in the presence of the oxygen, this suggests that the conductive particle might be carbon black.

![Figure 5 Thermogravimetric analysis of the pipe material, weight loss vs temperature](image)

Usually adding particles like carbon black to the polymer to gain specific property can have significant side effects on other properties, (Huang, 2002) such as elongation at break, Charpy impact resistance and long-term properties under stress.

MuoviTech has filed the following two patents: 1) The TurboCollector, (Ojala et al. 2009), an innovation of the collector shape 2) Geothermal Pipe Collector, (Kalantar & Skrifvars, 2013) enhancement of thermally-conductive PE composite.

The TurboCollector innovation, truly produces a unique inline extrusion process and micro fin design inside the geothermal pipe. This process, the die head orients melted material in hoop and axial direction, which vastly increases the pipe strength and toughness (Long et al., 1998; Nie, Bai & Wang, 2010). In addition, twisting the micro fins promotes turbulent flow in the collector. The advantage of this innovation is, to reduce the flow rate with the same heat transfer rate (Acuna, 2010 and Acuna & Palm, 2008).

Kalantar and Skrifvars have published several reports on the mechanical, thermophysical and rheological properties of the thermally-conductive composites in journal articles and conference proceedings (Mejrjerdi et al. 2013; Mehrjerdi et al. 2014; Kalantar, Ojala & Skrifvars, 2015). Per these articles, the thermal conductivity and thermal diffusivity of the thermally-conductive PE composites are 0.7 W/m.°K and 1 m²/s respectively.
Numerical model simulation

There are different numerical tools regarding the dimensioning of borehole heat exchangers, EED was used to simulate heat transfer associated with enhanced thermally conductive PE composites. EED requires the input of the design parameters of the ground, GHE, monthly heat loads and borehole field layout. Results obtained for the enhanced thermally conductive PE composites by Mehrjerdi et al. (2013) were used as input data in EED and the values were compared with the regular HDPE. Four different single U pipe configurations inside the groundwater-filled borehole were investigated with the model to illustrate how concepts can be used. The properties and dimensions (outer diameter* wall thickness in mm) of the pipes are given in Table 2. Synthetic building loads for a normal private house in Sweden were used with an average seasonal performance factor of 3 in the EED. Simulations were done for 25 years.

<table>
<thead>
<tr>
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</tr>
</thead>
<tbody>
<tr>
<td>Case I</td>
<td>0.4 W/m.K</td>
<td>0.255 m²/s</td>
<td>1.728 J/Kg.K</td>
<td>0.953 g/cm³</td>
<td>Ø40 mm*2.3 mm SDR 17</td>
</tr>
<tr>
<td>Case II</td>
<td>0.7 W/m.K</td>
<td>1.026 m²/s</td>
<td>0.562 J/Kg.K</td>
<td>1.204 g/cm³</td>
<td>Ø40 mm*2.3 mm SDR 17</td>
</tr>
<tr>
<td>Case III</td>
<td>0.4 W/m.K</td>
<td>0.255 m²/s</td>
<td>1.728 J/Kg.K</td>
<td>0.953 g/cm³</td>
<td>Ø40 mm*3.6 mm SDR 11</td>
</tr>
<tr>
<td>Case IV</td>
<td>0.7 W/m.K</td>
<td>1.026 m²/s</td>
<td>0.562 J/Kg.K</td>
<td>1.204 g/cm³</td>
<td>Ø40 mm*3.6 mm SDR 11</td>
</tr>
</tbody>
</table>

The output of the simulation program is the borehole length, the effective borehole resistance and mean temperature of the fluid at the end of month for the design period. Simulations were conducted for all four cases with Reynolds number 3000 ±100, and to get comparable results, the effective borehole thermal resistance, pipe thermal resistance and the length of the borehole were determined. The summary of these results are shown in the table 3.

<table>
<thead>
<tr>
<th>Case</th>
<th>Borehole Thermal Resistance</th>
<th>Borehole Length</th>
<th>Pipe Thermal Resistance</th>
</tr>
</thead>
<tbody>
<tr>
<td>Case I</td>
<td>0.1313 m.K/W</td>
<td>147 m</td>
<td>0.0486 m.K/W</td>
</tr>
<tr>
<td>Case II</td>
<td>0.1179 m.K/W</td>
<td>141.2 m</td>
<td>0.0278 m.K/W</td>
</tr>
<tr>
<td>Case III</td>
<td>0.1495 m.K/W</td>
<td>154.8 m</td>
<td>0.079 m.K/W</td>
</tr>
<tr>
<td>Case IV</td>
<td>0.1285 m.K/W</td>
<td>145.8 m</td>
<td>0.0451 m.K/W</td>
</tr>
</tbody>
</table>

Simulations indicate that the effective borehole thermal resistance can be decreased when using thermally enhanced pipe in GHE with a single U bend. The lowest borehole resistance occurs for case II where the pipe with SDR 17 and thermally enhanced material was used. The thermally enhanced pipe reduces the borehole thermal resistance by 10% and 14% for the pipes SDR 17 and SDR 11 respectively. The reduction is greater for SDR 11 because the pipe has thicker wall thickness and the performance of the thermal conductive PE would be more effective. The borehole length required for a given system can be reduced by decreasing the borehole thermal resistance, therefore the length of the GHE can be decreased to reduce the total installation costs.

Raymond et al. (2011) have studied the performance of thermally enhanced pipes in vertical GHE systems. They have shown in a numerical modeling by COMSOL Multiphysics that the enhancement of the thermal conductivity of the pipe reduces the borehole thermal resistance by up to 24% for GHEs made with a single U-bend. They also made several thermal response tests and the measurement of the decrease in borehole thermal resistance associated with the thermally enhanced HDPE pipe is on the order of 20% (Raymond et al., 2011). In fact, these simulations and in-situ thermal response test results are in good agreement with Figure 5. Due to an increase in the
thermal conductivity of the material, the pipe thermal resistance reduces and accordingly the overall borehole resistance reduces.

CONCLUSION

Polyethylene with its broad spectrum of mechanical and thermophysical properties can be used in many applications. The key to its adaptability is due to its tuneable semicrystalline morphology, which can be controlled by the polymerization conditions and methodology. Research and development continue both the polymerization phase and post-polymerization processes with the goal of tailoring resins to meet the requirements of more specialized markets. The tailoring of polyethylene resins can be achieved either during polymerization, by regulating reaction conditions to affect branching and molecular weight, or by post-polymerization treatments such as compounding or blending. Good tensile strength, corrosion resistance, stiffness and long term slow crack resistance of new bimodal HDPE are desirable pipe attributes for water, sewer and natural gas distributions. Development efforts were not restricted to the polymerization processes; various fillers and fibers were added to the polymer to change the properties of the material. Only two efforts to enhance the thermal conductivity of the GHEs have been reported. Numerical analysis and in situ thermal response tests show that the enhancement of the thermal conductivity of the material can reduce significantly the overall borehole thermal resistance and BHE length. TurboCollector innovation of twisted fins improves the COP of the GSHP system and efficient energy consumption.

ACKNOWLEDGEMENTS

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Research on the Prediction and Analysis of Solar-assisted Ground-coupled Heat Pump System

Zhaoyi Zhuang   Yongbin Li  Wentao Sun    Xianye Ben     Xin Zhang

ABSTRACT
Finding the most suitable setting parameters to design a solar-assisted ground-coupled heat pump system can provide help for the practical engineering application. However, the performance of heat pump after years of operation cannot be predicted because it is just ideal prediction with formula of current study. To address the problem, 2000 sets of performance data of solar-assisted GCHP systems operating over 20 years were simulated by professional software. Then, the classification and regression tree (CART) method is adopted to predict the design of collector area, and the multi-linear regression is adopted to predict average monthly per meter borehole heat exchange. Seasonal factor decomposition and exponential smoothing are adopted to analyze the average monthly temperature of the circulating fluid, circulating fluid inlet and outlet temperatures of the heat pump after 20 years when we perform the time series prediction. Experimental results demonstrate that CART, multi-linear regression, seasonal factor decomposition and exponential smoothing are promising for practical applications.

KEYWORDS: SOLAR-ASSISTED GROUND-COUPLED HEAT PUMP SYSTEM; CLASSIFICATION AND REGRESSION TREE; MULTI-LINEAR REGRESSION; SEASONAL FACTOR DECOMPOSITION; EXPONENTIAL SMOOTHING

1. INTRODUCTION
Nowadays, the environment and resources have become key factors in the development of every country around the world. Mineral resources are running out along with the worsening environmental pollution, therefore, we urgently need to seek for a new kind of green, pollution-free alternative energy source. The development and deployment of ground-coupled heat pump (GCHP) [1] is a perfect solution to this problem. The system performance degrades if the GCHP system installed in heating-dominated buildings rejects more heat extraction from the ground than into the ground. The annual accumulated heat makes the temperature of water entering the heat pump lower, and the possible solution is to increase the spacing between boreholes or use strip type and block layout with a high level of initial investment to eliminate this thermal imbalance [2]. Therefore, Trillat-Berdal et al [3] presented an experimental study of a GCHP combined with thermal solar collectors with the advantage of balance of the ground loads, longer operating time of the solar collectors and avoidance of overheating. Chen and Yang [4] designed and numerically simulated a solar assisted GCHP system in northern China, which showed that this optimal design reduced the borehole length of 3.9m/m² and the designed system efficiency could be 3.55 with 36% annual space heating solar fraction and 75% annual domestic hot water solar fraction. Wang et al. [5] also presented the experimental study of a solar-assist GCHP system with solar seasonal thermal storage added so as to balance the ground load in severe cold areas. The efficiency, cost-effectiveness and durability of the heat pump dictate successful acceptance of any new building heating
and/or cooling technology. In order to assess building energy performance, researchers have worked on various data mining techniques such as adaptive neuro-fuzzy inference systems (ANFIS) [6], Lin-kernel support vector machine (SVM) [7], iteratively reweighed least squared (IRLS) [8], random forest (RF) [8], multiple regression model [9], multivariate adaptive regression splines (MARS) [10] and artificial neural network (ANN) [11]. In our previous work [12], we have used Partial Least Squares Regression (PLSR), Support Vector Regression (SVR) and M5 Model Tree to predict the heat transfer performance for the GCHP system. However, there has been little work on predicting the performance of solar-assisted ground-coupled heat pump systems.

Finding the most suitable setting parameters to design a solar-assisted ground-coupled heat pump system can provide help for its practical engineering application. However, the performance of heat pump after years of operation cannot be predicted because it is just ideal situation prediction with formula of current study. To address the problem, 2000 groups of simulation data of solar-assisted GCHP systems operating for over 20 years were created by means of simulation with a professional software GeoStar. The simulation has considered a number of input physical quantity, including drilling vertical spacing, drilling column spacing, drilling radius, drilling geometry arrangement, drilling nominal external diameter, U-tube spacing, thermal conductivity coefficient, drilling depth, number of drilling, ground temperature, ground thermal conductivity, circulating fluid parameter, collector type, collector efficiency, collector installation angle, heat loss efficiency and surface albedo, while the design and performance outputs such as collector area, average monthly per meter borehole heat exchange, average monthly temperature of the circulating fluid, circulating fluid inlet and outlet temperatures of the heat pump, are obtained. Then, the classification and regression tree (CART), multi-linear regression, and time series analysis are used to predict the performance of solar-assisted ground-coupled heat pump system in a more general sense.

2 DATA

The data used in this paper are simulated by a software, and there are 2000 sets of performance data in total. Each item includes pump working condition parameters with 17 dimensions, collector area $Y_1$, average monthly per meter borehole heat exchange $Y_2$, average monthly temperature of the circulating fluid $Y_3$, circulating fluid inlet temperature of the heat pump $Y_4$ and circulating fluid outlet temperature of the heat pump $Y_5$. $Y_6$, $Y_7$, $Y_8$ and $Y_9$ are all time series data for 240 months (20 years).

The system working condition parameters includes drilling vertical spacing $X_1$, drilling column spacing $X_2$, drilling radius $X_3$, drilling geometry arrangement $X_4$, drilling nominal external diameter $X_5$, U-tube spacing $X_6$, thermal conductivity coefficient $X_7$, drilling depth $X_8$, number of drilling $X_9$, ground temperature $X_{10}$, ground thermal conductivity $X_{11}$, circulating fluid parameter $X_{12}$, collector type $X_{13}$, collector efficiency $X_{14}$, collector installation angle $X_{15}$, heat loss efficiency $X_{16}$ and surface albedo $X_{17}$. $X_4$, $X_6$, $X_{11}$ and $X_{12}$ are catalog, and their introduction can be found in Ref. [12]. $X_{13}$ is also catalog, and Table 1 provides the detailed collector type $X_{11}$ of different materials in this study. Table 2 provides descriptive statistics of these 17 system working condition parameters.

<table>
<thead>
<tr>
<th>Table 1. Collector type $X_{11}$</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Nominal value</strong></td>
</tr>
<tr>
<td>1</td>
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<tr>
<td>2</td>
</tr>
<tr>
<td>3</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Table 2. Statistics of these 17 pump working condition parameters</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Parameters</strong></td>
</tr>
<tr>
<td>-----------------</td>
</tr>
<tr>
<td></td>
</tr>
<tr>
<td><strong>Average</strong></td>
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<tr>
<td><strong>Standard deviation</strong></td>
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<tr>
<td><strong>Minimum</strong></td>
</tr>
<tr>
<td><strong>Maximum</strong></td>
</tr>
<tr>
<td><strong>Range</strong></td>
</tr>
<tr>
<td>Parameters</td>
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<td>-----------</td>
</tr>
<tr>
<td></td>
</tr>
<tr>
<td>Average</td>
</tr>
<tr>
<td>Standard deviation</td>
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<tr>
<td>Minimum</td>
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<tr>
<td>Maximum</td>
</tr>
<tr>
<td>Range</td>
</tr>
</tbody>
</table>

Figure 1  A sample of outputs

It can be seen from Fig.1 (1) that the data \( Y_1 \) has a strict periodical change, because the simulation is performed under an ideal condition. It can be seen that average monthly temperature of the circulating fluid \( Y_1 \) declines gradually, and the general trend between each year remains unchanged. It cannot be judged intuitively whether there is a change in the value from Fig.1 (3) and (4), but it can be seen that the trend of each year remains unchanged. So the prediction can be performed according to the change in the trend observed by performing seasonal factor decomposition.

3. METHODOLOGIES

3.1 Classification and regression tree (CART)

CART can not only create classification tree and regression tree, but also model tree. Moreover, when used in regression, CART firstly samples continuous numerical variables, then the data discretization, nodes selection and trees creation are performed afterwards. Compared with classification tree, the regression tree is merely different in node selection and feature reservation.

3.2 Time Series analysis

Time series refers to a group of data sorted by chronological order, the methods used in processing time series in this paper are seasonal decomposition, exponential Smoothing, etc.

3.2.1 Seasonal factor decomposition. The relevant temperature data, strongly influenced by the weather, is a seasonal periodical time series. Generally speaking, seasonal time series have following components: tendency (T), cycle(C), seasonality(S), irregular variations (I). Additive seasonal decomposition model can be expressed as: \( Y_t = T_t + S_t + C_t + I_t \), and a term-by-term separation is as follows:

1. Trend item: moving average prediction method

The observed value of time series is denoted as \( y_t, t=1, 2, 3, \ldots, N \), and the moving average value is calculated by

\[
M_t = \frac{y_t + y_{t-1} + \cdots + y_{t-n+1}}{n}, \quad t = 1, 2, 3, \ldots, N, \quad n \geq 2,
\]

consequently, the predicted value for \( t+1 \) would be \( \hat{y}_{t+1} = M_t \), therefore, the time series without trend term is obtained.

2. Seasonal factor item: trend extrapolation separation prediction method
Firstly we assume that the time series contains no trend item, so its model can be written as \( Y = S + C + I \), because the effect of periodical component \( C \) is subtle, so it can be integrated into stochastic error component \( I \). When separating the seasonal factor item, the periodical item and stochastic term can be viewed as a whole, and the stochastic error component is major. Then, the monthly average of \( Y \) is calculated and after subtraction the seasonal factor item \( S \) without stochastic error component is obtained.

3 Cycle item: periodogram method

Assuming that time series contains neither trend term nor seasonal term. Firstly, it is decentralized as \( x_i' = x_i - \frac{1}{N} \sum_{i=1}^{N} x_i \), then it can be written in the forms of Fourier Series. Denote \( A_\tau, B_\tau \) as Fourier coefficients, \( \tau \) as the period, and it is set that \( S_\tau^2 = A_\tau^2 + B_\tau^2 \), and there will be a periodical change in the data. The concrete computation process is as follows: the variance \( \sigma^2 \) of \( x_i \) is calculated, therefore, the periodic vibration can be formulated as \( \hat{C}_\tau(t) = C_\tau \sin \left( \frac{2\pi t}{\tau} + \phi_\tau \right) \). Moreover, multiple period calculations can be performed.

4 Stochastic error item

After having separated trend item, seasonal factor item and cycle item from the series, what is left is the stochastic error item.

### 3.2.2 Exponential Smoothing

Exponential smoothing can eliminate the effect of the statistic values by the weighted sum of the past values and present values. With the observation value getting away, its weight shows a corresponding exponential decrease. There are several conditions in exponential smoothing, and following is a brief introduction.

1 Single Exponential Smoothing

The formula is as follows: \( F_{t+1} = \alpha x_t + (1-\alpha)F_t \), where \( F_t \) represents the predicted value in a former period. The essence of exponential smoothing prediction model is that the predicted value in a certain period is weighted average of former predicted value and actual value. The range of weighting coefficient \( \alpha \) is \( 0 < \alpha < 1 \). The weight is obtained by decreasing at a geometric progression, further from the predicted value, smaller the weight and vice versa.

Generally there are two methods in selecting initial values: \( \square \) Use the actual value of the first period as the initial value directly. \( \square \) Use the average value of primary several periods as the initial value.

2 Double Exponential Smoothing can be formulated as \( F_{t+2} = a_t + b_T \), where \( T \) represents the number of periods between the present period and predicted period.\( a_t = 2S^{(1)}_t - S^{(2)}_t, \ b_t = \frac{\alpha}{1-\alpha}(S^{(1)}_t - S^{(2)}_t) \), and \( S^{(1)}_t \) is the single exponential smoothing value, \( S^{(2)}_t = ax_t + (1-\alpha)S^{(1)}_{t-1} \), \( S^{(2)}_t \) is the double exponential smoothing value, \( S^{(2)}_t = ax_t + (1-\alpha)S^{(2)}_{t-1} \).

3 Holt-winters additive exponential model can be formulated as \( F_{t+2} = a_t + b_T + I_{t+2} \), where \( a_t = \alpha(x_t-I_{t-1}) + (1-\alpha)(a_{t-1} + b_{t-1}) \), \( b_t = \beta(a_t - a_{t-1}) + (1-\beta)b_{t-1} \), \( I_t = \gamma(x_t - a_t) + (1-\gamma)I_{t-1} \), \( a_t + b_T \) is a trend factor, \( a_t \) denotes intercept, \( b_t \) denotes slope. \( I_t \) denotes seasonal factor, \( L \) is the length of the season and \( t + T \) denotes the period to be predicted. The value ranges of \( \alpha, \beta \) and \( \gamma \) are all between 0 and 1.

### 4 SIMULATION AND CALCULATION

#### 4.1 Performance Prediction on \( Y_t \)

Generally speaking, when establishing and training the prediction model, addition of irrelevant features will reduce the prediction performance. Moreover, a wide difference between the value ranges of different features will also affect the regression prediction. Based on reasons above, we mainly adopt feature selection and normalization to preprocess the original data.

4.1.1 Feature Selection. The procedure of feature selection is a procedure that investigate all the features to eliminate irrelevant ones, and the standards for investigation varies, now, data used in this paper are mainly investigated in two aspects, variance and correlation.
Variance:  
\[ S^2 = \frac{1}{N} \sum_{i=1}^{N} (x_i - \overline{x})^2, \]  
where \( \overline{x} \) is mean of a set \( \{x_i, i=1,L ,N\} \), and \( N \) is the size of the set.

Pearson correlation coefficient:  
\[ r = \frac{1}{N-1} \sum_{i=1}^{N} \frac{(x_i - \overline{x})(y_i - \overline{y})}{S_x S_y}, \]  
where \( S_x \) and \( S_y \) are standard deviations of \( \{x_i, i=1,L ,N\} \) and \( \{y_i, i=1,L ,N\} \), and \( \overline{x} \) is mean of a set \( \{x_i, i=1,L ,N\} \). Pearson Correlation Coefficient ranging in value from -1 through +1, +1 means that there is a linear positive correlation between two variables, -1 means that there is a linear negative correlation, and 0 means irrelevance.

Firstly, the variances of 17 variables are calculated and the threshold, as shown in Fig.2, was set to 0.1. Meanwhile, any feature whose variance is lower than 0.1 among the 2000 samples will be eliminated for it cannot show sufficient differences and has little help in establishing the model. Feature \( X_3 \) and \( X_{17} \) are eliminated during the regression prediction of \( Y_1 \) for smaller variances.

4.1.2 Normalization. Firstly, the two features with smaller variance are eliminated and then the dimensionality of features is reduced to 8 (remaining \( X_1, X_2, X_4, X_5, X_{12}, X_{14}, X_{15}, X_{16} \)) from 15 by calculating and comparing the Pearson correlation coefficient. Dimensionality reduction has not only reduced the workload in establishment of prediction model, but also can improve its performance. Then, the data is normalized by obtaining the average and standard deviation of the remaining 8 feature variables, therefore avoiding harmful effects may caused by a wide difference between value ranges.

After the normalization of \( Y_i = \frac{X_i - \overline{X}}{S_X} \), the average of variables will reach around 0, and standard deviation approaches 1. CART is used to make regression prediction of \( Y_i \), the average error is 3.031, the error rate was 0.97%, the RMSE is 4.044. The red curve shown in Fig.4 is the actual value of \( Y_i \), and the green curve is the predictive value of \( Y_i \).

Figure 2 Feature Variance Histogram  
Figure 3 Histogram of Each Feature Pearson Correlation Coefficient  
Figure 4 Predictive Results of Collector Area \( Y_i \)
4.2 Performance Prediction on \( Y_2 \)

Using 12 months as a period for \( Y_2 \), multiple linear regression and least square method are adopted. The prediction expression of \( Y_2 \) for using the former 1600 groups of data as a training set and remaining 400 ones as a testing set is:

\[
Y_2 = -15.972 + 0.002X_1 + 0.001X_2 + 0.359X_3 + 0.009X_4 + 0.002X_5 - 0.005X_6 + 0.003X_7 + 0.001X_8 + 0.002X_9 + 0.02X_10 - 0.253X_{11} 
\]

The result is \( F=754.284 \), significance SIG=0.000. So it is judged at a probability over 99.9% that arguments \( X_1, L, X_{17} \) all have a significant effect on dependent variable \( Y_2 \) and the RMSE is 0.006, quite ideal. Then each of results on 12 months can be obtained in turn, and the relationship between average monthly meter borehole heat exchange of all 12 months and 17 inputs are formulated as \( Y_2 = AX + \beta \)

\[
Y_2 = \begin{bmatrix} Y_2^1 \\ Y_2^2 \\ \vdots \\ Y_2^{12} \end{bmatrix}, X = \begin{bmatrix} X_1 \\ X_2 \\ \vdots \\ X_M \end{bmatrix}, \beta = [-15.972, -10.942, -5.161, 6.281, 9.025, 8.461, 6.793, 7.060, 7.254, 4.010, -6.242, -14.701] 
\]

where \( A = \begin{bmatrix} 0.002 & 0.001 & 0.359 & 0.009 & 4.476E-5 & 1.514E-5 & 0.002 & -0.001 & 0.003 & -2.697E-5 & 0.001 & -2.957E-5 & 7.082E-5 & 0.020 & 0.000 & -0.253 \\ 0.002 & 0.000 & 0.302 & 0.006 & 3.608E-5 & 1.511E-5 & 0.001 & -0.004 & 0.001 & 1.906E-5 & 0.001 & 1.915E-5 & 7.244E-5 & 0.013 & 0.000 & -1.181 \\ 0.001 & 0.001 & 0.035 & 0.003 & 3.209E-5 & -1.469E-5 & 0.001 & -0.002 & 0.000 & 3.662E-5 & 0.001 & 0.000 & 6.315E-5 & 0.007 & 0.000 & -0.079 \\ 0.000 & 0.000 & 0.000 & -0.001 & 2.732E-5 & 0.000 & 0.001 & 0.002 & 0.000 & 0.000 & 0.001 & 0.000 & 0.000 & 0.000 & 0.000 \\ -0.002 & 0.000 & -0.349 & -0.001 & -3.480E-5 & -0.003 & 0.000 & 0.000 & 0.000 & 0.000 & 0.000 & -0.001 & 0.000 & 0.000 & 0.000 \\ -0.003 & 0.001 & -0.067 & -0.003 & 8.448E-6 & 0.000 & 0.000 & 0.000 & 0.000 & 0.000 & 0.000 & 0.000 & 0.000 & -0.001 & 0.000 \\ -0.004 & -0.002 & -0.457 & -0.005 & 2.120E-5 & -0.001 & 0.000 & 0.000 & 0.000 & 0.000 & 0.000 & 0.000 & 0.000 & 0.000 & 0.000 \\ -0.003 & -0.001 & -0.380 & -0.003 & 2.947E-5 & -0.001 & -0.004 & 0.004 & 0.001 & 0.000 & 0.000 & 0.000 & 0.000 & 0.000 & 0.000 \\ 0.006 & 2.973E-5 & -0.040 & -0.004 & 3.917E-5 & -3.998E-5 & 0.000 & 0.002 & -0.002 & 0.000 & 0.000 & 0.000 & 0.000 & 0.000 & 0.000 \\ 0.002 & 0.001 & 0.274 & -0.002 & 2.992E-5 & 0.000 & 0.002 & 0.002 & 0.009E-6 & -1.090E-5 & 0.002 & 1.071E-5 & 0.000 & -3.454E-5 & 0.014 & 0.000 \\ 0.001 & 0.001 & 0.382 & 0.004 & 4.065E-3 & 0.000 & 0.000 & 0.002 & -0.004 & 0.000 & -7.226E-5 & 0.002 & 0.000 & 7.343E-5 & 0.001 & 0.000 \\ 0.001 & 0.001 & 0.384 & 0.008 & 4.850E-5 & 0.000 & 0.000 & 0.002 & -0.003 & 0.000 & -5.767E-5 & 0.001 & 0.002 & -2.502E-5 & 7.773E-5 & 0.022 \end{bmatrix} 
\]

4.3 Performance Prediction on \( Y_3 \)

4.3.1 Seasonal factor decomposition. As can be seen from Section 2, \( Y_3 \) shows a cyclical downtrend. We can analyze by the seasonal factor decomposition and Fig.5 shows the seasonal factor decomposition results. The effect of seasonal factor item on time series is fixed and remains unchanged every year. The error term is a random value around 0. Due to that the appearance of error is random, we mainly explore the trend circulation item in time series.

As we can see from Fig 5(3), the trend circulation item shows a fluctuant downtrend.

\begin{figure}
\centering
\includegraphics[width=\textwidth]{Fig5.png}
\caption{Results of seasonal factor decomposition of \( Y_3 \)}
\end{figure}

4.3.2 Time series analysis of \( Y_3 \). We use exponential smoothing method to predict the result after 20 years. The result respectively figured by (1) Simple non-seasonal model (Single exponential smoothing), (2) Simple seasonal model (Second exponential smoothing method) and (3) Holt-winters additive model (Triple exponential smoothing) shows in Fig.6.
It can be seen that if we use a smoothing method, it is not able to predict the time series, and the predicted value is a straight line. Table 3 lists $R^2$, root-mean-square error (RMSE), Mean absolute error percentage (MAPE), maximum absolute error percentage (MaxAPE), mean absolute error (MAE), maximum absolute error (MaxAE) and Standardized BIC.

**Table 3. Various statistical indexes**

<table>
<thead>
<tr>
<th>Method</th>
<th>$R^2$</th>
<th>RMSE</th>
<th>MAPE</th>
<th>MaxAPE</th>
<th>MAE</th>
<th>MaxAE</th>
<th>Standardized BIC</th>
</tr>
</thead>
<tbody>
<tr>
<td>Second Exponential Smoothing Method</td>
<td>0.9</td>
<td>0.03</td>
<td>0.257</td>
<td>0.689</td>
<td>0</td>
<td>0.074</td>
<td>-6.973</td>
</tr>
<tr>
<td></td>
<td>93</td>
<td>1</td>
<td>26</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Holt-winters Additive Model</td>
<td>0.9</td>
<td>0.03</td>
<td>0.226</td>
<td>0.634</td>
<td>0</td>
<td>0.070</td>
<td>-6.973</td>
</tr>
<tr>
<td></td>
<td>94</td>
<td>0</td>
<td>25</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

Simple seasonal model (Second exponential smoothing method) predicts the time series successfully, and the value of $R^2$ is 0.993. Holt-winters additive model (Triple exponential smoothing) also predict the time series successfully, and we can see the value of $R^2$ is 0.994. As can be seen from the above methods, the Holt-winters additive model gets the maximum $R^2$ and the best prediction.

### 4.4 Performance Prediction on $Y_4$

4.4.1 Seasonal factor decomposition. As can be seen from Section 2, $Y_4$ basically changes circularly and periodically, and the specific trend can be judged by seasonal decomposition.

As can be seen from Fig.7, after seasonal decomposition, the effect of Seasonal factor item on time series is fixed and remains unchanged every year. The error term is a random value around 0. As we can see from the Fig.7 (3), the trend circulation item also shows a fluctuant declining trend.

4.4.2 Time Series Analysis of $Y_4$. The prediction of $Y_4$ is performed by the Winters additive model in exponential smoothing method, and its result is shown in Fig.8.
It can be seen from Fig. 8 that this Winters additive model has successfully predicted the time series $Y_4$, the Table 4 provides all the same statistical indexes as Table 3, and the value of $R^2$ is 1, quite ideal.

### 4.5 Performance Prediction on $Y_5$

**4.5.1 Seasonal Factor Decomposition.** The results of seasonal factor decomposition for $Y_4$ are shown in Fig. 9. The results are similar with that of $Y_4$.

**4.5.2 Time Series Analysis of $Y_5$.** The prediction of $Y_5$ is performed by the Winters additive model in exponential smoothing method, and its result is shown in Fig. 10 and all statistical indexes of the prediction are listed in the Table 5.

![Figure 8. Predictive results of by Time series analysis](image)

![Table 4. Various statistical indexes](image)

<table>
<thead>
<tr>
<th>Model</th>
<th>$R^2$</th>
<th>RMSE</th>
<th>MAPE</th>
<th>MaxAPE</th>
<th>MAE</th>
<th>MaxAE</th>
<th>Standardized BIC</th>
</tr>
</thead>
<tbody>
<tr>
<td>Holt-winters Additive Model</td>
<td>1</td>
<td>0.055</td>
<td>0.497</td>
<td>15.729</td>
<td>0.033</td>
<td>0.582</td>
<td>-5.740</td>
</tr>
</tbody>
</table>

The results of Seasonal Factor Decomposition of $Y_5$ are shown in Fig. 9.

![Figure 9. Results of Seasonal Factor Decomposition of $Y_5$](image)

![Figure 10. Predictive results of by Time series analysis](image)
Table 5. Various statistical indexes

<table>
<thead>
<tr>
<th>Holt-winters Additive Model</th>
<th>$R^2$</th>
<th>RMS</th>
<th>MAPE</th>
<th>MaxAPE</th>
<th>MAE</th>
<th>MaxAE</th>
<th>Standardized BIC</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>1</td>
<td>0.055</td>
<td>1.442</td>
<td>59.417</td>
<td>0.032</td>
<td>0.594</td>
<td>-5.745</td>
</tr>
</tbody>
</table>

CONCLUSIONS

The collector area has been triumphantly predicted by feature Selection based on variance and Pearson correlation coefficient together with CART with the average error of 3.031, the error rate of 0.97% and the RMSE of 4.044. The average monthly per meter borehole heat exchange has been predicted by multi-linear regression, and the relationship between average monthly per meter borehole heat exchange of all 12 months and 17 inputs are formulated as a matrix product form. The average monthly temperature of the circulating fluid, circulating fluid inlet temperature of the heat pump, circulating fluid outlet temperature of the heat pump has been successfully analyzed by seasonal factor decomposition, and their performances have been estimated by exponential smoothing. Experimental results demonstrate that the CART, multi-linear regression, seasonal factor decomposition and exponential smoothing are promising for practical applications in predicting performances of the solar-assisted GCHP systems.

ACKNOWLEDGEMENT

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ABSTRACT

We propose new granule type grouting materials with optimized ratio of bentonites and sands for use in the ground source heat pump systems. The grouting materials could be prepared in various size of 1.18 to 4.75 mm. The granules consist of bentonite powders and sand particles inside and extra bentonite powders exists on the surface of granules. In different ratio (1:0-1:8) of bentonites and sands, the granules with 1:8 ratio show the best thermal conductivity (ca., 1.8 W/mK). In comparison to the existing powder type mixed grouting materials, the present granules show the lower values of fluid loss and the minimum values at 1:5 ratio of bentonites and sands. This might be due to the fact that the bentonites on the surface of granules strongly hold water, resulting in decrease of penetration of water into granules. We also investigated the effect of bentonites between the granules on fluid loss property by addition of them as slurry form. When only water is added, water channels were formed. When the slurry of bentonite and water is added, water channels were decreased, resulting from that the additional bentonites interacted with granules. According to aging time, the homogeneity of grouting layer is improved.

INTRODUCTION

Bentonite, discovered near Fort Benton region in Montana of United States in 1898 and named officially by Wilbur C. Knight, is one of the clay minerals. It has the incredible swelling ability in water and its volume can increase as much as about 20 times of its original volume. Since then, the bentonite has been well known for a brand name. Bentonite is part of the smectite class of clays that are less than 2 micrometers in largest dimension. It is a layered aluminum phyllosilicate, which consists mostly of the mineral montmorillonite produced in the Cretaceous period. (Ingleethorpe 1993) Nowadays, bentonite is used in different fields such as casting, engineering works, paper-making, feedstuff, and agriculture because it has various physico-chemical properties such as high swelling, high surface area, high ion exchangeability, high water adsorption, high surface charge and so on. (Ko 2000)

The grouting layer plays very important role in determining the overall efficiency of ground source heat pump systems due to the direct heat transfer from the ground materials to the heat exchangers. Among various grouting materials, bentonite has been widely used in ground source heat pump systems, because it is capable of not only holding water tightly due to its self-swelling property, but also preventing the migration of heat transfer layer due to its...
high viscosity (Park 2006). Nevertheless, the bentonite has a fatal weakness of low thermal conductivity (Salomone 1989). To improve it, sands have been utilized as an additive for increasing thermal conductivity. However, when the sands are added to bentonites, the excess contents of sands (ca., > 30%) cause several problems such as the separation between bentonites and sands, the increase of fluid loss, and the failure of slurry pump blades. Therefore, it is known that sands of less than 30% are suitable for use in actual grout slurry systems (Sohn 2006).

![Figure 1](image1.png) Concept of the mixed grout granules of bentonites and sands

In this work, we tried to overcome the problems mentioned above by forming the sands pre-mixed granules of bentonites as shown Figure 1. The effects of various factors, such as ratio of bentonites and sands, diameter of granules, and use of water or bentonite slurry, on thermal conductivity and fluid loss properties of the grouting materials prepared were systematically investigated. In addition, we revealed visually the change of microstructures of the mixed granule grout by using an optical microscope.

**EXPERIMENTS**

**Specimen Preparation**

As mentioned in introduction, bentonite can be easily swelled in water with plasticity. Using these properties, we prepared mixed granules by spraying water to bentonites and sands, and centrifugal force. Mixed ratio of bentonites (Barotherm Gold, Baroid Industrial Drilling Products Co) and sands (local product) was adjusted in the range of 1:0-1:8. The five different size granules from 1.18 to 4.47 mm were prepared by sieving separation (see Figure 2).

![Figure 2](image2.png) Mixed grout granules of bentonites and sands
Thermal Conductivity Measurements

Thermal conductivity of bentonite was measured as a paste forms using non-steady-state probe (TP02, Hukseflux Co.) having two K-type thermocouples and a reference RTD (① in Figure 3) inside. The measured data were collected using data logger (④ in Figure 3, CR 10X, Campbell Co.) with automatic scanning program (② in Figure 3, Loggernet, Campbell Co.), and were calculated to thermal conductivity by software package (Hukseflux Co.). The measurements of thermal conductivity were performed in a temperature controlled chamber to maintain constant temperature of 25 °C.

Fluid Loss Measurements

Fluid loss test represents the water storage ability of bentonite. In the ground source heat pump systems, grouting layer plays an important role in protecting heat transfer layer from surface water or underground water. Typically, the fluid loss properties are tested by measuring amount of water passing through filter press paper in cylinder under 7 atm. of nitrogen gas pressure until no water drop according to test method used in the companies generally. When amount of water become lower, water storage ability becomes better. In this work, we prepared two kinds of paste form specimens to reveal the effect of bentonites near granule boundary on fluid loss; (1) mixed granules + water, and (2) mixed granules + bentonites/waters slurry. Figure 4 shows the measurement apparatus (Filter Press API, Fann Co.) for fluid loss properties.
RESULTS AND DISCUSSION

Thermal Conductivity Properties

Figure 5-a) shows thermal conductivity change as a function of mixed ratio of bentonites and sands. Thermal conductivities of the present mixed materials show gradual increase with increase of sand content. This result shows similar to the previous results shown in existing powder-typed grouting materials. (Baroid Industrial Drilling Products, 2011) Also, we investigated thermal conductivity change as a function of granule sizes of mixed specimens with ratio of 1:3 (bentonite/sand). As plotted in Figure 5-b), thermal conductivity change on the different granule sizes shows below 10% in the range of 1.18 and 4.75 mm, suggesting that the granule size does not affect thermal conductivity properties.

Fluid Loss Properties

Figure 6 shows fluid loss property change as functions of a) ratio of bentonite/sand and b) granule sizes, respectively. As can be seen in Figure 6-a), fluid loss of the existing powder-typed grouting materials increase as increase of the amount of sands. This phenomenon means the decrease of water preventing performance as increase of the amount of water passed into bentonite layers due to high sand content. Interestingly, however, the mixed granules grouting materials show the gradual decrease of fluid loss in the range from 1:0 to 1:4 of bentonite/sand, and increase from ratio of 1:5. In order to find these reasons, microstructures of specimens were closely investigated. As can be seen in Figure 7, sands exist in granule inside and excess bentonites are placed on the surface of granules. In cases of over ratio of 1:5, it is thought that the bentonite content on the surface of granules is not enough and thus water channels gradually increase. Being used bentonite-water slurry instead of water, excess bentonites play a role in decreasing water channels. Therefore, it can be seen that the rate of increase of fluid loss is low. Also the fluid loss change on the different granule sizes shows below 10% in the range of 1.18 and 4.75 mm as shown in Figure 6-b), suggesting that the effect of granule sizes on fluid loss is relatively small. However, these values do not represent the actual circumstances in boreholes because the pressure is changed with depth mainly due to hydrostatic pressure differences, but it plays an important role as a comparative basis of water storage ability for grouting materials.
Figure 6  Fluid loss change with a) mixed ratio of bentonite/sand, b) granule sizes for mixed ratio 1:3 and 1:8 specimens of mixed grout granules

Microstructures

Figure 7 shows the photographs of fine structure of mixed granules prepared as a paste forms. When only water was added, water channel formed as shown in Figure 7-a). When water was used with bentonite, it could be seen that forming water channel decreased as shown in Figure 7-b). This can be explained by the fact that the granules are surrounded by bentonites and thus water is difficult to pass them. Consequently, the number of water channel formed between granules plays a role in determining the fluid loss ability. Meanwhile, it was not observed that there is separation phenomenon by adding excess sands.

Figure 7  Microstructures of mixed grouting granules added a) only water, b) slurry of bentonites and waters
Moreover, we monitored a change of microstructures of mixed granules according to aging time. As shown in Figure 8, it is gradually stabilized. In this reason, we can expect that the paste forms of mixed granules became stabilize, and forms uniform heat transfer layer with time.

![Figure 8](image.png)

**Figure 8** Stabilization process for mixed grout granules added slurry of bentonite and water

**CONCLUSION**

In this study, we investigated thermal conductivity and fluid loss properties of new grouting materials. It was prepared by pre-mixed granules of bentonites and sands. As we studied two properties of the grouting materials, the results were obtained as follows:

1. The fluid loss decreases to mixed ratio of 1:4(bentonite/sand), and then increases with the amount of sands increases from ratio of 1:5. In comparison with use of only water, the rate of increase in using slurry of bentonites and water is much lower. However, the fluid loss of existing powder type grouting materials increases steeply.

2. By investigating microstructures of granule-typed grouting materials, the gradual decrease of fluid loss might be due to the decrease of water channels presented in interfaces between granules. Additionally, it is gradually stabilized according to aging time and consequently a uniform grouting layer forms.

3. The thermal conductivity increases with sand content increases. This result is similar to that obtained from existing powder-typed grouting materials.

As a result, the mixed granule grouting materials show superior fluid loss properties to the existing powder type ones. This indicates that the former can be utilized with much more amount of sands than latter. Also, the fluid loss in specimen of ratio of 1:5 is the lowest and the thermal conductivity in specimen of 1:8 ratio increases up to 1.8 W/mK. Consequently, we can select the high content of sand granules, which means high thermal conductivity, on basis of the limited value of fluid loss for use in actual environment. Future work to measure the permeability of mixed granule grouting materials is still needed.
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A Fractal Approach to Calculate the Thermal Conductivity of Moist Soil

Shanshan Cai Tengfei Cui Boren Zheng Pingfang Hu

ABSTRACT

The ground heat exchanger (GHE) is a key component in the design of a GSHP system and the effective thermal conductivity is one of the most important parameters that determine the heat transfer underground. In this paper, the effect of particle sizes and distributions on the sand thermal conductivity were studied both experimentally and analytically. Fractal method was considered for simulating the thermal conductivity of both dry and moist, unsaturated sand. Seven types of dry sand samples and six types of moist, unsaturated sand were selected in the experiments and results showed that both porosity, fractal dimension and particle size ratio affect the sand thermal conductivity. Based on the fractal theory, the fractal models were applied to predict the sand thermal conductivity under both dry and wet conditions. By comparing to the experimental findings, the proposed model was able to predict the variation on the sand thermal conductivity. However, the contact thermal resistance and water distribution pattern are two key impacts on the soil behaviors and need to be further studied.

INTRODUCTION

The ground heat exchanger (GHE) is a key component in the design of a GSHP system and the soil thermal conductivity is one of the most important parameters that correlates to the amount of heat exchanged between the GHE and soil (Nam, et al. 2008; Schibuola and Tambani 2013). Thermal response test is normally used to determine the effective thermal conductivity of soil on site (Sanner, et al. 2005; Signorelli, et al. 2007). It is usually found that the measured result from one site of the borehole deviates from the test result derived from the borehole located next to it. This may lead to large differences during real operation. Although there are several empirical values and correlations published on soil according to the experimental measurements, the values are treated as constant for different types of soil and the correlations are normally correlated with macroscopic parameters, such as porosity and moisture content (Abu-Hamdeh, et al. 2001; Hwang, et al. 2010; Vijdea, et al. 2014). This rough estimation may decrease the accuracy of model prediction. Therefore, it is necessary to correlate the thermal property of soil with the specific mesoscopic structures for better prediction of the heat transfer procedure in soil.

Soil is acknowledged as natural substance with fractal geometry (Katz and Thompson 1985; Thompson and Krohn 1987; Lehmann, et al. 2003). Fractal analysis is a useful tool to describe the natural structures with irregular component sizes and phase arrangements (Bartoli, et al. 1991; Adler and Thovert 1993; Perfect and Kay 1995). Developments in fractal geometry help lead to better understanding of material properties and apparently chaotic processes in nature (Perrier, et al. 1999; Perrier and Bird 2002; Lehmann, et al. 2003; Dathe and Thullner 2005). In this study, the effect of soil particle sizes, distributions and solid thermal conductivity on the soil thermal conductivity were studied both experimentally and theoretically. The simulation results were compared to the experimental findings from laboratory tests.

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EXPERIMENTAL METHOD

Preparation and characterization of sand samples

In order to check the impacts of particle sizes and distribution on the sand thermal conductivity, seven groups of sand samples with fractal characters were prepared according to Table 1 and the main parameters were provided in Table 2. All of them satisfied the requirement for fractal geometry with \( M(Z_p) = M Z_p^D \) (Mandelbrot 1983). \( M(Z_p) \) is the cumulative weight of the particles with the sizes no larger than \( Z_p \), \( Z_p \) represents the diameter of the particles, \( M \) is the total weight of the test sample and \( D \) is the fractal dimension. It should be noted that according to ASTM D653-14, sand is defined as particles of rock that will pass the No. 4 [4.75 mm/0.19 in.] U.S. standard sieve and be retained on the No. 200 [75 µm/0.003 in.] sieve. The particle size of our sample ranged from 0.045 to 5mm (0.002 to 0.197 in.), and the dominant components of our samples are quartz and feldspar. Therefore, our samples belong to “sand” samples.

<table>
<thead>
<tr>
<th>Particle Size mm (in.)</th>
<th>Weight (%)</th>
<th>1</th>
<th>2</th>
<th>3</th>
<th>4</th>
<th>5</th>
<th>6</th>
<th>7</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.045-0.1 (0.002-0.004)</td>
<td>5.94</td>
<td>53.41</td>
<td>20.85</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>0.1-0.125 (0.004-0.005)</td>
<td>1.33</td>
<td>2.44</td>
<td>2.46</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>0.125-0.15 (0.005-0.006)</td>
<td>1.30</td>
<td>2.07</td>
<td>2.23</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>0.15-0.2 (0.006-0.008)</td>
<td>2.53</td>
<td>3.43</td>
<td>3.95</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>0.2-0.7 (0.008-0.028)</td>
<td>17.04</td>
<td>67.49</td>
<td>67.41</td>
<td>37.42</td>
<td>25.68</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>0.7-1.3 (0.028-0.051)</td>
<td>12.71</td>
<td>8.90</td>
<td>10.39</td>
<td>8.88</td>
<td>13.57</td>
<td>20.01</td>
<td></td>
<td></td>
</tr>
<tr>
<td>1.3-1.6 (0.051-0.063)</td>
<td>6.11</td>
<td>3.24</td>
<td>3.79</td>
<td>3.24</td>
<td>5.58</td>
<td>8.22</td>
<td></td>
<td></td>
</tr>
<tr>
<td>1.6-2.3 (0.063-0.091)</td>
<td>13.86</td>
<td>6.00</td>
<td>7.00</td>
<td>5.99</td>
<td>11.25</td>
<td>16.59</td>
<td></td>
<td></td>
</tr>
<tr>
<td>2.3-2.6 (0.091-0.102)</td>
<td>5.80</td>
<td>2.13</td>
<td>2.12</td>
<td>4.29</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>2.6-3 (0.102-0.118)</td>
<td>7.63</td>
<td>2.54</td>
<td>2.54</td>
<td>5.35</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>3-4 (0.118-0.157)</td>
<td>18.66</td>
<td>5.34</td>
<td>5.34</td>
<td>11.98</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>4-5 (0.157-0.197)</td>
<td>18.20</td>
<td>4.37</td>
<td>4.35</td>
<td>10.56</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

Table 2. Summary of the Main Parameters in Test Samples

<table>
<thead>
<tr>
<th>Group No.</th>
<th>Total Weight g(oz.)</th>
<th>Density kg/m³(lb/ft³)</th>
<th>Porosity (-)</th>
<th>Fractal Dimension (-)</th>
<th>Zp(max)/Zp(min) (-)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>70 (2.47)</td>
<td>1707 (106.55)</td>
<td>0.34</td>
<td>2.1</td>
<td>25</td>
</tr>
<tr>
<td>2</td>
<td>70 (2.47)</td>
<td>1707 (106.55)</td>
<td>0.34</td>
<td>2.1</td>
<td>51</td>
</tr>
<tr>
<td>3</td>
<td>70 (2.47)</td>
<td>1707 (106.55)</td>
<td>0.34</td>
<td>2.8</td>
<td>25</td>
</tr>
<tr>
<td>4</td>
<td>70 (2.47)</td>
<td>1707 (106.55)</td>
<td>0.36</td>
<td>2.8</td>
<td>51</td>
</tr>
<tr>
<td>5</td>
<td>68 (2.40)</td>
<td>1659 (103.50)</td>
<td>0.36</td>
<td>2.8</td>
<td>25</td>
</tr>
<tr>
<td>6</td>
<td>70 (2.47)</td>
<td>1707 (106.55)</td>
<td>0.33</td>
<td>2.5</td>
<td>25</td>
</tr>
<tr>
<td>7</td>
<td>75 (2.65)</td>
<td>1829 (114.17)</td>
<td>0.29</td>
<td>2.5</td>
<td>51</td>
</tr>
</tbody>
</table>

Thermal conductivity measurement of dry and moist sand samples

Considering that there would be moisture in the test samples, a hot wire transient method (ASTM C1113) is used for the thermal conductivity measurement (Assael, et al. 2002). Steady state methods usually require long time during testing and the results would be highly affected by moisture redistribution in the test samples. TC3000 from Xiatech Instrument were selected for thermal conductivity measurement. The range of the test instrument was from 0.001 to 10 W/m·K (0.007 to 69.3 Btu·in/(hr·ft²·°F)) with the accuracy of ±3%. The test samples were placed in two test frames with dimensions of 50.5mm ×40.5mm ×20mm (1.99in.×1.59in.×0.79in.). The top and bottom surfaces of each frame were wrapped with plastic films to hold sand in position and prevent moisture evaporation happened during the measurement of moist samples. The sensor was placed in the middle between two frames during measurement. In order to evaluate the effect of two additional layers of plastic film on both sides of the sensor, calibration tests were applied by measuring two standard materials with films between the top and bottoms samples.
The dry sand samples were provided by pre-conditioning in the oven at 105°C (221°F) for 24 hours, or until the two excessive measurement on the weight were within 0.1%. The samples were cooled to room temperature before the measurement. The wet samples were prepared by adding water to sand in a beaker and mixed uniformly. During thermal conductivity measurement, each sample was tested at least three times with three groups of data (3 minutes apart from the last measurement) derived each time (at least nine data points in total). The test conditions were maintained at 25°C (77°F), 1atm (14.7psi).

MODELLING METHOD

Approximation of the structure of sand by Sierpinski carpet

Fractal method is applied in this study to simulate the variation on the sand thermal conductivity. The internal structure of sand is described by Sierpinski carpet model and the thermal resistance network is built according to the Sierpinski geometry based on the 1-D steady state heat transfer analysis (Ma, et al. 2003; Feng, et al. 2004; Feng, et al. 2007; Li, et al. 2012; Jin, et al. 2016). The basic Sierpinski geometry (Mandelbrot 1983) is shown in Figure 1. If the black square represents the solid particle, the model is considered as pore-mass fractal model, and if the black square represents the pore, the model should be treated as solid-mass fractal model. Sand can either be considered as solid- or pore-mass fractal, however, the dimensions of the pores can hardly be determined and sand is treated as pore-mass fractal in the following model. The specifications of the Sierpinski geometry can be computed from Equations (1) to (2). Equation 1 (Mandelbrot 1983) shows the expression for fractal dimension $D$, which is determined by the side length of the Sierpinski carpet and the side length of the center matrix. It is different from dimensions of integer in Euclidean geometry and fractal dimension is often used to describe objects found in nature, such as rough surfaces, coastlines, soil, which are highly disordered and irregular. The porosity $\phi$ of Sierpinski carpet can be determined from Equation 2. Equation 3 represents the particle size ratio between the maximum and minimum particle sizes and the sizes are determined from Equation 4. The basic inputs, such as fractal dimension ($D$), porosity ($\phi$) and diameter of the particle ($ZP$) are determined from preliminary experiments. Different from most empirical correlations, porosity is no longer the only parameter that being considered on thermal conductivity. The particles sizes and distributions also play important roles in the thermal conductivity of porous sand.

\[
D = \frac{\ln(L^2 - C^2)}{\ln L} \tag{1}
\]

\[
\phi = [1 - (C/L)^2]^{n+1} \tag{2}
\]

\[
Z_{p,max}/Z_{p,min} = 1/((L - C)/2L)^n \tag{3}
\]

\[
Z_{p,min} = C((L - C)/2L)^n, \ Z_{p,max} = C \tag{4}
\]

Where $L (\cdot)$ is the side length of the Sierpinski carpet, $C (\cdot)$ is the side length of the center matrix, $\phi (\cdot)$ is the porosity, $Zp (\cdot)$ represents the particle diameter, and $n$ is the number of step.

Figure 2 shows the correlations between porosity ($\phi$) and different parameters, side length of the center matrix ($C$), fractal dimension ($D$) and number of step ($n$). Results showed that for the same porosity, there are different combinations of parameters $C$, $D$ and $n$. Therefore, the model for predicting sand thermal conductivity should not only depend on the porosity, but also include other parameters that would better describe the structure of sand. According to the input values provided in Table 2, the other parameters $C$, $L$, and $n$ of each sample are determined from Equations (1) to (4) and the results are tabulated in Table 3.

![Figure 1 Sierpinski carpet geometry for modeling sand (Jin, et al. 2016)](image)

![Figure 2 Correlations among different parameters](image)
**Two-phase fractal model for dry sand**

Under dry state, only solid and gas phase are considered in the model. According to the specific Sierpinski carpet geometry with appropriate values on C and L, there is a corresponding thermal resistance network existed as shown in Figure 3 (iteration step = 0). The thin, solid bar represents the contact thermal resistance between two particles. By assuming 1-D heat conduction, the thermal resistance network is composed of thermal resistances of solid and gas, plus the contact thermal resistance in a combined arrangement. The effective thermal resistance of each layer can be expressed as Equations (5) and (6). The contact thermal resistance, which represents by the value of \( \tau \) can be neglected when \( \tau < 0.013C/L \) (Ma, et al. 2003). Under this configuration, the difference caused by the contact thermal resistance is less than 1% and the thermal resistance \( R_2 \) can be rewritten in Equation (6).

\[
R_1 = R_3 = \frac{R_{11}R_{12}}{R_{11}+2R_{12}} = \frac{L-C}{2k_aL[(1-\tau)+C]} \tag{5}
\]

\[
R_2 = \frac{k_m(C-\tau)+k_1}{k_a(k_m(L-C)+k_m(C-\tau))} \approx \frac{C}{k_a(L-C)+k_m} \tag{6}
\]

Where \( \tau = t/L \) and it represents the dimensionless contact thermal resistance, \( \kappa = k_m/k_a \) which is the thermal conductivity ratio between solid and gas phase. By building the thermal resistance network, the dimensionless thermal conductivity can be expressed in different steps, as shown in Equations (7) to (8).

\[
K^{(0)} = \frac{k^{(0)}}{k_a} = \left[ \frac{1-\alpha}{\tau(k^{(0)}-1)+1} + \frac{\alpha}{a(k^{(0)}-1)+1} \right]^{-1} \tag{7}
\]

\[
K^{(n)} = \frac{k^{(n)}}{k^{(n-1)}} = K^{(n-1)} \left[ \frac{1-\alpha}{\tau(k^{(n)}-1)+1} + \frac{\alpha}{a(k^{(n)}-1)+1} \right]^{-1} \tag{8}
\]

**Three-phase fractal model for unsaturated, moist sand**

For moist sand, liquid phase is added in the two-phase model and fills in the cavities occupied by the gas phase. The liquid water cannot exist independently in the cavities under unsaturated condition, but covers the exterior surface of solid particles (water films) and accumulate around the intersection points (water bridges) (Ma, et al. 2003; Feng, et al. 2007; Jin, et al. 2016), as shown by the blue regions in Figure 4. The degree of moisture level can be expressed by the degree of saturation \( S_h \) and \( S_v = V_v/V \), where \( V_v (\text{m}^3) \) and \( V (\text{m}^3) \) represent the total volume of the liquid phase and total volume of the sample respectively. If assume the amount of liquid that covers the solid particles is \( S_h \) and the amount of liquid that forms water bridges is \( S_v \), then \( S_v = S_f + S_h \). According to the Sierpinski model, the amount of liquid can be expressed as Equations (9) to (12).

---

**Table 3. Summary of the Main Parameters in Test Samples**

<table>
<thead>
<tr>
<th>Group No.</th>
<th>( \emptyset )</th>
<th>D</th>
<th>( Zp(max)/Zp(min) )</th>
<th>C</th>
<th>L</th>
<th>n</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>0.34</td>
<td>2.1</td>
<td>25</td>
<td>1.45</td>
<td>0.77</td>
<td>2.21</td>
</tr>
<tr>
<td>2</td>
<td>0.34</td>
<td>2.1</td>
<td>51</td>
<td>1.36</td>
<td>0.67</td>
<td>2.87</td>
</tr>
<tr>
<td>4</td>
<td>0.36</td>
<td>2.8</td>
<td>51</td>
<td>3.66</td>
<td>1.75</td>
<td>2.93</td>
</tr>
<tr>
<td>5</td>
<td>0.36</td>
<td>2.8</td>
<td>25</td>
<td>4.77</td>
<td>2.47</td>
<td>2.26</td>
</tr>
<tr>
<td>6</td>
<td>0.33</td>
<td>2.5</td>
<td>51</td>
<td>2.01</td>
<td>1.09</td>
<td>2.18</td>
</tr>
<tr>
<td>7</td>
<td>0.29</td>
<td>2.5</td>
<td>25</td>
<td>1.93</td>
<td>1.02</td>
<td>2.72</td>
</tr>
</tbody>
</table>

---

*Image 4* Thermal resistance network for Sierpinski carpet when step = 0 (three phases) (Jin, et al. 2016)
Certain amount of water was added uniformly in the sample to control the degree of saturation gradually when compared to the dry values and saturation degree is the portion of water volume over the total vacancy in the different moisture levels. From Figure 5, it was observed that samples 4 and 7 showed faster increasing rate when proportional to the particle size ratio, and this explains the phenomenon that observed on samples 4 and 5. Sample 4 findings on the variation of thermal conductivity ratio with moisture content (expressed as the degree of saturation) were similar to the two-phase model, the dimensionless thermal conductivity model developed for unsaturated, moist phase can be simulated according to Equations (13) and (14).

Where $\beta$ and $\omega$ are the dimensionless width of water films and water bridges, respectively. $\beta^{(n)}$ and $\omega^{(n)}$ represent the corresponding dimensionless forms at the $n$th stage. The test results of seven dry sand samples were listed in Table 4. Among test samples 1, 2 and 3, results showed that sample 5 with higher porosity performed more conductive than the behavior of sample 3 with lower porosity. Higher porosity provides more vacancies for trapping air and increases the overall thermal resistance. Similar phenomenon was also observed between samples 6 and 7, however, it should be noted that the 22.2% increase on the thermal conductivity with regarding to the sample 6 was not only caused by the porosity, but also affected by the sizes of the particles in the samples. The thermal conductivity behaves proportional to the values of porosity, but inverse proportional to the less uniform sizes in the sample 4.

<table>
<thead>
<tr>
<th>Table 4. Summary of the Main Parameters in Test Samples</th>
</tr>
</thead>
<tbody>
<tr>
<td>Group No.</td>
</tr>
<tr>
<td>Thermal Conductivity</td>
</tr>
</tbody>
</table>

Six test samples, 1, 2, 4, 5, 6 and 7 were selected for the measurement of thermal conductivity with various amount of water. The sand thermal conductivity was observed fluctuating during wet test and the experimental findings on the variation of thermal conductivity ratio with moisture content (expressed as the degree of saturation) were provided in Figure 5. The thermal conductivity ratio represents the increasing on the sand thermal conductivity when compared to the dry values and saturation degree is the portion of water volume over the total vacancy in the sand. Certain amount of water was added uniformly in the sample to control the degree of saturation gradually increased up to 70%. Six trend lines were applied to estimate the variation of thermal conductivity ratio of sand under different moisture levels. From Figure 5, it was observed that samples 4 and 7 showed faster increasing rate when compared to the other four samples. This phenomenon can be explained with the following three reasons. First, these
two test samples were composed of particles at small sizes. During the experiment, it was found that liquid water existed in the cavities in two forms – the water bridge among different particles and the water film around the particles as shown in the picture taken by the optical microscope (Figure 6). Compared to large particles, the small ones have larger surface-to-volume ratio and have higher possibilities to form water bridges with the other particles located nearby. For the same thickness of water film and water bridge, water bridge leads to stronger effect on the overall thermal resistance than water film. Second, these two samples had higher particle size ratio when compared to most of the others. Similar to the explanation under dry conditions, higher particle size ratio would lead to higher possibility on the formation of thermal paths and increase the total thermal conductivity of sand. The effect of particle size ratio seems to vary with different fractal dimensions. For example, both samples 1 and 2 were prepared with fractal dimensions around 2.1 and it was observed that although the dry values of these two samples were different, the thermal conductivity ratio measured during wet tests were very similar. However, for test samples 3 and 4 with fractal dimensions around 2.8, the differences on the thermal conductivity ratio were higher than the samples 1 and 2 which were prepared with more uniform mixture of the particles at different sizes. Test samples 6 and 7 with fractal dimensions around 2.5 also showed large differences on the thermal conductivity ratio than the differences between samples 1 and 2. Therefore, the particle size ratio would affect the thermal conductivity of the sand and the effect behaves more significant in the less uniform samples. It should be noted besides particle size ratio, the porosity were different in samples 6 and 7 and this parameter should also be considered as one of the impact factors to explain the difference of the thermal conductivity ratio between these two samples. If the porosity of samples 6 and 7 are the same, the difference on the variation of thermal conductivity would between sand samples with higher fractal dimension (samples 3 and 4) and those with lower fractal dimension (samples 1 and 2). Therefore, porosity is the third reason that would explain the higher thermal conductivity ratio derived in some sand samples, especially for sample 4. The porosity of test sample 4 is 0.29, which is the lowest among all the six test samples. For the same degree of saturation, the amount of water in sample 4 is more than the other samples with thicker water films and water bridges, which result in higher values of the thermal conductivity ratio.

![Figure 5](image)

**Figure 5** Thermal conductivity ratio of moist sand

![Figure 6](image)

**Figure 6** Optic photo of moist sand by optical microscope

**Modelling results**

The fractal model described in the previous section was used to simulate thermal conductivity of sand in both dry and wet conditions. According to the fractal dimensions, porosity and particle size ratio, parameters $C$, $L$ and $n$ can be determined from Equations (1) and (2) and tabulated in Table 3.

The simulated thermal conductivity of six test samples are provided in Table 5. The thermal conductivity of air was selected at 0.026 W/m-K (0.2 Btu·in/(hr·ft²·°F)). Actually, our sandstone samples were taken directly from the building yard, the dominant component is quartz and also contain some of feldspar. It’s difficult to give the exact thermal conductivity values of the solid components, because quartz sandstones of our samples may have thermal conductivity quite different from that of the quartz crystals. Furthermore, quartz is highly anisotropic, with a thermal conductivity in the direction of the crystal axis twice higher than that in the direction perpendicular to the crystal axis (Woodside and Messmer 1961). Therefore, the value of $k_s$ was determined by using a geometric mean equation (Lu,
et al. 2007), shown in Equation (15):

\[ k_s = k_{qs}^\theta k_o^{1-\theta} \]  

(15)

Where \( \theta \) is the content of quartz sandstones (\( \theta = 0.8 \)) with thermal conductivity \( k_{qs} = 2.5 \) W/m·K (17.3 Btu·in/(hr·ft²·°F)), and other minerals, such as feldspar, thermal conductivity \( k_0 \) was taken as 1 W/m·K (6.9 Btu·in/(hr·ft²·°F)). In this work, a value for \( k_s \) of 2 W/m·K (13.9 Btu·in/(hr·ft²·°F)) at room temperature has been selected.

The thermal conductivity of dry sand could be determined according to Equation (8) and results showed that the simulated values matched experimental data within 1.7%. It should be noted that the contact thermal resistance is a factor that correlates with the shape of particles, ways of contact and degree of compactness, all of which are difficult to be accurately measured. Rough estimation was given in Table 5 on the contact thermal resistance during the model calculation. A sensitivity analysis was applied on the contact thermal resistance to investigate its impact on the sand thermal conductivity and the findings were plotted in Figure 7. Results showed that if the contact thermal resistance varies by ±20%, the maximum difference on the sand thermal conductivity was within ±5%. The impact of the contact thermal resistance was more significant in the more uniform sand samples with lower fractal dimension and particle size ratio. The impacts of the other two parameters, porosity and particle size ratio, were also included in the sensitivity analysis as shown in Figure 8. The parameters considered in the baseline case was provided in the caption of the figure. It was observed that porosity and particle size ratio led to inverse impacts on the sand thermal conductivity and such phenomenon matched with the previous experimental findings. Compared to the particle size ratio, porosity played more significant role in the total value of sand thermal conductivity. A ±20% variation on the sand porosity led to ±20% difference on the sand thermal conductivity, but the same variation on the particle size ratio of the sand would only lead to differences around ±5%. The effect of solid thermal conductivity behaved similar to that of particle size and ±20% variation would cause the effective thermal conductivity vary by ±5%.

**Table 5. Experimental and simulated thermal conductivity of dry sand**

<table>
<thead>
<tr>
<th>Samples</th>
<th>1</th>
<th>2</th>
<th>4</th>
<th>5</th>
<th>6</th>
<th>7</th>
</tr>
</thead>
<tbody>
<tr>
<td>Measured thermal conductivity (W/m·K)</td>
<td>0.279</td>
<td>0.316</td>
<td>0.256</td>
<td>0.24</td>
<td>0.271</td>
<td>0.333</td>
</tr>
<tr>
<td>Measured thermal conductivity (Btu·in/(hr·ft²·°F))</td>
<td>(1.93)</td>
<td>(2.19)</td>
<td>(1.77)</td>
<td>(1.66)</td>
<td>(1.88)</td>
<td>(2.34)</td>
</tr>
<tr>
<td>Simulated thermal conductivity (W/m·K)</td>
<td>0.274</td>
<td>0.315</td>
<td>0.258</td>
<td>0.24</td>
<td>0.271</td>
<td>0.332</td>
</tr>
<tr>
<td>Simulated thermal conductivity (Btu·in/(hr·ft²·°F))</td>
<td>(1.90)</td>
<td>(2.18)</td>
<td>(1.79)</td>
<td>(1.66)</td>
<td>(1.88)</td>
<td>(2.30)</td>
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<tr>
<td>Dimensionless contact thermal resistance</td>
<td>-0.005</td>
<td>0.006</td>
<td>0</td>
<td>0.003</td>
<td>0.004</td>
<td>0.0003</td>
</tr>
<tr>
<td>Difference</td>
<td>%</td>
<td>1.7</td>
<td>0.3</td>
<td>0.7</td>
<td>0.08</td>
<td>0.08</td>
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**Figure 7** Sensitivity study of contact thermal resistance

**Figure 8** Sensitivity study of three main parameters

(Baseline: \( \Phi = 0.35 \), \( D = 2.8 \), \( Zp(max)/Zp(min) = 25 \), \( k_s = 0.026 \) W/m·K (0.2 Btu·in/(hr·ft²·°F)), \( k_m = 2 \) W/m·K (13.9 Btu·in/(hr·ft²·°F)))
The thermal conductivity for moist, unsaturated sand was computed according to Equations (13) and (14). Figure 9 indicates the comparison between simulation results with the experimental findings on six types of sand samples. The simulation curves almost matched the experimental trends with certain water distribution pattern by adjusting the thickness of water film around the solid particles and water bridges among them. The parameters for the amount of water films ($S_f$) formed around particles were provided in the caption of Figure 9, and the amount of water formed as water bridges among particles were present as $(1-S_f) \cdot S_w$. This amount of water correlated with the dimensionless thickness of water films, as well as water bridges, and higher thickness led to fast deterioration on the sand thermal conductivity. The amount of water film was in the range of 60% to 99% of the maximum saturated amount ($S_w$). According to the previous findings, more water was covered around contact area and decreased the contact thermal resistance for the less uniform samples with large number of small particles. However, the water distribution pattern closely correlates with the particle shapes and size ratio and it would further affect the values on the sand thermal conductivity. It should be noted that soil-water retention curve would be very helpful to compare the amount of water retained in different types of samples with various porosity and particle size, and it will be considered in the further study on the modelling of moisture accumulation and distribution in the sand. Figure 10 provided the variation on the thermal conductivity with different water distribution structures. The moisture accumulation in the first pattern was assumed to form water bridge only when the moisture content below 30%, and when the moisture content reached above 30%, the moisture accumulate as water film around the sand particles. The second pattern was considered with constant number of water bridge and the thickness of water film was gradually increased. Results showed that the variations caused by these two different patterns were ranged from -2.5% to 7% when compared to the baseline data derived on test sample 4. Therefore, it is important to determine the rules for moisture accumulation and distribution among particles in sand or soil and this work need to be further studied.

**CONCLUSION**

This paper aims to study the impacts of particle sizes and distribution on the values of thermal conductivity of both dry and moist, unsaturated sand. The thermal conductivity of seven dry sand samples and six wet sand samples were tested according to the hot wire test device. Fractal method was proposed in this study to include the mesoscopic effect in the simulations of sand thermal conductivity. The main findings from the experimental and simulation results are summarized as follows. First, porosity, particle sizes and distributions affect the sand thermal conductivity.
conductivity. Between fractal dimension and the particle size ratio, it seems that the particle size ratio plays a more significant role. Larger differences on the sizes of the particles in the sand lead to higher values of thermal conductivity. Second, the thermal behavior of moist, unsaturated sand also correlates with porosity, particle sizes and uniformity. More uniform sand with smaller surface-to-volume ratio lead to much flatter variation on the thermal conductivity ratio. Third, fractal method is a promising technique to correlate sand thermal conductivity with mesoscopic geometry of sand and improve the accuracy of predicted values on the sand thermal conductivity. It seems that the simulation results from current fractal model generally matched with the experimental findings. However, the contact thermal resistance and water distribution patterns are still need to be further studied in the future work.

ACKNOWLEDGMENTS

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NOMENCLATURE

\( C \) = side length of the central matrix (\( \cdot \))
\( D \) = fractal dimension (\( \cdot \))
\( K \) = dimensionless thermal conductivity (\( \cdot \))
\( k \) = thermal conductivity (\( \text{W/m-K or Btu·in/(hr·ft}^2·°F} \))
\( L \) = side length of the Sierpinski carpet (\( \cdot \))
\( M \) = mass (kg or lbm)
\( n \) = number of iteration
\( q \) = specific heat load (\( \text{W/m or Btu/hr ·ft}^2 \))
\( R \) = thermal resistance (K/W)
\( r \) = thickness of water layer (\( \cdot \))
\( S \) = saturation degree (\( \cdot \))
\( V \) = volume (m\(^3\))
\( w \) = width of water bridges (\( \cdot \))
\( Z \) = diameter of particle (\( \cdot \))
\( \alpha \) = dimensionless side length of the central matrix, defined by \( C/L \)
\( \beta \) = dimensionless thickness of surrounding water layer, defined by \( r/L \)
\( \Phi \) = porosity (\( \cdot \))
\( \kappa \) = ratio of matrix to dry air thermal conductivity (\( \cdot \))
\( \theta \) = the content of quartzitic sandstones
\( \tau \) = dimensionless width of virtual thermal resistance, defined by \( t/L \)
\( \omega \) = dimensionless width of connected water bridges, defined by \( w/L \)

GREEK SYMBOLS

Subscripts

\( a \) = air
\( q_s \) = quartzitic sandstones
\( o \) = other solid particles
\( b \) = connected water bridges
\( f \) = surrounding water layer
\( m \) = matrix
\( p \) = particle
\( s \) = surface
\( w \) = water

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