

PERFORMANCE CHARACTERIZATION OF SHALLOW HELICAL
GROUND HEAT EXCHANGERS FOR GROUND SOURCE
HEAT PUMP APPLICATIONS

by

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of the requirements for the degree

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DEDICATION

To my family

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NOMENCLATURE AND ACRONYMS

Δ	delta (difference)
$^{\circ}\text{C}$	degree Celsius
$^{\circ}\text{F}$	degree Fahrenheit
A	area
ANSI	American National Standards Institute
ASTM	American Society for Testing and Materials
A_T	annual semi-amplitude of the monthly air temperature
Btu	British thermal unit
C	volumetric heat capacity; thermal capacitance
CO_2	carbon dioxide
c_p	specific heat
d	total derivative
DAQ	data acquisition
DHW	domestic hot water
DX	direct expansion
EWT	entering water (fluid) temperature
ft	foot
g	internal heat generation
gal	gallon
GCHP	ground coupled heat pump
GHE	ground heat exchanger
GHP	geothermal heat pump
gpm	gallons per minute
GSHP	ground source heat pump
h	convective heat transfer coefficient
hr	hour
HRV	heat recovery ventilation
i, j	discretization indices
ICF	insulating concrete form
in	inch
K	Kelvin (absolute temperature)
k	thermal conductivity
kg	kilogram
L	length
l	liter
lb_m	pound-mass
LWT	leaving water (fluid) temperature
$\dot{m}$	mass flow rate
m	meter
MBE	mean bias error
min	minute

MSU	Montana State University
NCDC	National Climatic Data Center
NCEI	National Centers for Environmental Information
NOAA	National Oceanic and Atmospheric Administration
NO _x	nitrogen oxides
Nu	Nusselt number
PEX	cross-linked polyethylene
PEXa	cross-linked polyethylene (peroxide method)
Pr	Prandtl number
psia	pounds per inch absolute
q	rate of heat transfer
$\dot{Q}$	volumetric flow rate
r	radial coordinate
R	thermal resistance
Re	Reynolds number
RMEH	REHAU [®] Montana Ecosmart House
RMSE	root mean squared error
RSC	REHAU [®] Smart Controls
RSS	root of the sum of the squares
s	second
SIP	structural insulated panel
T	temperature
USCS	unified soil classification system
UTES	underground thermal energy store
V	volume
W	watt
x	horizontal coordinate
z	depth coordinate
α	thermal diffusivity
∂	partial derivative
ε	emittance
ρ	density
σ	Stefan-Boltzmann constant
τ	time

ABSTRACT

The use of the ground as heat source or sink medium for heating and cooling the human built environment has the potential of saving energy and reducing greenhouse gas emissions significantly. One of the main components of this system is the heat exchanger that is buried in the ground. This thesis explores the performance of a specific type of ground heat exchanger (GHE): the shallow vertical helical GHE. This type of GHE occupies considerably less land when compared to horizontal configurations and are less influenced by outdoor temperature and weather conditions. When compared to traditional deep vertical U-tube probes, savings on drilling costs can be significant. However, performance data and design information is limited for these types of heat exchangers, which has limited their adoption amongst ground source heat pump (GSHP) system designers and installers. Various in-situ heating and cooling tests were performed at a residence in Bozeman, Montana, with a system containing three helical GHEs. The heat exchangers are coupled with a GSHP with variable capacity compressors. Moreover, a recently developed numerical model (named CaRM-He) was used to compare the experimental results with the simulated performance. The model is based on the analogy between thermal and electrical phenomena, where the domain (comprised of GHE, surrounding ground, grouting material, and heat carrier fluid) is discretized as a linked network of thermal nodes with thermal capacitances and resistances. Heat exchanger outlet temperature as predicted by CaRM-He model was compared with experimental data, resulting in different degrees of accuracy. This research presents a method to characterize the performance of these types of GHEs, and a comprehensive analysis of uncertainties and sources of error inherent to in-situ testing. Also, it is recommended that the model is improved to include extraneous heat inputs and horizontal runs. Still however, the advantages of this ground coupling method and the possibility to predict its performance make the helical GHE an interesting alternative for the geothermal designer.

INTRODUCTION

Motivation

Geothermal energy is the heat generated and stored in the earth's crust. In the context of building systems, geothermal systems are those that make use of the geothermal energy to heat or cool building spaces. At the core of these systems there is typically a geothermal heat pump (GHP).

According to a study from Liu (2010), retrofitting the traditional systems for space heating and cooling and water heating in single-family U.S. homes to geothermal heat pump (GHP) systems would have significant benefits, especially in:

- Energy savings, both for the utility companies and the consumer
- Lowering the summer electrical peak demand
- Reduction of CO₂ and other greenhouse gas emissions

The secret to all these benefits lies with the fact that under most circumstances, a well-designed and installed geothermal system operates at higher efficiency than its equivalent traditional counterpart.

Geothermal systems use the ground as heat “sink” and “source” rather than the air. Below the surface of the earth, the temperatures are much more constant than in the atmosphere, very much like in a cave. The deeper the distance below the surface, the more constant the temperature is (Figure 1); this constant is approximately equal to the annual average outdoor air temperature (around 50°F [10°C] in many mid-latitude locations). This phenomenon creates an advantage for the “ground source” heat pump, which does not need to work as hard to “pump” heat in or out of the conditioned spaces,

therefore consuming less energy than the traditional “air-source” air conditioners. In addition, a heat pump can provide heating and cooling with the same equipment.

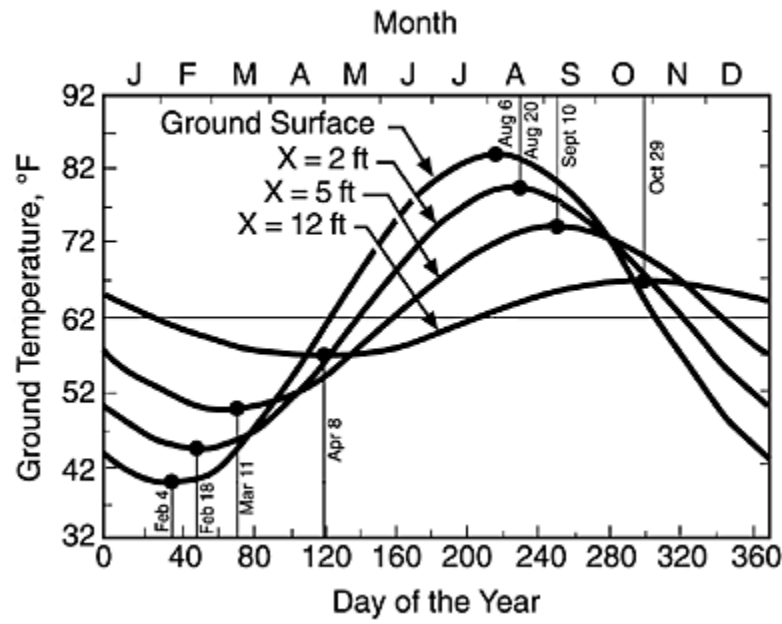


Figure 1. Temperature dampening effect as a function of underground depth (Source: www.builtitsolar.com)

However, the implementation of these systems is not widespread. The major limiting factors for the installation of geothermal systems are:

- High initial cost
- Lack of knowledge or trust in the system benefits
- Limited design/installation infrastructure
- Lack of new technologies and techniques

Therefore, expanding the body of knowledge of any part of these systems would help to overcome the barriers that limit their implementation.

A type of geothermal heat pumps is the ground coupled heat pump (GCHP), also called closed-loop ground source heat pump (GSHP). This type of system exchanges heat with the ground by means of a pressurized loop of thermoplastic pipes buried in the ground. This is called the ground heat exchanger (GHE).

Typically, closed-loop GHEs can be buried horizontally or vertically into the ground. Horizontal installations consist of a loop of pipes buried in shallow trenches excavated in the surrounding of the building. Vertical loops are placed in narrow boreholes, drilled down to a depth of 300 feet (91 m) or more. Vertical U-tubes are the most common configuration amongst vertical GHEs. This thesis investigates the performance of a novel type of GHE: the helical GHE.

The helical GHE is more compact than the straight U-tube; therefore, it can be buried at a much shallower depth, typically less than 20 feet (6.1 m). Although the borehole is wider in diameter, it can be excavated with an earth auger, which is less expensive than the regular deep borehole drilling rigs. In addition, the helical GHE can be advantageous in locations where there is an impediment to drilling deep wells because of geological, hydrological or local regulatory reasons. Vertical GHEs take advantage of the constant temperature of the deep ground; however, at the typical depth of the helical GHE the yearly temperature swing is still noticeable.

Compared with horizontal configurations, the helical GHE occupies considerably less land, providing an advantage in populated areas or where there is simply not enough space for a horizontal geothermal field.

A solid understanding of the behavior of such GHEs is important for designing successful and energy efficient geothermal systems. However, performance data and design guidelines are not readily available for helical GHEs, consequently limiting their adoption amongst GSHP system designers and installers.

Purpose

This thesis explores the performance of the shallow depth vertical helical GHE in different conditions. In order to do that, in-situ tests were carried out in a testing facility in Bozeman, Montana and the experimental results were compared with simulations from a previously reported numerical model.

Objectives

The objectives of this research are to:

- Provide the HVAC community with performance information and insight about the helical shaped vertical ground heat exchanger for ground coupled heat pump applications
- Validate a previously reported model with real data obtained from in-situ testing to prove that it is valid under a variety of operational conditions
- Help the HVAC industry to design quality and energy efficient systems ultimately leading to improve the environmental sustainability

BACKGROUND

Types of Geothermal Energy

Geothermal energy can be split in two main types:

- High grade. This is the type of geothermal energy that comes from the earth's core. Sometimes, this energy rises up near the crust, heating up the water and turning it into steam. This type of energy can be used for bathing, direct space heating and electrical generation.
- Low grade. This is actually solar energy stored in the upper layers of the earth's crust. This is the type of geothermal energy that is used as heat sink/source for geothermal HVAC applications. This thesis delves into this type of geothermal energy.

Ground Source Heat Pumps

Ground source heat pumps (GSHP) are divided into the following types attending to the type of exterior heat exchanger (Kavanaugh, Rafferty, & American Society of Heating Refrigerating and Air-Conditioning Engineers., 2014):

- Ground-coupled heat pumps (GCHP). They have a closed loop ground heat exchanger buried in the ground (Figure 2. Bottom left and bottom right)
- Ground-water heat pumps (GWHP) have an open-loop piping system of wells to exchange heat with aquifers underground (Figure 2. Upper right)

- Surface-water heat pumps (SWHP). These types of heat pumps exchange heat with a mass of water that is close to the surface, like a lake. They can be open or closed loop (Figure 2. Upper left)
- Direct expansion (DX) GCHP. This is a type of heat pump where the exterior side of the refrigerant loop is directly connected to the ground source. This type is not very common.

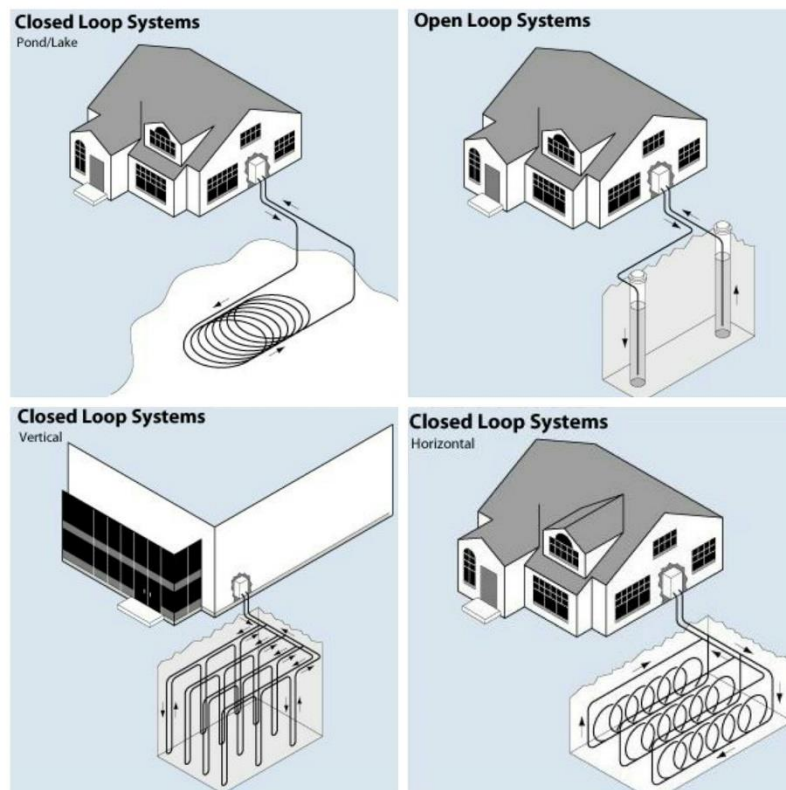


Figure 2. Different types of ground source heat pump (Source: www.facebook.com)

Ground Heat Exchangers

Ground heat exchangers (GHE) can be horizontal or vertical. Examples of horizontal GHEs are: single pipe, multi pipe or Slinky™. They are typically buried at

depths around 5 feet (1.5 m) in trenches. They require the maximum amount of ground area but are also the least expensive.

Vertical configurations include single and double U-tube and coaxial pipes. Deep vertical boreholes are typically 200 to 300 feet (61 to 91 m). More than 400 ft (122 m) is usually not recommended because of possible pressure instabilities and increased drilling and pumping cost. Deep vertical GHEs occupy the least ground area and deliver great efficiencies but are also the most expensive.

A relatively new configuration has been introduced in the market which is a compromise between horizontal and deep vertical GHEs. It can be considered vertical, but it is buried at much shallower depths. It consists of a pipe shaped into a helix. This thesis focuses on the study of this specific type of ground heat exchanger for ground-coupled heat pumps (GCHP).

An example of what a helical GHE application would look like can be seen in Figure 3.

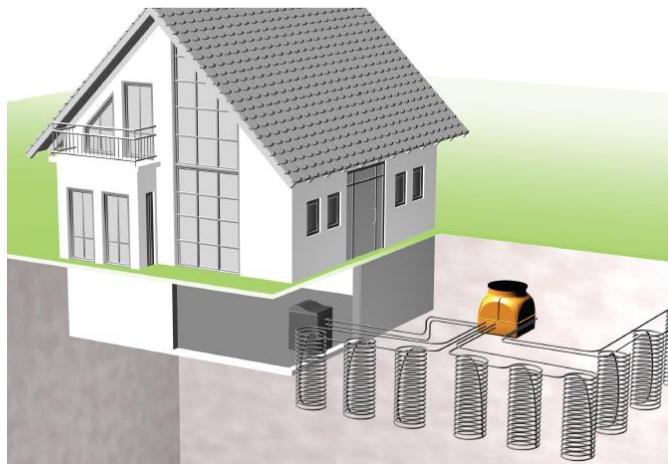


Figure 3. Representation of an example installation of helical GHE (Source: www.rehau.com)

LITERATURE REVIEW

In this section, a comprehensive analysis of studies found in literature regarding helical ground heat exchangers is presented. Starting with a historical approach to the GSHP applications in general, it is organized in chronological order, emphasizing the different modeling approaches used to predict the exchanger's performance. The section ends with a note about a commercial software application for GHE design.

Literature Review

Using the ground as a heat source/sink for space heating and cooling applications is well documented in literature. A comprehensive technology review of ground source heat pumps in general can be found in Rawlings and Sykulski (1999), with an application dating back to Switzerland in 1912. But it was not until 1947 that Kemler described many of the basic types of GHEs used today, including the vertical helical GHE (Kemler, 1947). An excellent historical review of the use of GSHPs was done by Sanner (2008) in his work "Ground Source Heat Pump: A Guide Book".

Development of models regarding conventional deep vertical boreholes are widespread and their implementation is already available in commercial geothermal design software. This is not surprising because those types of GHE – together with the horizontal trench-installed ones – are the most commonly used nowadays. Conversely, simulation tools for helical GHE are not so readily available. This idea is supported by the fact that vertical helical GHE are not yet covered in the ASHRAE's latest edition of geothermal heating and cooling for commercial and institutional applications design

guidelines (Kavanaugh et al., 2014). This is most likely to change in the near future since the vertical helical GHE is an alternative to the conventional deep vertical borehole.

Svec and Palmer (1989) investigated the performance of an experimental vertical spiral GHE and claimed that it produced improved heat exchange rates; thus, the length of the GHE could be reduced, ultimately reducing installation costs. Moreover, they found that low operation temperatures, such that freezing-thawing cycles took place, enhanced the overall performance as well, as long as the structure of the ground was not affected.

Rabin, Korin, and Sher (1991) developed a simplified numerical model to analyze thermal performance of a helical heat exchanger for the purpose of storing energy in the soil. This model used semi-infinite 2-D axisymmetric transient heat conduction and was solved by means of finite difference method over a simplified geometry, with the helix modeled as a series of equally spaced horizontal rings. The results from the numerical model were compared with analytical solutions and field experiments showing a strong correlation. Interestingly, the REHAU® Montana ecosmart house uses the concept of seasonal solar energy storage into the ground, the “heat sink”, to enhance the efficiency of the GSHP; although, it has not been investigated in depth yet. The first two authors of the previous study later reported findings from a parametric study using the same numerical model (Rabin & Korin, 1996). The parameters involved were: thermal properties of the soil, length of charging/discharging period, and total height and pitch distance of the heat exchanger.

Nir, Doughty, and Tsang (1992) presented a comprehensive study on thermal energy storage in unsaturated soils. As in the previous work, the type of GHE chosen was a helical shaped one. Thus, this type of application can be considered the precursor to modeling this type of geometry for ground heat exchangers. The authors developed a 2-D axisymmetric transient mathematical model based on conservation of energy with only heat conduction occurring in the soil and conduction plus convection in the heat exchanger side. In this case, the geometry of the heat exchanger was approximated as an annular cylindrical conduit. The translation of the mathematical model into a computational form was done by means of the integral-finite-difference method. They validated the model with experimental results and performed a sensitivity analysis.

Together with the underground thermal energy storage application, also known as UTES, one of the motivators for modeling vertical helical GHEs was undoubtedly the arising of geothermal foundation piles or “energy piles”. Energy piles are clever devices that take advantage of the need of foundation piles in certain types of buildings to embed a closed-loop pipe system into the piles and then couple it to a GSHP. This dual-purpose building element can help mitigate the high initial cost of GHE installation, well known to be one of the most important portions of the total geothermal installation budget. One way to increase the exchange surface and the overall efficiency of the heat exchanger is to shape it into a spiral or helix that fits near the exterior surface of the structural pile (Figure 4 and Figure 5).

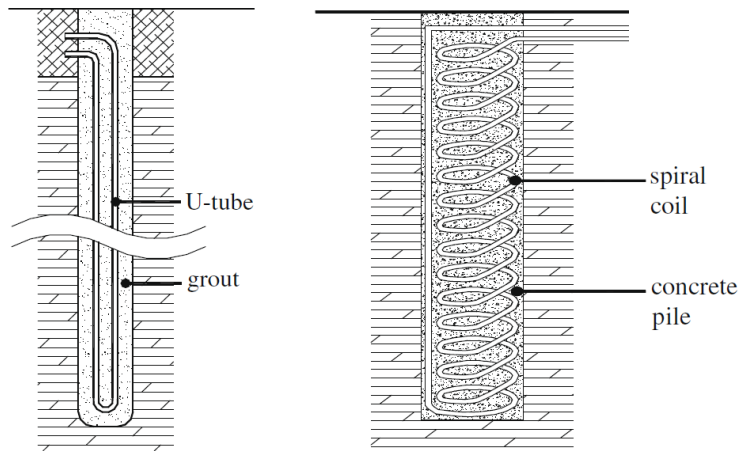


Figure 4. Two possible configurations of an energy pile (Man, Yang, Diao, Liu, & Fang, 2010)



Figure 5. Energy pile foundation structure with thermoplastic pipe in spiral configuration (Source: www.rehau.com)

Effectively, the energy pile and the short depth helical GHE have the same geometry and can be similarly modeled. Energy piles have been covered in literature and several scholarly articles deal with their thermal behavior. For instance, Man et al. (2010) developed a new model and analytical solutions for borehole and pile heat exchangers with helical shape. Evolving from the classical models, i.e. the infinite and finite line source model and the “hollow” cylindrical model, they presented a new one called the “solid” cylindrical source. The line source model has been extensively used for geothermal applications, especially in determining the thermal properties of the ground by means of a thermal response test (TRT). Having an accurate estimation of the thermal properties of the ground is key to a successful engineering design of the geothermal system, especially the GHE sizing. However, the line source method comes from the assumption that the classic deep borehole, typically a U-tube backfilled with grouting material, can be approximated as a 1-D “hot rod” where axial dimension is neglected. This is certainly a good assumption for a typical 300 foot deep vertical borehole of only 6.5 inches in diameter (depth-to-radius ratio = 1108), but not necessarily for a short depth vertical helical GHE that is 12 feet deep and 18 inches in diameter (depth-to-radius ratio = 16). The “hollow” cylindrical model partly overcomes this limitation, but still neglects the heat capacity of the material inside the borehole. The “solid” cylindrical model takes into account the axial dimension of the heat exchanger, improving the accuracy of the results, especially for short term response; but in order to obtain a relatively simple analytical solution, the interior of the borehole is assumed to be made of the same material as the surrounding ground. This is not a good assumption for the energy pile

simulation since they are normally made of reinforced concrete. The analytical solutions for this model were derived by means of Green's functions method. Green's functions have been proven to be very versatile in ground thermal analysis of any type of ground heat exchanger, e.g. earth-air heat exchangers (Denowh, 2015).

A further improvement of the heat transfer analysis of the pile geothermal heat exchanger with helical coils was performed by Cui, Li, Man, and Fang (2011). They presented a transient ring-coil heat source model, where the geometry of the helix was simplified as a series of rings separated by the pitch of the helix. Explicit analytical solutions were derived by means of the Green's functions theory and the image method. The ring-like geometry makes the analysis of the influence of the coil pitch straight forward, unlike its precursor, the "solid" cylindrical source model. One of the most important assumptions of this model is that the ground is regarded as a semi-infinite homogeneous medium with constant temperature as the boundary condition at the surface. This makes sense when modeling a pile heat exchanger, since the pile lies beneath the ground slab of the building it supports, and it can be assumed the interior temperature of a building is fairly constant throughout the year. However, if the heat exchanger is in the open field, as is normally the case for a non-pile geothermal application, this assumption is far from accurate, especially in the shallower layers of the ground. Nevertheless, this model was a step forward compared to the previous models to simulate shallow depth GHEs.

Evolved from the previous model, Man et al. (2011) developed a so-called "spiral heat source model" for pile GHEs. In this case, the geometry of the GHE was considered

fully in 3 dimensions. They presented and compared an infinite spiral source model and a finite spiral source model under steady and variable loads. They continued to use the Green's functions method to derive the analytical solutions under the following general assumptions: the ground is considered a homogeneous infinite medium, with thermo-physical properties independent of temperature and with uniform initial temperature. Again, they assumed that the medium inside and outside the helix was the same, making the model unsuitable for GHEs grouted with a different material than the original native soil. They found that the influence of the axial heat transfer is high, especially for long term simulations. The influence of anisotropy in the medium was investigated by Li and Lai (2012). They claim that sedimentary rocks and stratified aquifers are typical examples of anisotropic media. It is true that anisotropy could have a significant influence in the thermal response of deep energy piles buried in rocky media and very deep straight vertical boreholes that might come in contact with the water table. However, this thesis focuses on short helical boreholes that are typically buried in the shallower layers of the ground, where effects of anisotropy are less important.

A case study comparison between some existing models and numerical and analytical solutions of helical GHEs was performed by H. Park, Lee, Yoon, Shin, and Lee (2012). More specifically, they compared the previously reported solid cylindrical and ring coils models with indoor and field TRTs obtaining relatively good agreement. Only during the first stages of the test, the deviation was apparent. This could have been caused by those models not considering the straight axial section of pipe that typically runs through the center of the helix. When this section was approximated by the finite

line source model and superimposed to the previous models, the level of agreement increased, proving that that section of the pipe has certain influence in the overall thermal response of the GHE especially in early stages of the transient heat transfer process. They also pointed out the necessity of a reliable characterization of the thermal properties of the ground, i.e. thermal conductivity and thermal diffusivity in the engineering process.

In a later paper, S. Park, Lee, Park, Yoon, and Chung (2013) presented a fully 3-D analytical solution for spiral coil type GHEs. The analytical solution is based on Green's function theory and is further simplified for computational reasons via error functions. This work was mainly aimed to the energy pile application, therefore some of the typical assumptions for this type of GHE were in place – e.g. the boundary condition at the ground surface is a constant temperature rather than a harmonic function affected by the weather variations. Moreover, the helix geometry is modeled as a line source without mass, heat capacity or thickness; and there is no vertical internal pipe for supply or return. The ground being a homogeneous semi-infinite medium with temperature-independent thermal properties is common across the majority of the models proposed in literature.

Around the same time, De Carli, Tonon, Zarrella, and Zecchin (2010) presented a different approach to studying the thermal behavior of deep borehole GHEs. Their model was based on the thermal-electrical analogy and was called CaRM (Capacity Resistance Model). The model uses lumped capacities and thermal resistances to solve the heat transfer in unsteady state by finite difference method. The ground surrounding the

borehole is discretized as horizontal slices in the vertical direction and annular regions in the radial direction (Figure 6).

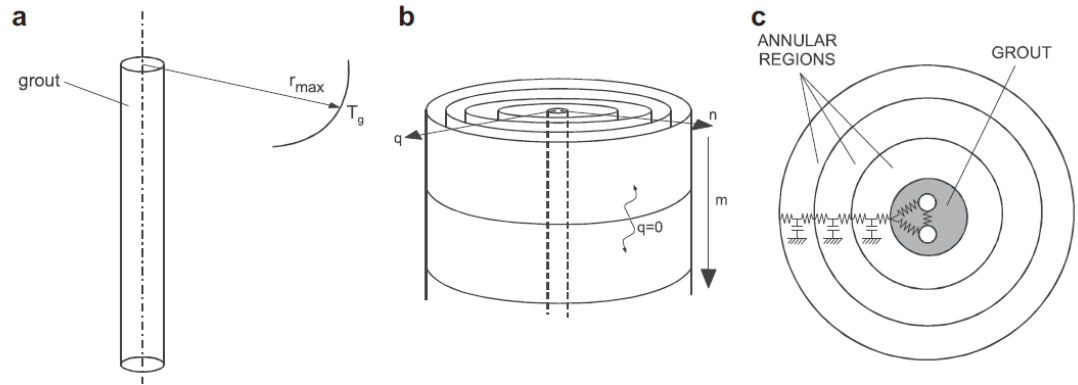


Fig. 3. Approach of the model.

Figure 6. Discretizing the ground in CaRM model (De Carli et al., 2010)

In addition, the model considers the fluid flow pattern, type of borehole configuration (single U-tube, double U-tube or coaxial), heat transfer between the fluid and the pipe, heat transfer between the pipe and the grout and from the grout to the ground. Finally, the interaction between different boreholes in a borehole field in various configurations is also taken into account. Results were validated against experimental data obtained from a thermal response test (TRT) and from survey data from an office building equipped with a ground coupled heat pump (GCHP) and double U-tube GHEs. This model neglects the thermal capacity of the grout and the fluid, which has influence in short term analysis, and the axial heat transfer from the top and bottom of the heat exchanger to the surface and the deep ground respectively. Both assumptions are reasonable because of the slender geometry of these types of GHE.

In a later paper, Zarrella, Scarpa, and De Carli (2011) presented an improved version of the previous model CaRM, in which the thermal capacitance of both the borehole backfilling material, i.e. the grout, and the heat carrier fluid were considered. The new release revealed the importance of these properties for short-time step analysis.

So far, this thermal capacitance-resistance model covered the simulation of deep boreholes in different configurations, single and double U-tubes as well as coaxial pipes. Two years later, a paper titled “Heat transfer analysis of short helical borehole heat exchangers” was published (Zarrella & De Carli, 2013). The authors presented a new model to analyze the thermal behavior of shallow vertical helical GHEs based on their own previous thermal circuit simulation tool. They called it CaRM-He for “capacity resistance model for helical heat exchangers”. Additional aspects were introduced in order to account for the new geometry, namely the heat transfer in the axial direction for both the grouting material and the surrounding ground, the heat capacity of the grout and the heat carrier fluid, and the interaction between the ground and the environment. These are all aspects that affect the behavior of the shallow vertical helical GHE because they are normally buried at depths where seasonal temperature swings are still noticeable (Figure 7). The approach of the model can be graphically seen in Figure 8.

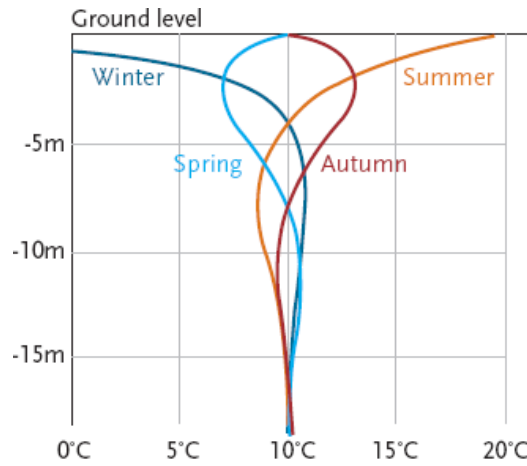


Figure 7. Seasonal temperature “swing” in the shallower layer of the ground (Source: www.cibsejournal.com)

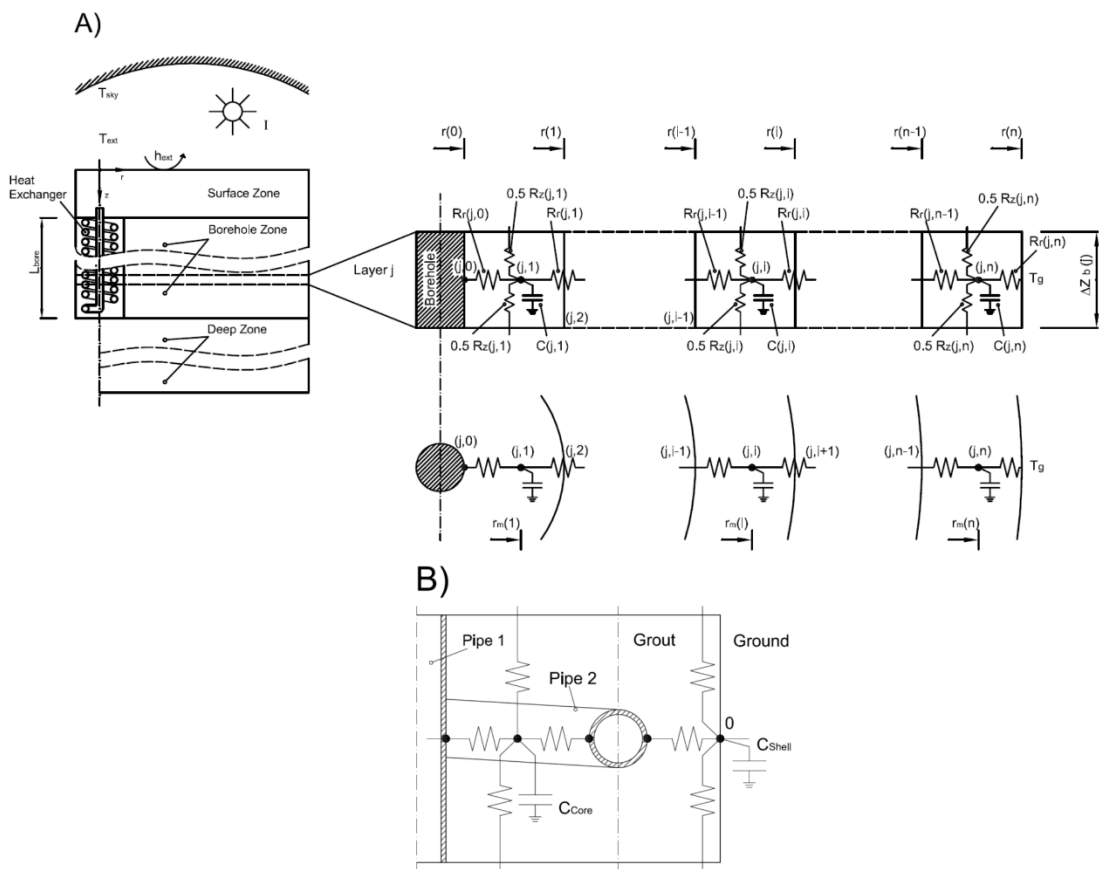


Figure 8. The approach of CaRM-He (Alvarez-Revenge, Amende, & Zarrella, 2015)

CaRM-He model was tuned up and validated by experimental measurements from a facility in Erlangen (Germany). Furthermore, the authors performed a sensitivity analysis and a parametric analysis showing that the interaction with the environment and the axial heat transfer had indeed a considerable influence on the performance of the GHE and thus in the inlet and outlet temperature of the heat carrier fluid, which ultimately affects the energy efficiency of the GSHP system. The model still neglects the possible influence that moisture content changes or ground water movement may have over the performance of the system. In this thesis, field testing measurements from the operation of the helical GHEs at the REHAU® Montana ecosmart house were used as input for this model in order to have a gauge of the ability of the model to predict the thermal behavior of this type of ground heat exchangers in different conditions, as it will be extensively explained in later sections of this thesis.

It is clear that the efficiency, i.e. COP, of a GSHP system is affected not only by the performance of the heat exchanger, but also by the performance of the heat pump equipment as it is coupled with the building side. In the design process, it would become handy to have an integrated tool that not only simulated the GHEs but also the operation of the heat pump given the building heating and cooling loads. In this regard, Zarrella, Capozza, and De Carli (2013) presented a model – called GeoHP-Calc – that is capable of this integration.

Moch, Palomares, Claudon, Souyri, and Stutz (2013) investigated the performance of the helical GHE by adapting the previously reported “ring-coil” model from Rabin and Korin (1996). They considered the effects of the ground water freezing

around the heat exchanger when the temperatures fall below 32°F (0°C), which can be common in closed-loop GSHP applications. The article demonstrated the importance of taking this phenomenon into account, but still this complex process had to be simplified by means of several assumptions. For example, the possibility of the freezing/thawing cycles generating an air cavity in the close proximity to the heat exchanger wall that changes the effective thermal resistance of that layer was not considered. Nevertheless, the model can be used to predict heat exchanger outlet temperatures and long term ground temperature evolution by inputting simulated building loads. In a later work, the same authors compiled and adapted the aforementioned ring-coil model and the annular cylindrical conduit model (Nir et al., 1992) and performed a computer based case comparison (Moch, Palomares, Claudon, Souyri, & Stutz, 2014). They took into account the weather variations at the surface boundary conditions as well as the effect of freezing of the earth if temperatures were low enough. They found that the latter was computationally more efficient; although, both approximated to the experimental results with similar degree of convergence. Furthermore, in a recent paper, the same team investigated the long term effects that different types of load cycling would have over the overall performance of the system when coupled to a low-energy house (Moch, Palomares, Claudon, Souyri, & Stutz, 2015).

In almost every model reported in literature, the axisymmetric assumption is made. However, this assumption is only good as far as one single GHE is analyzed or if the influence of neighboring exchangers can be neglected. Care must be taken if a number of GHEs is going to be put in place in the bore field so the interaction between

adjacent GHEs can be accounted. For instance, Dehgham, Sisman, and Aydin (2015) investigated the effect of the distance between GHEs in the overall performance of the system. They used the computer tool COMSOL[®] to simulate the resulting temperature distribution and thus optimize the spacing between GHEs.

Commercial Design Tools

An example of commercial software to design GHEs for geothermal systems is Ground Loop Design[®] (Gaia Geothermal, 2012). This tool makes use of two different theoretical approaches for vertical boreholes: one is based on the solution for heat transfer from a cylinder buried in earth presented by Carslaw and Jaeger (1947), and the other is based on the infinite line-source heat conduction problem in homogeneous medium developed by Eskilson (1987). It also has a module for calculating various types of horizontal GHEs. So far, there are no theoretical models integrated in commercial tools that allow for the calculation of short boreholes with a helical shape.

EXPERIMENTAL SETUP

In this section, the experimental site is presented, along with an overview of its HVAC systems. Then the experimental apparatus and the influencing environmental conditions are explained. Finally, the experiments are described in detail. The purpose of these experiments is to analyze the performance of the helical-shaped GHEs. A list of sources of error and uncertainty analysis theory pertaining to this research is introduced.

The REHAU® Montana Ecosmart House Project

The REHAU® Montana ecosmart house (RMEH) is a 4,550 ft² (423 m²) demonstration house built in 2012 in Bozeman, Montana as a result of the joint efforts of various corporate sponsors and the MSU's Creative Research Lab (Figure 9). The purpose of the project is to expand the body of knowledge in residential building and human sustainability. The house was designed and built in a holistic way, incorporating cutting-edge technology in architectural, mechanical systems and human accessibility design. The main goal is to investigate and optimize these systems and construction materials to ultimately reduce greenhouse gas emissions and overall energy consumption, increase comfort and reduce life-cycle costs. This is accomplished primarily by combining a tight envelope with energy efficient heating and cooling systems. At the same time, the house will provide a comfortable and friendly environment for the occupants under the integrated concept of aging-in-place and inter-generational living. Interdisciplinary collaboration and public outreach is also a key objective of the project. The house is Energy Star® and LEED® for homes, certified with a HERS index score of

32 (near net-zero). More information about the RMEH project can be found at

www.na.rehau.com/ecosmart.



Figure 9. View of the REHAU® Montana ecosmart house

Architectural features include:

- Structural insulated panels (SIPs)
- Insulating concrete forms (ICFs)
- Vinyl window and door designs
- Sunlight responsive thermochromic (SRT) glazing
- Hybrid curtain wall

Human sustainability features:

- Wheelchair accessible
- Elevator shaft connects three stories

- Telemedicine communication technology
- Tambour cabinetry

Mechanical systems:

- Geothermal water-to-water heat pump
- Closed-loop ground heat exchangers
 - Helix probes
 - Double U-tubes
- Low NO_x condensing boiler
- Earth-Air heat exchangers
- Heat recovery ventilation (HRV) unit
- Solar thermal panels
- Heat-sink storage
- Radiant heating and cooling system
- Forced-air heating and cooling system
- Chilled acoustic ceiling
- Snow and ice melting
- Sprinkler fire suppression system
- PEX plumbing
- Web-based climate control technology

This “living laboratory” is an incredibly valuable research tool for energy efficiency. It is filled with more than 300 sensors and data acquisition (DAQ) systems to provide continuous monitoring and recording of temperatures and other physical

properties across the architectural and mechanical systems of the house, including a dedicated weather station in the backyard. Students from various disciplines were heavily involved in the design and construction process, a great example of hands-on teaching and collaborative work. As part of the RMEH project, a comprehensive set of experiments was carried out to assess the performance of the various systems in different real-life scenarios. Since some of the mechanical systems are redundant; innovative systems can be effectively compared to more traditional ones, for instance, ground-coupled heat pump (GCHP) versus gas-fired boiler for heat production, or radiant floors versus forced-air for space heating and cooling. Findings of these experiments will be made available to the public through the websites of the major corporate sponsor and the Department of Mechanical and Industrial Engineering at MSU. Other research works directly derived from the RMEH Project include a PhD dissertation about Green's function application for earth-air heat exchangers (Denowh, 2015) and a conference paper about in-situ testing of shallow helical GHEs accepted for presentation at the ASHRAE Winter Conference 2016 in Orlando, Florida (Alvarez-Revenga et al., 2015).

Overview of the HVAC System at the RMEH

The mechanical systems at the RMEH are rather complex for what is typical in a residential construction (Figure 10). During the summer, the closed-loop GSHP takes care of the cooling loads. The chilled water it produces is circulated to and from a 120 gal (450 L) buffer tank. This is the primary loop of the hydronic system. The control system is configured in such a way that the GSHP is commanded to maintain a temperature setpoint in this buffer tank, within a given dead band. This setpoint can be manually

determined by the user, or be the result of a reset schedule calculation. From the buffer tank, a secondary loop distributes the cold supply to the different end consumers or terminal systems, i.e. the radiant floor system, the cold ceiling panel system, and the water-to-air coil in the air handling unit for the centralized forced-air system.

During the winter, the hot water supply can be produced either with the GSHP in heating mode or with a natural gas fired condensing boiler. The “terminal units” or consumers are in this case the radiant floor system, the water-to-air coil of the air handling unit, and the snow and ice melting system. The domestic hot water (DHW) production is attended only by the boiler, but assisted with solar thermal panels. During summer time, if the DHW demand is covered and yet there is an excess of heat collected by the solar panels, the energy can be diverted to a heat sink. The heat sink is a grout block buried in the front porch area of the house with embedded PEX pipe loops in it. Energy is stored by heat exchange with the diverting loop coming from the solar panels (the “charging” loop) and in turn, another loop (the “discharge” loop) is connected in series with the geothermal loop. Therefore, the GSHP heating system can make use of this extra energy during the first stages of the heating season, by increasing the efficiency of the GSHP and thus producing some operational cost savings.

In order to control the mechanical devices and the comfort level, a building automation system called REHAU Smart Controls –or RSC– is in place at the RMEH. This is a convenient way to automatically program, monitor and integrate complex control sequences to optimize energy consumption and at the same time, provide the research team with a flexible tool to carry out different tests in a variety of scenarios. The

control system also allows for web-based access for end users as well as for remote administrators for diagnostic purposes.

Experimental Apparatus

The experiments were made possible through the use of the in-situ systems. The experimental apparatus explained hereafter consist of the subset of systems associated with the experiments, i.e. the geothermal heating and cooling primary production equipment, the heat carrying fluids, the circulating pumps, the ground heat exchangers and finally the sensors and data acquisition system. See APPENDIX A for manufacture's specification sheets.

Ground Source Heat Pump

The primary heating and cooling equipment consists of a closed-loop ground source heat pump (GSHP). This unit is a one-of-a-kind piece of equipment custom manufactured by AAON[®]. The water-to-water GSHP has two 34,500 Btu/hr (10.1 kW) Copeland[®] digital scroll compressors in two separate refrigerant circuits that allow for versatile modulation from 7.5% to 100% of total power range (15% to 100% on each compressor). It is also equipped with a proprietary internal control system that works in tandem with the central building management system. This way, the unit can be adjusted to meet any type of custom need for research purposes. Geothermal fluid-to-refrigerant and refrigerant-to-water heat exchange occur in two plate type heat exchangers. Four thermostatic expansion valves and two reversing valves complete the main components of the vapor-compression circuit of this fully reversible unit (Figure 11).

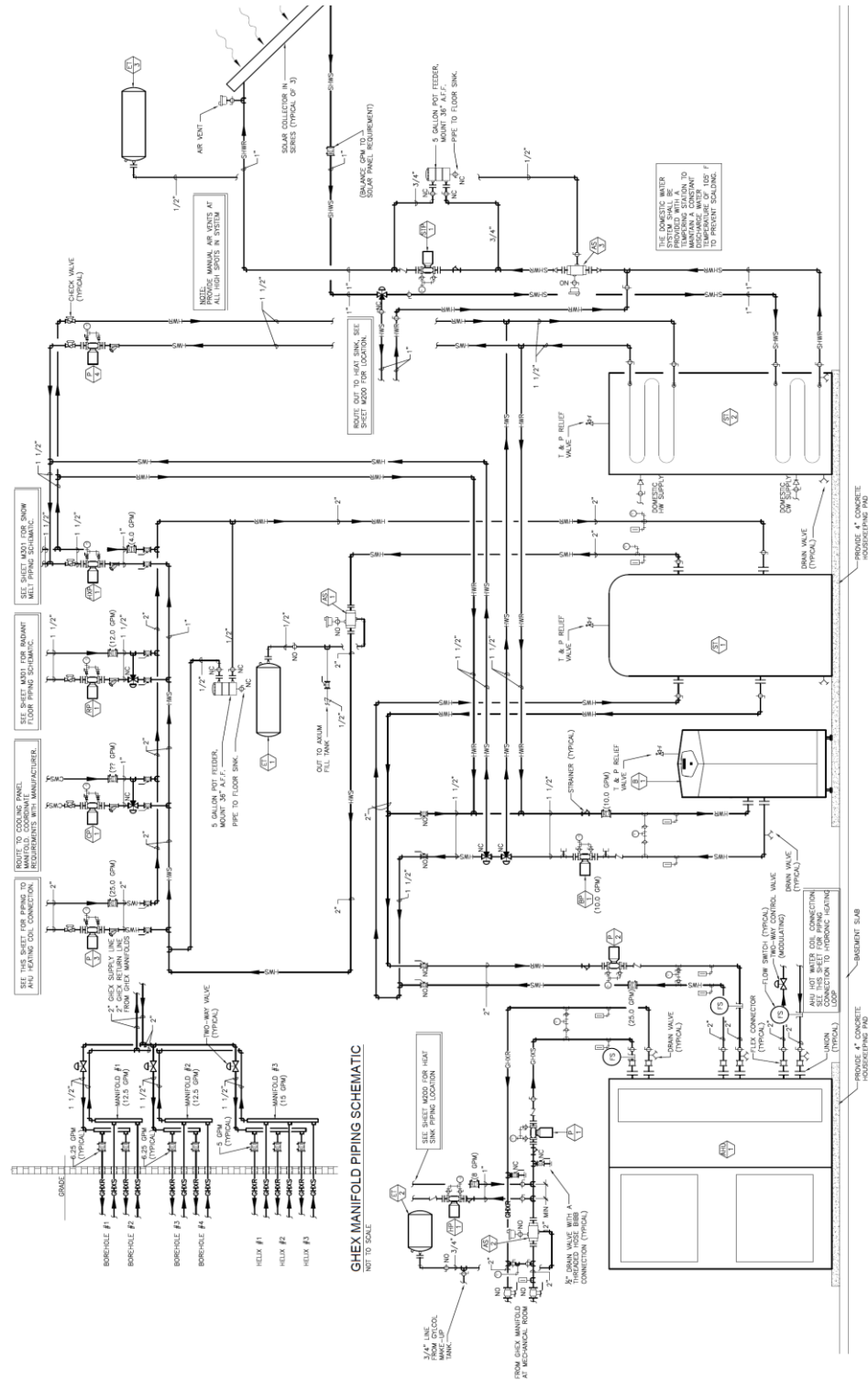


Figure 10. Mechanical piping schematics (Source: Project drawings from Energy 1 – RMEH’s mechanical contractor)



Figure 11. AAON® ground source heat pump in the mechanical room at the RMEH

Geothermal Fluid

The geothermal fluid consists of 20% propylene glycol and 80% water by volume. The geothermal field contains three 18 ft (5.5 m) deep 1" RAUGEO HELIX® PEXa pipe helical probes installed in parallel and four 300 ft (91.4 m) deep 1" RAUGEO® PEXa pipe double U-tube vertical probes that can be independently isolated through manifolds (Figure 12). For the purpose of this thesis, only the three helical GHEs were used.



Figure 12. REHAU® PROBALANCE® XP manifolds in the mechanical room at the RMEH are used to distribute and regulate the flow of the geothermal fluid in the ground loops. Headers are made of extruded polypropylene pipe

Circulating Pumps

The circulating pumps involved in the tests are high efficient variable speed in-line Grundfos® circulators (Figure 13). Specific models and functions are listed in Table 1.

Table 1. Circulators used in the Experiments

Element	Function	Model
P-1	Geothermal fluid circulator	Magna 65-120F
P-2	Primary water circulator	Magna 32-100F 165



Figure 13. One of the Grundfos® circulators in the mechanical room at the RMEH. They are high efficient electronically controlled variable speed and have different types of automatic settings

Helical Ground Heat Exchangers

Each helical GHE –or simply “the helix”– is made of a pre-shaped 1” PEXa pipe and is 12 ft (3.7 m) long with a pitch between turns of 4.3 in. (0.11 m) when fully expanded. The helix diameter is 15 in. (0.38 m). During construction, the helix was inserted into an 18 ft (5.5 m) deep and 18 in. (0.46 m) diameter hole created by an earth auger (Figure 14) and backfilled with the same native soil extracted during the excavation (Figure 15). The helical heat exchangers are located in the backyard of the residence and spaced 11 ft (3.4 m) apart from each other. See Figure 17 for a graphical view of the location of the GHEs within the property parcel.



Figure 14. Earth auger creating the borehole for a helix probe. This is one of the key reasons why this type of installation can be a cost-effective alternative to the conventional deep boreholes (Source: www.rehau.com)



Figure 15. Installation of one the helical GHE at the RMEH

A 6 ft (1.8 m) deep horizontal trench conducts all the pipes from the GHE heads towards the mechanical room (Figure 16), which is located in the northwestern corner of the house in the basement. Consequently, there is approximately 128 ft (39 m) of horizontal run, which is assumed to be insulated with closed-cell foam pipe sleeves (Figure 17).



Figure 16. View of the distribution trench and detail of entry to the house through the wall during construction

Sensors

In order to assess the performance of the GHEs, the temperature of the fluid entering and leaving the GHE and its mass flow rate must be known. Entering water (or antifreeze fluid) temperature (EWT) is a typical designation for the temperature of the fluid entering the heat pump; thus, it is the temperature at the outlet of the ground heat

exchanger assuming there are no losses between GHE and GSHP. Similarly, leaving water temperature (LWT) is the temperature of the fluid leaving the heat pump, i.e. the temperature at the inlet of the exchanger. EWT is an important parameter since it influences the efficiency (COP) of the heat pump. Type-T thermocouples were used to measure EWT and LWT.

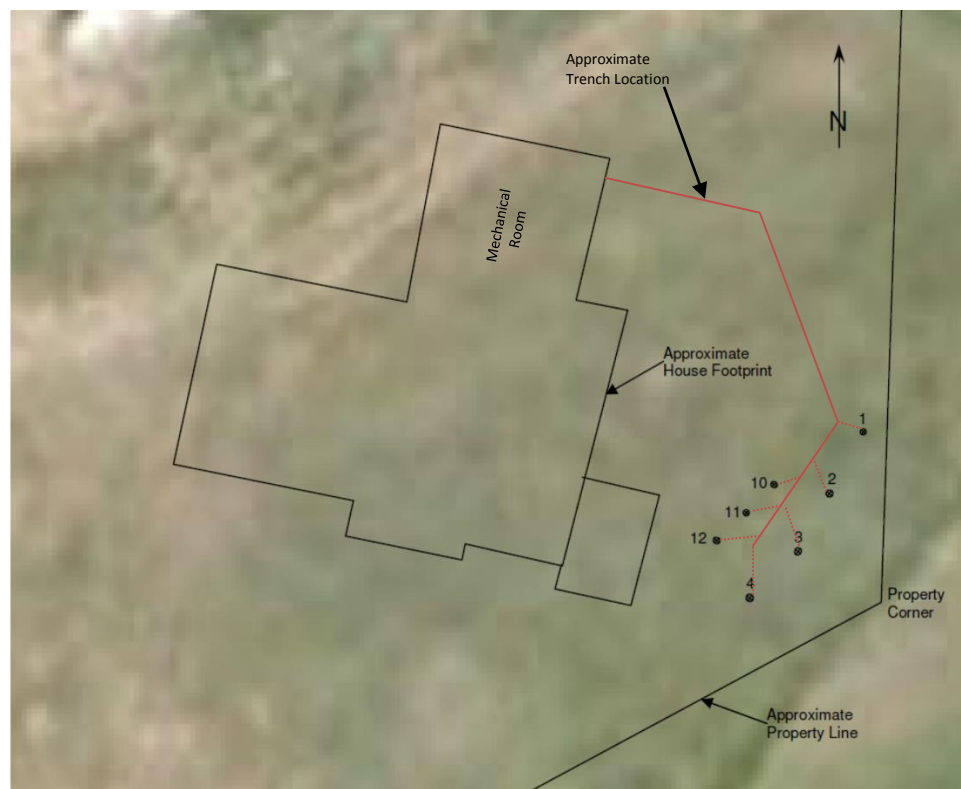


Figure 17. Location of the geothermal boreholes at the RMEH with respect to the footprint of the house. Boreholes marked as 1, 2, 3 and 4 correspond to the location of the deep vertical GHEs and 10, 11 and 12 represent the location of the three helical GHEs

The type-T thermocouples were attached to the pipes in the mechanical room, secured with a zip tie and covered with asphaltic tape. Then, the entire assembly was covered with a sleeve of closed-cell foam pipe insulation to avoid external influence so

the sensor reflected the temperature of the fluid within the pipes as closely as possible (Figure 18).



Figure 18. Detail of a pair of type-T thermocouple sensors attached to the copper pipes and covered with asphaltic tape in the mechanical room at the RMEH

The volumetric flow rate of the geothermal loop was measured with a paddle-wheel flowmeter (Figure 19). The Badger Meter® Data Industrial 228BR2005 model sensor outputs a signal at a frequency proportional to the flow rate. This frequency is then converted to a linear analog 4-20 mA signal by a Badger Meter® Data Industrial Series 310 programmable loop-powered analog transmitter. Figure 20 shows a simplified wiring diagram of the flow meter. Mass flow rate is easily derived from volumetric flow rate since the density of the geothermal fluid mix is known.

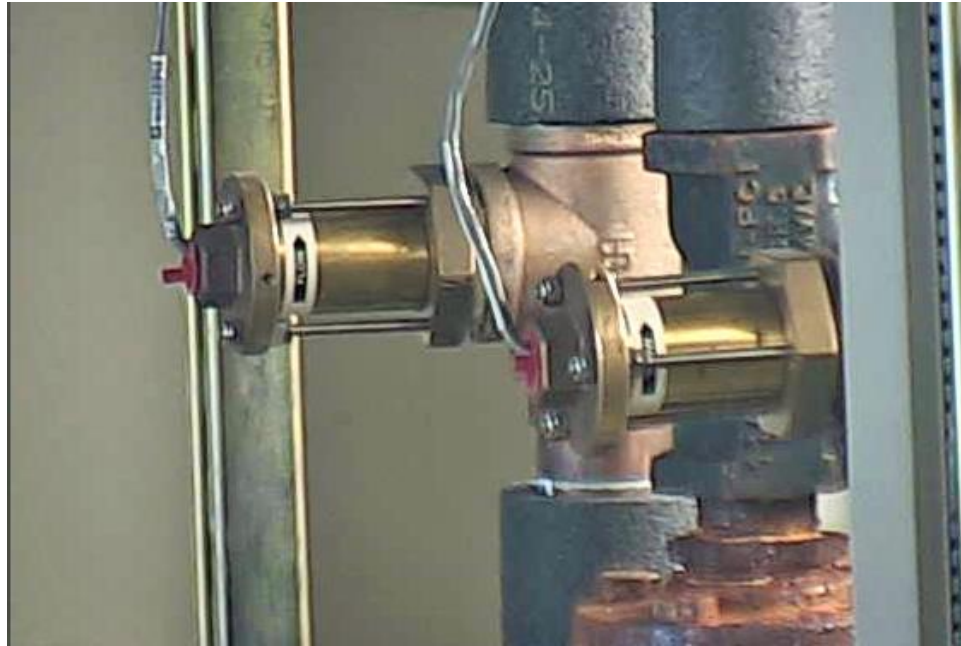


Figure 19. Two Badger Meter® Data Industrial 228BR2005 paddle-wheel flow sensors in the mechanical room at the RMEH. These sensors feature a six-blade impeller design with a non-magnetic sensing mechanism to output a square signal at a frequency that is proportional to the flow rate

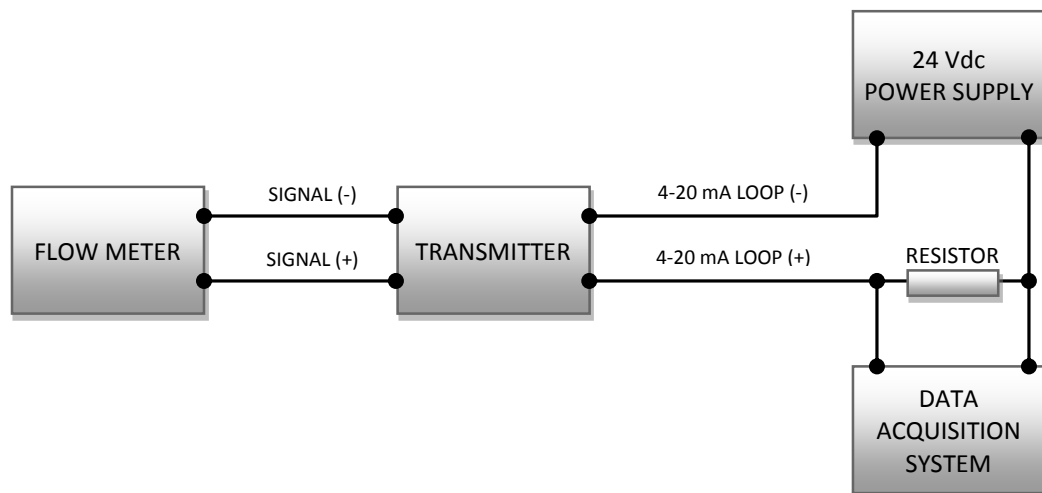


Figure 20. Simplified wiring diagram of the flow meter. Flowmeter outputs frequency, the transmitter transforms frequency into scaled current, and the resistor produces a voltage drop that is read by the data acquisition system module

Data Acquisition System

The main DAQ system consists of a central computer with LabVIEW 2011 software and National Instruments (NI) monitoring equipment. The computer is located in a special place in the basement adjacent to the mechanical room and it stores all data collected from the monitoring equipment. The bulk of the temperature sensors scattered throughout the house is connected to various 32-channel amplifiers for low voltage (SCXI-1102) via isothermal terminal blocks with cold-junction compensation sensor (SCXI-1102). All the modules are encased in a rack-mounted SCXI-1001 chassis near the main computer (Figure 21. Left). A second DAQ system is located in the mechanical room to measure additional sensors. This DAQ system is a reconfigurable embedded chassis with integrated intelligent real-time controller called cRIO-9073 with various modules (Figure 21. Right). The modules pertaining to this research include a ± 10 V analog input with 16-bit resolution analog-to-digital converter (NI 9215) and a ± 80 mV thermocouple input with 24-bit analog-to-digital converter and cold-junction compensation for high-accuracy thermocouple measurements (NI 9211).

Environmental Conditions

Two environmental conditions have the most influence on the performance of the shallow depth helical GHEs: soil conditions and weather.

Soil Thermal Properties

Soil conditions are clearly the most important parameters affecting the performance of ground heat exchangers in general, since heat exchange occurs primarily

by conduction. Therefore, a lot of effort is invested in determining the ground formation thermal conductivity and thermal diffusivity.

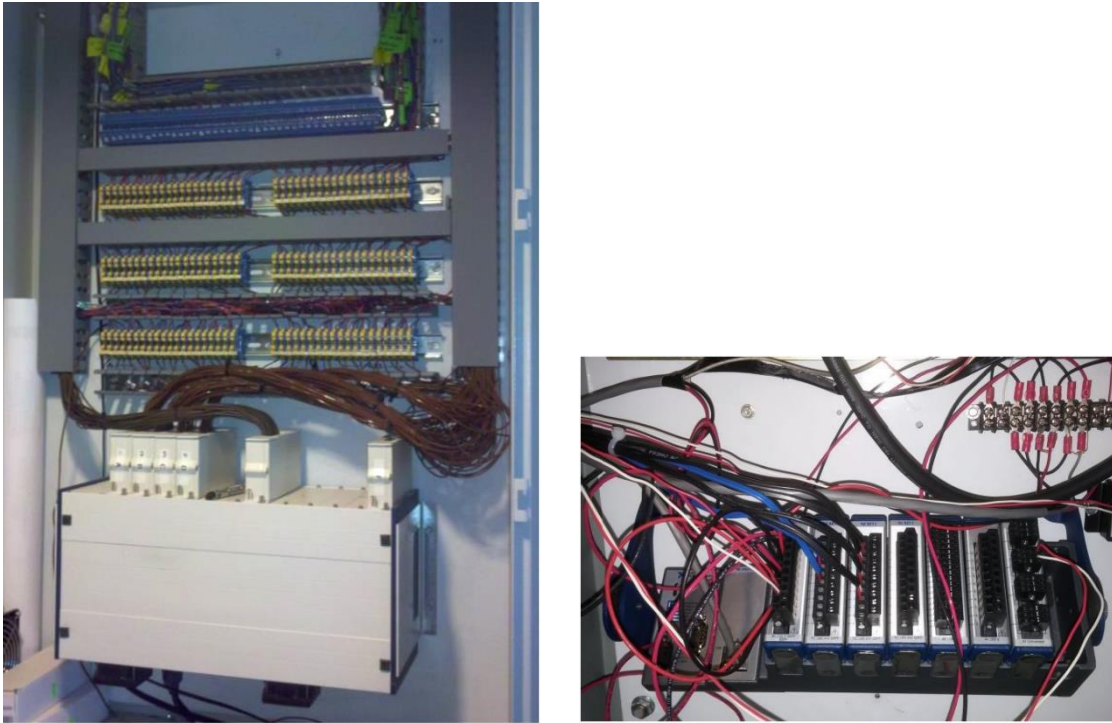


Figure 21. Data acquisition system at the RMEH. Left: NI® SCXI-1001 DAQ system with distribution DIN rails on top for wiring. Right: View of the NI® cRIO-9073 DAQ system in the mechanical room at the RMEH (Denowh, 2015)

Local soil formation conditions were determined at the RMEH through ground thermal response tests performed during building construction. These tests were carried out for one of the 300 foot deep double U-tube boreholes and also for one of the helical probes. However, the thermal conductivity for the helical probe test could not be determined because the shape factor of the short helical GHE invalidates the analysis. The thermal response test for the deep borehole showed a thermal conductivity of 1.26 Btu/(hr·ft·°F) (2.18 W/(m·K)).

Thermal properties of the soil for the deep boreholes do not necessarily have to match those of the shallow layers of the ground, given the natural non-uniform layered configuration present in typical ground formations. They represent the properties that an equivalent uniform soil formation thermal response test would yield. A drilling log was performed on site during the drilling works, and soil samples were collected for analysis. The soil corresponding to the helical borehole was determined to be a moderately plastic and sticky clay, group symbol CL according to Unified Soil Classification System (USCS) – ASTM D-2487 (ASTM International, 2011). A summary of the field classification according to ASTM D-2487 standard is shown in Table 2.

Table 2. Summary of Field Classification (USCS) in helical borehole at the RMEH

Property	Value
Sample Location	Borehole 12 (see Figure 17)
Depth	0-15 (8) feet
< 200 Fines	85%
200-4 Sand	15%
>4 Gravel	0%
Dilatency	None
Toughness	High
Dry Strength	High
Ribbon	Medium
Liquid Limit	Moderate
Stickiness	Medium
Group Symbol	CL
	Lean clay with sand.
Remarks	Liquid limit >35 and <45. Most of sample took >30 sec to settle in a jar of water

When there are budget limitations, as is typically the case in residential construction, it is not possible to perform thermal response tests to investigate the formation thermal properties. Besides, the standard line source methodology prescribed

by ASHRAE (ASHRAE, 2015) is incompatible for boreholes as short as a typical helical GHE. Consequently, an alternative method to estimate the soil thermal properties is by means of tables from Chapter 34 of the 2015 ASHRAE Handbook – HVAC Applications (ASHRAE, 2015). These tables correlate various types of common soils, rocks and common grouts with a range of thermal conductivity and thermal diffusivity values. Geological information can be gathered from national or local databases or from well logs reported in the surrounding area. Table 3 shows a selection of tabulated thermal properties for common soil and rock formations.

Table 3. Thermal Properties of Selected Soils and Rocks. Adapted from Table 4 in Chapter 34 of the ASHRAE Handbook – HVAC Applications (ASHRAE, 2015)

		Dry Density, lb/ft ³	Conductivity, Btu/h·ft·°F	Diffusivity, ft ² /day
<u>Soils</u>				
Heavy clay,	15% water	120	0.8 to 1.1	0.45 to 0.65
	5% water	120	0.6 to 0.8	0.5 to 0.65
Light clay,	15% water	80	0.4 to 0.6	0.35 to 0.5
	5% water	80	0.3 to 0.5	0.35 to 0.6
Heavy sand,	15% water	120	1.6 to 2.2	0.9 to 1.2
	5% water	120	1.2 to 1.9	1.0 to 1.5
Light sand,	15% water	80	0.6 to 1.2	0.5 to 1.0
	5% water	80	0.5 to 1.1	0.6 to 1.3
<u>Rocks</u>				
Granite		165	1.3 to 2.1	0.9 to 1.4
Limestone		150 to 175	1.4 to 2.2	0.9 to 1.4
Sandstone			1.2 to 2.0	0.7 to 1.2
Shale, wet		160 to 170	0.8 to 1.4	0.7 to 0.9
	dry		0.6 to 1.2	0.6 to 0.8

It follows a sample calculation for estimating the thermal conductivity and thermal diffusivity of the soil at average depth for a helical GHE (8 feet [2.4 m]) at the RMEH. The soil sample from the helical bore was reported to be 85% clay and 15% sand

with moderate water content (Table 2). Water content is assumed to be in the mid-range (10%). Dry density is unknown, so a medium range value is considered between light and heavy soil: 100 lb/ft³ (1602 kg/m³). Finally, an average value from the range of thermal conductivity and diffusivity will be considered. Interpolating and applying a proportional weight for each component, clay and sand, the thermal properties are summarized in Table 4.

Table 4. Sample calculation of estimated thermal properties in soil at the RMEH

Soil fraction	% in soil	Water content (%)	Conductivity (Btu/h·ft·°F)	Weighted conductivity (Btu/h·ft·°F)	Diffusivity (ft ² /day)	Weighted diffusivity (ft ² /day)
Clay	85	10	0.64	0.54	0.51	0.43
Sand	15	10	1.29	0.19	1.25	0.19
Total	100			0.74		0.62

As can be seen in Table 4, the thermal conductivity was estimated in 0.74 Btu/h·ft·°F (1.28 W/(m·K)) and the thermal diffusivity was estimated in 0.62 ft²/day (0.667·10⁻⁶ m²/s).

Weather Conditions

The most important weather factors affecting the performance of a helical GHE are the outdoor temperature and the solar radiation.

Helical GHEs are installed typically in the first 6 to 18 feet (1.8 to 5.5 m) of the ground. This means they are laid at a depth where there is still influence from the yearly temperature swings (recall Figure 7). Soil temperature as a function of depth can be approximated with different degrees of accuracy given the atmospheric air temperature of a location (Kusuda & Achenbach, 1965).

The RMEH has various means of recording the atmospheric air temperature. A dedicated weather station placed in the backyard collects dry and wet bulb temperatures, as well as wind direction and speed, and precipitation. The NI[®] DAQ has one of its thermocouple sensors placed outside so it can measure outdoor air temperature. Finally, the REHAU[®] Smart Controls (RCS) has also an outdoor temperature sensor. Nowadays, the design engineer has several resources to investigate the outdoor temperature. For example, the National Oceanic and Atmospheric Administration (NOAA), through the National Centers for Environmental Information (NCEI) (formerly the National Climatic Data Center [NCDC]), offers global historical weather and climate data, including 30-year climate normals (NOAA, 2015).

Climate of Bozeman. A compilation of the most relevant climate facts about Bozeman, Montana is listed below. The source is NOAA (2015) unless otherwise specified. A climate graph of Bozeman can be seen in Figure 22.

- Elevation: 4816 ft (1468 m)
- Type (Wikipedia, 2015): dry continental climate (Köppen code: Dfb)
- Monthly normal low: 23.4°F (-4.8°C) (December)
- Monthly normal high: 67.1°F (19.5°C) (July)
- Annual mean temperature: 44.6°F (7°C)
- Yearly average precipitation: 19.2 in (490 mm)
- Yearly average snowfall: 86.1 in (218.7 cm)
- Heating degree days (HDD): 7701 (Fahrenheit; base 65)
- Cooling degree days (HDD): 254 (Fahrenheit; base 65)

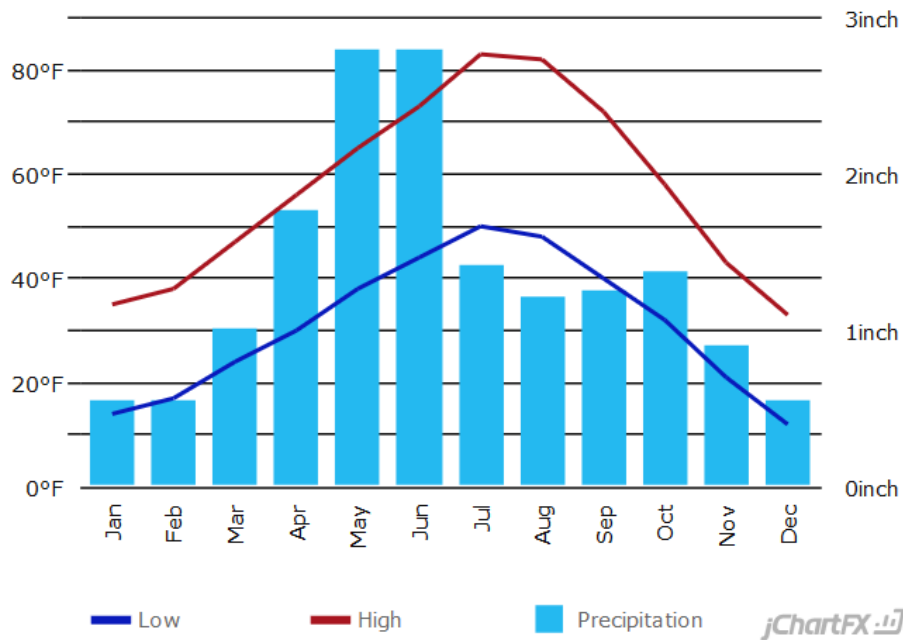


Figure 22. Climate graph of Bozeman, Montana. Source: www.usclimatedata.com

Solar Radiation. Short and long wave solar radiation affects the conditions at the soil surface level. Short wave radiation corresponds to the direct and diffuse solar radiation, whereas the long wave radiation refers to the exchange of energy with the sky. Global horizontal solar radiation at the site was estimated by retrieving average hourly statistics provided by EPA's energy modeling software for the Gallatin Field weather station near Bozeman, Montana (EnergyPlus, 2015).

Description of Experiments

In-situ tests pertaining to this thesis were made in two phases, spanning two summers and two winters. The first phase includes one 3-day test in cooling mode (August 2nd to 5th, 2014) and one 5-day test in heating mode (February 9th to 14th, 2015).

The second phase includes one 42-day test in cooling mode (August 11th to September 22nd) and a 29-day test in heating mode (October 13th to November 11th, 2015). The second phase complements the first phase in two ways: 1) it provides insight on long term heat exchanger behavior and 2) it enables for more accurate performance calculations due to improved control and sensor installation. Table 5 provides an overview of testing dates. Cooling Test III and Heating Test III were intended to complement the information and provide additional insight about the performance of the helical GHE in different conditions. Experimental and simulation results for these experiments can be found in APPENDIX C.

Table 5. Dates of the experiments

Phase	Test ID	Start Date	Finish Date	Span (days)	Temp. Sensors
I	Cooling Test I	8/1/2014	8/5/2014	4	Original position
	Heating Test I	2/9/2015	2/14/2015	5	
II	Cooling Test II	8/11/2015	9/22/2015	42	Improved sensor installation
	Heating Test II	10/13/2015	11/11/2015	29	
	Cooling Test III	7/15/2015	7/20/2015	5	
	Heating Test III	12/11/2015	12/11/2015	1	

Cooling Test I

During Cooling Test I, a steady load was applied to the geothermal loops, in a similar way as it is done in a formation thermal conductivity test (ASHRAE, 2015), where a steady load is applied to a test bore to infer the formation thermal properties, namely the formation thermal conductivity and thermal diffusivity. However, these thermal conductivity tests are based on the solution of the infinite line source method, which is not suitable for the helical GHE due to its shape factor. Nevertheless, applying a

steady load can simplify the performance calculations because a steady state situation can be assumed after a sufficient period of heat exchange time as will be seen later in the Experimental Results section. In order to apply a steady load to the GHEs, the control system was set in such a way that the GSHP was constantly working with only one compressor at 15% capacity. In turn, a sufficiently high load was set on the building side to avoid temperature on the buffer tank becoming too low so as to make the system “believe” the demands had been met; this way the compressor was constantly running. The GSHP’s compressor capacity was set based on the initial estimated performance of the helical GHE according to product literature (See equipment specifications in APPENDIX A).

Heating Test I

During Heating Test I, a slightly different approach was taken. The GSHP was initially switched on under a control scenario that allowed the GSHP’s own internal controls to modulate capacity to meet real building demands. More specifically, the controls were set up in such a way that the GSHP worked to maintain a given setpoint in the buffer tank. In order to control the demand on the building side, a PID (proportional-integral-derivative) control loop was set up on the supply-return mixing valve arrangement of the radiant floor heating system so a fixed temperature difference (ΔT) was maintained. This control strategy resulted in the system cycling and providing intermittent heating pulses to the GHEs. Later during the same experiment, a steady load was applied in order to investigate the steady state performance. This was done in a similar way to the previous Cooling Test I, but this time the capacity was set to one

compressor at 60% running continuously. Then, the ΔT on the building side was adjusted manually by observation until the temperatures entering and leaving the geothermal loops (EWT & LWT) stabilized. After 72 hours in this configuration, LWT was approaching 33°F, which falls within the recommended temperature range for geothermal fluid entering the GSHP (ASHRAE, 2015). *Phase I* testing conditions are summarized in Table 6.

Table 6. Phase I (Cooling Test I and Heating Test I) testing conditions

Description	Value (I-P Units)	Value (SI Units)
<u>Geothermal Fluid</u>		
Type of fluid	Propylene glycol 20% vol.	
Average volumetric flow rate in loop (cooling test – heating test)	9.0 – 9.9 gpm	0.57 – 0.62 l/s
Average fluid temperature during test (cooling test – heating test)	77.5 – 44.1°F	25.3 – 6.7°C
<u>Layout and Geometry of Heat Exchanger</u>		
Type of probe	Helical	
Number of probes	3	
Spacing	11 ft	3.4 m
Depth of installation	6 ft	1.8 m
Length of probe	12 ft	3.7 m
Diameter of borehole	18 in	0.46 m
Diameter of helix	15 in	0.38 m
Pitch between Turns	4.3 in	0.11 m
Total length of pipe	131 ft	40 m
Material of pipe	Cross-linked polyethylene (PEXa)	
Inside diameter	0.875 in	22.2 mm
Outside diameter	1.125 in	28.6 mm
Density of material	58.68 lb _m /ft ³	940 kg/m ³
Thermal conductivity of material	0.24 Btu/(hr·ft·°F)	0.41 W/(m·K)
<u>GSHP Capacity Control</u>		
Number and speed of compressors used (cooling test)	1 compressor – fixed 15%	
(heating test)	2 compressors – variable, then fixed 60%	
<u>Properties of Ground</u>		
Type of soil	85% clay – 15% sand	
Estimated thermal conductivity	0.74 Btu/(hr·ft·°F)	1.28 W/(m·K)
Estimated thermal diffusivity	0.62 ft ² /day	0.667·10 ⁻⁶ m ² /s
Backfilling material	Native soil (no grout)	
<u>Climate Data of Location</u>		
Location	Bozeman, MT	
Annual mean temperature	44.6°F	7°C
Annual variation (semi-amplitude)	21.9°F	12.2°C
Average outdoor temperature during test (cooling test – heating test)	70.0 – 44.4°F	21.1 – 6.9°C

Cooling Test II

Cooling Test II ran for 42 days during the summer of 2015. For this test, a few improvements were implemented for the whole HVAC system at the RMEH that affected the outcome of the experiments compared to *phase I*. Communications between the REHAU® Smart Controls (RSC) and the GSHP control board were upgraded thanks to a Modbus protocol connection. This enabled the GSHP to be able to control the building demands more effectively. For instance, the buffer tank setpoint calculated by RSC as a result of a reset schedule can be read by the GSHP control board to be understood as its new target temperature. Another improvement was the installation of three thermocouples in each of the return legs of the helical GHEs. These thermocouples were installed on 90-degree elbow fittings near the entrance to the ground, in the mechanical room. Unlike the rest of the main geothermal loop piping, these elbow fittings were made of brass, which has a much larger thermal conductivity than PEXa; therefore, the temperature sensed by the thermocouple would be closer to the fluid temperature inside the pipes at any given time. The temperature sensors were then covered with a layer of asphaltic tape and an additional wrapping with closed-cell foam insulation material to avoid external influence. Three thermocouples were already reading temperatures of the supply legs of each helical GHE, but they had been positioned on regular PEXa pipe; consequently, these sensors were relocated to brass elbow fittings to match conditions with those on the supply legs.

During Cooling Test II, the GSHP was set to a fixed 25% capacity on one compressor. The purpose was to investigate long term thermal behavior applying a fixed load to the heat exchanger, similarly to how it was done in Cooling Test I. But in this

case, instead of controlling the building side load with a PID loop tied to the ΔT on the radiant floor system, the PID loop was tied to the temperature in the buffer tank with a setpoint of 50°F. This way, not only the temperatures on the geothermal field loops would be stable but also those on the building side of the GSHP. This type of control strategy would be interesting for optimizing GSHP energy efficiency; although, this analysis is outside of the scope of this thesis.

Heating Test II

Finally, Heating Test II was carried out during the first stages of 2015's heating season in Bozeman, MT. The purpose of this test was to gain knowledge on how the helical GHE behaves when pulsating loads are applied to the system, in contrast with Cooling Test II, which ran at a fixed compressor capacity. Heating Test II ran for about 29 days. The control strategy was a "real-life" one: GSHP running freely under its own internal capacity modulation logic; a target temperature determined by an outdoor temperature reset schedule on the buffer tank temperature sensor calculated by the BMS; and a building side demanding heat on the radiant floor system as per real needs on the different spaces of the residence. This control strategy resulted in an intermittent operation of the GSHP, with peak loads during the coldest hours of the day, typically the night and early morning, and resting periods during the warmer hours in the midday (see Experimental Results. Heating Test II section). Conditions and parameters during the tests in *phase II* are summarized in Table 7.

Table 7. Phase II (Cooling Test II and Heating Test II) testing conditions

Description	Value (I-P Units)	Value (SI Units)
<u>Geothermal Fluid</u>		
Type of fluid	Propylene glycol 20% vol.	
Average volumetric flow rate in loop (cooling test – heating test)	11.9 – 10.1 gpm	0.75 – 0.64 l/s
Average fluid temperature during test (cooling test – heating test)	94.4 – 60.0°F	34.7 – 15.6°C
<u>Layout and Geometry of Heat Exchanger</u>		
Type of probe	Helical	
Number of probes	3	
Spacing	11 ft	3.4 m
Depth of installation	6 ft	1.8 m
Length of probe	12 ft	3.7 m
Diameter of borehole	18 in	0.46 m
Diameter of helix	15 in	0.38 m
Pitch between Turns	4.3 in	0.11 m
Total length of pipe	131 ft	40 m
Material of pipe	Cross-linked polyethylene (PEXa)	
Inside diameter	0.875 in	22.2 mm
Outside diameter	1.125 in	28.6 mm
Density of material	58.68 lb _m /ft ³	940 kg/m ³
Thermal conductivity of material	0.24 Btu/(hr·ft·°F)	0.41 W/(m·K)
<u>GSHP Capacity Control</u>		
Number and speed of compressors used (cooling test)	1 compressor – fixed 25%	
(heating test)	2 compressors – variable	
<u>Properties of Ground</u>		
Type of soil	85% clay – 15% sand	
Estimated thermal conductivity	0.74 Btu/(hr·ft·°F)	1.28 W/(m·K)
Estimated thermal diffusivity	0.62 ft ² /day	0.667·10 ⁻⁶ m ² /s
Backfilling material	Native soil (no grout)	
<u>Climate Data of Location</u>		
Location	Bozeman, MT	
Annual mean temperature	44.6°F	7°C
Annual variation (semi-amplitude)	21.9°F	12.2°C
Average outdoor temperature during test (cooling test – heating test)	62.9 – 43.6°F	17.2 – 6.4°C

Performance Calculations

The first law of thermodynamics for a stream of fluid states:

$$q + \sum \dot{m}_i \cdot \left(h_i + \frac{v_i^2}{2} + gz_i \right) = \sum \dot{m}_o \cdot \left(h_o + \frac{v_o^2}{2} + gz_o \right) + \dot{W} \quad (1)$$

where

q = rate of heat transfer
 $\dot{m}$ = mass flow rate
 h = enthalpy
 v = velocity
 g = acceleration due to gravity
 z = altitude
 $\dot{W}$ = work

and the subscripts i and o correspond to “in” and “out” of the system respectively.

For steady flow, the kinetic and potential energy terms vanish and $\dot{m}_i = \dot{m}_o = \dot{m}$. In the absence of work, the expression takes the form:

$$q = \dot{m} \cdot (h_o - h_i) \quad (2)$$

For a liquid, the difference in enthalpy is equivalent to:

$$\Delta h = c_p \cdot (T_o - T_i) = c_p \cdot \Delta T \quad (3)$$

where

Δh = enthalpy difference between inlet and outlet of the system
 c_p = specific heat of the fluid
 T = temperature of the fluid
 ΔT = temperature difference between inlet and outlet of the system

Substituting Equation (3) into Equation (2) yields the well-known Equation (4).

$$q = \dot{m} \cdot c_p \cdot \Delta T \quad (4)$$

In turn,

$$\dot{m} = \dot{Q} \cdot \rho \quad (5)$$

where

$$\begin{aligned} \dot{Q} &= \text{volumetric flow rate} \\ \rho &= \text{density} \end{aligned}$$

Equation (4) relates the rate of heat transfer in a system with the specific heat capacity and the change in temperature. Therefore, it can be used as a first approach to calculate the performance of the GHE.

However, this equation is only valid when any increase in energy is due only to sensible heat transfer. In addition, it only holds if c_p is assumed constant. For the range of temperature differences typically occurring in ground heat exchangers, the assumption is valid. As an example, the difference in specific heat of pure water at 30 psi ($2.1 \cdot 10^5$ Pa) between 65 and 75°F (18.3 and 23.9°C) is only 0.4% (IRC, 2015).

The use of this equation within appropriate uncertainty limits is straightforward since direct measurements of the temperatures in and out of the GHE (LWT and EWT, respectively) and the mass flow rate ($\dot{m}$) in the ground loop are available from the DAQ system.

In this thesis, temperature differences (ΔT) in cooling and heating mode are calculated as can be seen in Equation (7). The reason is to obtain positive ΔT 's and rates of heat transfer in both modes for ease of use. However, it has to be noted that the direction of the heat flow is opposite to one another.

$$\begin{aligned} \Delta T_{Cooling} &= T_i - T_o \\ \Delta T_{Heating} &= T_o - T_i \end{aligned} \quad (6)$$

Transient Heat Conduction

Equation (4) is very useful to investigate the instantaneous heat rate that is happening in the GHE at any given time. However, the mechanism of heat transfer between the GHE outer pipe wall and the surrounding medium is primarily heat conduction. In addition, given the great amount of thermal capacitance of the portion of ground affected by the energy transfer process, the time dependency is clear, and the heat transfer is transient in nature, especially in the first stages of a startup operation. The fluid temperature difference between inlet and outlet start small and progressively increases with time. Both inlet and outlet temperature also change with time.

However, given enough time, the temperature gradients within the medium surrounding the GHE will reach a quasi-steady state with the “infinite” far field, and the amount of heat absorbed or taken away from the ground will equate that given away or absorbed by the GSHP on the other side of the loop. At that time, the temperatures in and out of the heat exchanger will be constant as well as its difference.

Real-life systems, though, typically work intermittently, as in any other HVAC system, by trying to maintain a certain setpoint to ensure cooling and heating needs are met. Also, residential and small commercial GSHPs have one, or two at the most, stages of compressor capacity. In order to predict what happens in the ground when intermittent pulses of heat are applied to the GHE, with transient heat transfer in a complex geometry, a more elaborate model is needed. As outlined in the LITERATURE REVIEW chapter, this research will make use of a previously reported CaRM (Capacity-Resistance Model) (Zarrella & De Carli, 2013). The model will be explained in detail in the chapter CAPACITY-RESISTANCE SIMULATION TOOL.

Sources of Error

In the calculation of the performance of the helical GHE in this thesis, many sources of error must be taken into account. In the following list, the most important ones are presented.

- Some runs of the ground source pipe work in the mechanical room are not insulated. As a consequence, there can be heat losses due to temperature difference between geothermal fluid and the mechanical room air temperature. For example, in heating mode, the fluid leaving the GSHP can reach temperatures as low as 29°F (-1.7°C); with uninsulated pipes, heat gains could be up to 2°F (1.1°C) by the time the fluid leaves the mechanical room. In addition, the horizontal runs connecting the helical GHEs with the mechanical room are assumed to be insulated, but still they are long enough so there is a chance of heat losses occurring along them.
- The heat carrier fluid in the geothermal loops flows thanks to a wet rotor type circulating pump. These types of circulators transmit the residual heat produced from the motor operation to the fluid they propel. This heat was not accounted for in the performance calculations.
- Equation (4) is only valid for a steady state situation and depends on the assumption that the thermal properties of the heat carrier fluid at the inlet and outlet of the heat exchanger are equal.

Uncertainty Analysis

An experimental uncertainty analysis was carried out according to the *root of the sum of the squares* (RSS) method (Wheeler & Ganji, 2010). The analysis was outlined and applied to the Cooling Test II as an example, obtaining a 23% uncertainty for the performance calculation described earlier in this section. Details can be found in APPENDIX B.

CAPACITY-RESISTANCE SIMULATION TOOL

The capacity-resistance model (CaRM) used in this thesis was introduced in the LITERATURE REVIEW and is explained in detail in the following references:

- De Carli et al. (2010)
- Zarrella et al. (2011)
- Zarrella and De Carli (2013)

In this section, the most important features of the model and its mathematical foundations are explained. Then how the model was applied to the experimental data from the tests conducted at the RMEH is presented.

Description of the Model

Mathematical Foundations

Derived from the principle of conservation of energy and Fourier's law of heat conduction, the heat equation defines the transient heat transfer through a body as a function of position and time (Haberman, 2004). In its more general form it can be expressed as:

$$\rho c_p \frac{\partial T}{\partial \tau} = \nabla \cdot (k \nabla T) + g \quad (7)$$

where

- c_p = specific heat
- ρ = density
- T = temperature
- τ = time
- k = thermal conductivity
- g = internal heat generation

Equation (7) in Cartesian coordinates and one dimension becomes:

$$\rho c_p \frac{\partial T}{\partial \tau} = \frac{\partial}{\partial x} \left(k \frac{\partial T}{\partial x} \right) + g \quad (8)$$

where x is distance. If the material is uniform and the thermal coefficients (ρ , c , and k) are considered constant, and there is no internal heat generation, Equation (8) takes the form:

$$\frac{\partial T}{\partial \tau} = \alpha \frac{\partial^2 T}{\partial x^2} \quad (9)$$

where the constant α ,

$$\alpha = \frac{k}{\rho c_p} \quad (10)$$

is the thermal diffusivity. Equation (9) is the prototype of parabolic partial differential equation, and it has been extensively studied.

For a stationary system (steady state), where there is no change in temperature with time, the left-hand side of Equation (9) is equal to 0 and the coefficient α vanishes. The heat equation then becomes Laplace's equation.

$$\frac{d^2 T}{dx^2} = 0 \quad (11)$$

1-D, Steady-State Heat Conduction through a Wall. For the particular case of steady-state heat transfer through a plane wall, Equation (11) can be solved to give:

$$T(x) = C_1 x + C_2 \quad (12)$$

where C_1 and C_2 are arbitrary constants. With initial conditions

$$\begin{aligned} T(0) &= T_1 \\ T(L) &= T_2 \end{aligned} \tag{13}$$

with L representing the thickness of the wall. Equation (12) becomes

$$T(x) = T_1 + \frac{T_2 - T_1}{L}x \tag{14}$$

Equation (14) is a straight line between temperature T_1 and T_2 across the length L .

The rate of heat transfer through the wall can be evaluated by means of Fourier's law:

$$q = -kA \frac{dT}{dx} \tag{15}$$

where A is area of the wall. Substituting Equation (14) into Equation (15) gives:

$$q = -kA \frac{d}{dx} \left(T_1 + \frac{T_2 - T_1}{L}x \right) \tag{16}$$

and performing the differentiation yields:

$$q = -kA \frac{T_2 - T_1}{L} = kA \frac{T_1 - T_2}{L} \tag{17}$$

Rearranging Equation (17):

$$q = \frac{T_1 - T_2}{L/kA} = \frac{T_1 - T_2}{R} \tag{18}$$

where R is the thermal resistance for 1-D steady state heat transfer through a plane wall.

$$R = \frac{L}{kA} \tag{19}$$

Following a similar derivation for the axisymmetric steady-state heat transfer through a cylindrical wall, the rate of heat transfer is:

$$q = 2\pi Lk \frac{T_i - T_o}{\ln(r_o/r_i)} \quad (20)$$

where T_i and T_o are the inner and outer surface temperatures of the cylindrical wall and r_i and r_o are the inner and outer radius of the cylinder, respectively. The value of the thermal resistance takes the form:

$$R = \frac{\ln(r_o/r_i)}{2\pi Lk} \quad (21)$$

These thermal resistances can be calculated with Equations (19) and (21) even if the heat transfer is not in equilibrium, but the 1-D assumption must be kept.

Heat Capacity of a Body. In a real system, when heat is transferred through a body, part of that heat is stored according to its heat capacity. In the absence of a change of state, this increase in internal energy can be approximated to its rate of change in temperature as follows:

$$q = \rho c_p V \frac{dT}{d\tau} \quad (22)$$

where V is the volume of the body. Equation (22) assumes no temperature differences within the body; in other words, the body is considered a lumped system. This is key in the development of the CaRM model, since the whole system is discretized in a series of “nodes” which adequately describe the geometry of the ground heat exchanger. Each node holds the mass and thermal properties of the control volume it occupies in space and is generally located in its barycenter.

The Electrical Analogy. Given the analogy between Fourier's law and Ohm's law, it becomes handy to represent the system as a network of thermal nodes linked with thermal capacitances and thermal resistances. It is especially suitable for a discretized, lumped definition of the domain as in the CaRM model. For informational purposes, Table 8 shows a summary of the analogies between thermal and electrical phenomena.

Table 8. Analogy between heat transfer and electrical phenomena. Adapted from Kasap (2003)

Thermal Phenomena	Electrical Phenomena
Rate of Heat Transfer (q)	Current (I)
Temperature (T)	Voltage (V)
Thermal Resistance (R)	Electrical Resistance (R)
Thermal Capacitance (C)	Electrical Capacitance (C)
Heat Reservoir	Electromotive Force
Absolute Zero	Ground
Heat Generator	Current Supply

Finite Difference Approach. To numerically solve the differential equation, a finite difference method approach is used. A heat balance is applied to each node as follows:

$$\text{Heat Flow In} - \text{Heat Flow Out} = \text{Rate of Heat Accumulation} \quad (23)$$

Equation (23) is Equation (22) put in words. In terms of a generic i -th node (Figure 23), the heat balance is:

$$q(i-1, i) + q(i, i+1) = \rho(i) \cdot c_p(i) \cdot V(i) \cdot \frac{dT(i)}{d\tau} \quad (24)$$

The values obtained from Equation (18) are substituted into the left hand side of Equation (24) and the time derivative is approximated by a backward difference numerical differentiation method:

$$\frac{T(i-1) - T(i)}{R(i-1, i)} + \frac{T(i+1) - T(i)}{R(i+1, i)} = \rho(i) \cdot c_p(i) \cdot V(i) \cdot \frac{T(i) - T(i)_{-\Delta\tau}}{\Delta\tau} \quad (25)$$

where R is the resistance between nodes and $T(i)_{-\Delta\tau}$ represents the temperature of the i -th node at previous time step.

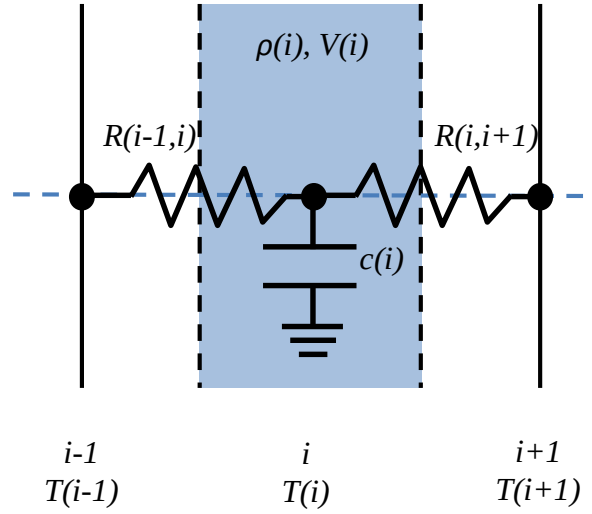


Figure 23. Example of control volume for CaRM node energy balance. Adapted from De Carli et al. (2010)

Error Analysis. This numerical approximation introduces some truncation error due to neglecting higher order terms of the polynomial Taylor expansion (Chapra, 2012). In this case, for the backwards difference approximation of the derivative, taking only the first term, the truncation error has the order of magnitude of the selected time step.

$$\text{Truncation Error} = O(\Delta\tau) \quad (26)$$

Ground Modeling

To model the ground, three zones are considered:

- The Borehole Zone

- The Surface Zone
- The Deep Zone

The general approach of the model is shown in Figure 24.

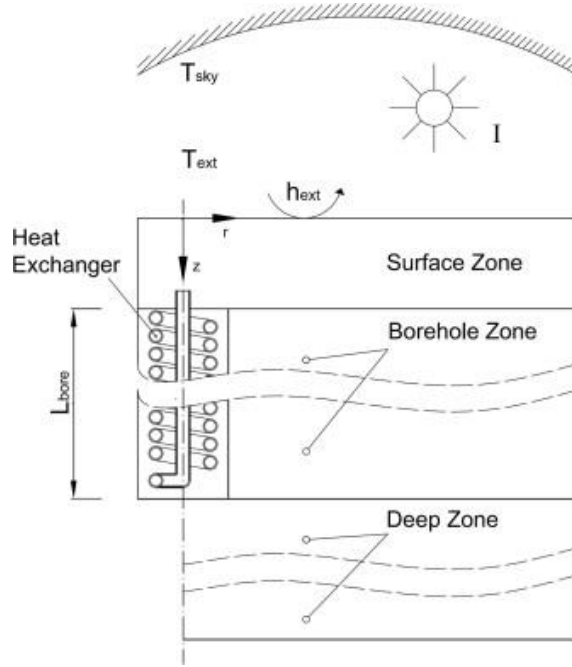


Figure 24. General approach of the CaRM-He model (Zarrella and De Carli 2013)

The borehole zone is discretized in m slices and n annular regions (Figure 25). In this area, heat conduction in the axial and in the radial direction is considered. Cylindrical symmetry is assumed. Then the energy balance for the j -th slice and the i -th annular region is:

$$\begin{aligned}
 & \frac{T(j, i-1) - T(j, i)}{R_r(j, i-1)} + \frac{T(j, i+1) - T(j, i)}{R_r(j, i)} \\
 & + \frac{T(j-1, i) - T(j, i)}{0.5 \cdot R_z(j-1, i) + 0.5 \cdot R_z(j, i)}
 \end{aligned} \tag{27}$$

$$\begin{aligned}
& + \frac{T(j+1, i) - T(j, i)}{0.5 \cdot R_z(j+1, i) + 0.5 \cdot R_z(j, i)} \\
& = \frac{C(j, i) \cdot T(j, i) - T(j, i)_{-\Delta\tau}}{\Delta\tau}
\end{aligned}$$

with:

$$\begin{aligned}
R_r(j, i) &= \frac{\ln\left(\frac{r_m(i)}{r_m(i-1)}\right)}{2\pi \cdot \Delta z_b(j) \cdot k(j)} \\
R_z(j, i) &= \frac{\Delta z_b(j)}{k(j) \cdot \pi \cdot [r^2(i) - r^2(i-1)]}
\end{aligned} \tag{28}$$

where R_r and R_z are thermal resistances between two adjacent annular regions and slices in the radial and vertical direction respectively, and C is the thermal capacitance of the region. The farthest nodes in the radial direction are coupled with the far field (undisturbed) temperature at that point in space, which in turn depends on depth and time, as will be explained later.

The surface and the deep zones are modeled taking into account only heat conduction in the vertical direction. In addition, in the ground surface, the effect of the environment is modeled attending to the effect of exterior air temperature, solar radiation and radiative exchange with the sky. In this case, the zones are discretized only in slices, with lumped properties in their barycenter (Figure 26 and Figure 27). In the last node of the deep zone, the boundary condition is only the undisturbed ground temperature. The energy balance for the surface zone is then:

$$\frac{T(j_s - 1) - T(j_s)}{0.5 \cdot R_z^s(j_s - 1) + 0.5 \cdot R_z^s(j_s)} + \frac{T(j_s + 1) - T(j_s)}{0.5 \cdot R_z^s(j_s + 1) + 0.5 \cdot R_z^s(j_s)} \tag{29}$$

$$= \frac{C(j_s) - T(j_s) - \Delta\tau}{\Delta\tau}$$

with:

$$R_z^s(j_s) = \frac{\Delta z_s(j_s)}{k(j_s) \cdot \pi \cdot r^2(n)} \quad (30)$$

$$C(j_s) = \rho(j_s) \cdot c_p(j_s) \cdot \Delta z_s(j_s) \cdot \pi \cdot r^2(n)$$

where R_z^s represents the thermal resistance in the axial direction between two adjacent slices and $C(j_s)$ is the heat capacity of the slice considered. The energy balance for the deep zone is derived in a similar fashion.

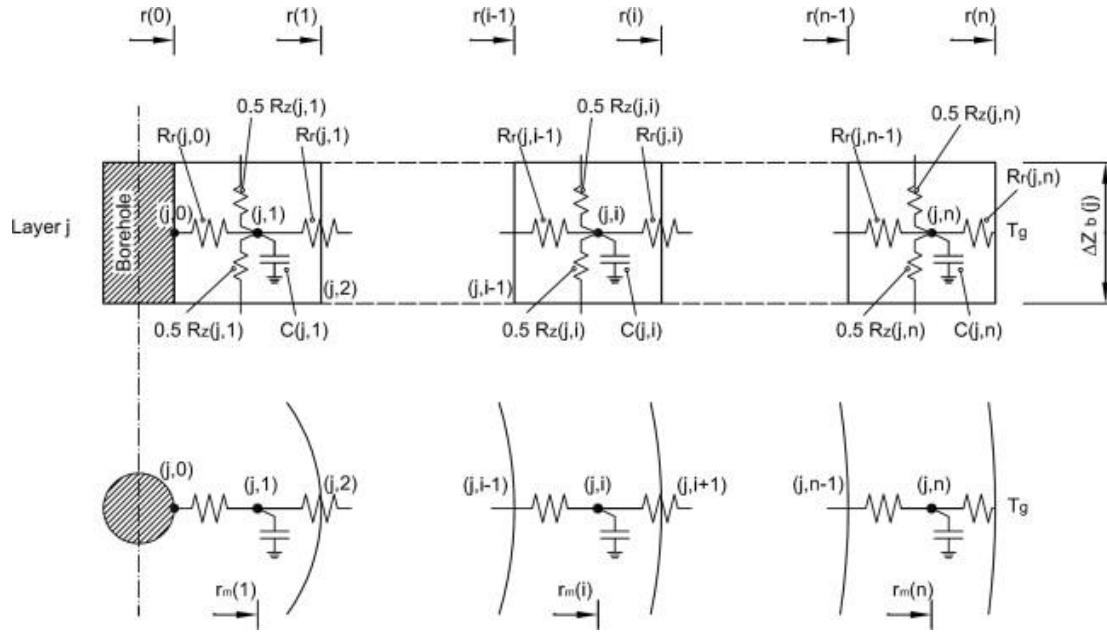


Figure 25. Borehole zone modeling in CaRM-He model (Zarrella and De Carli 2013)

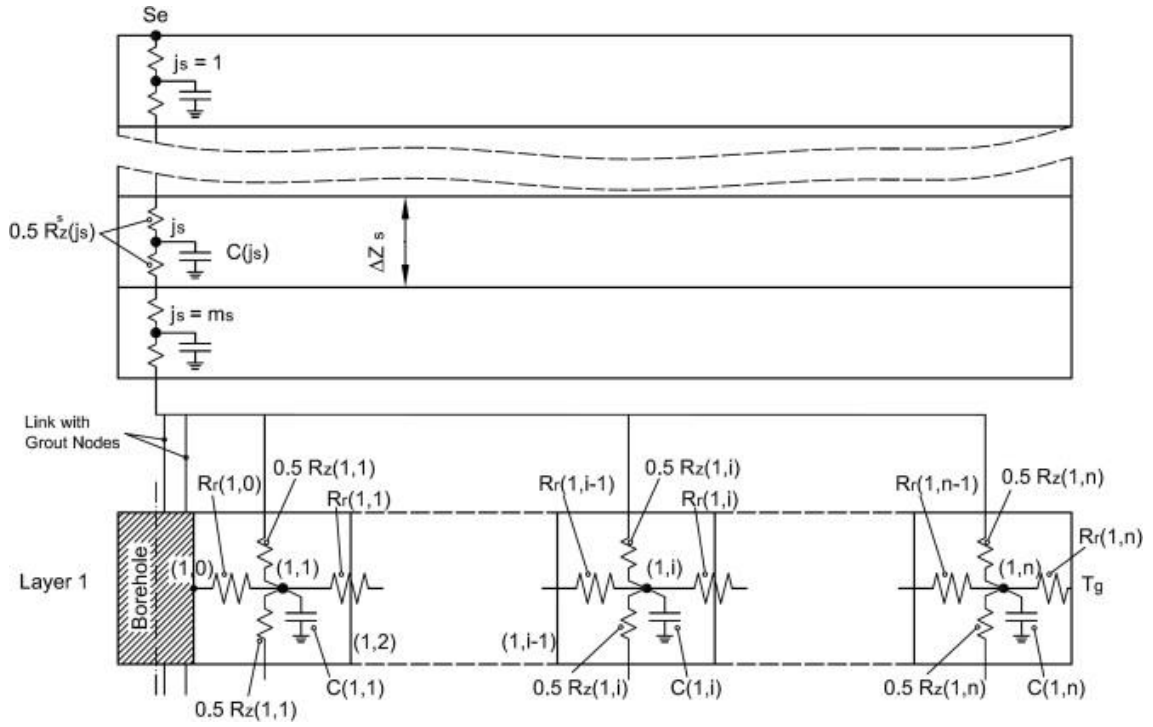


Figure 26. Surface zone modeling in CaRM-He model (Zarrella and De Carli 2013)

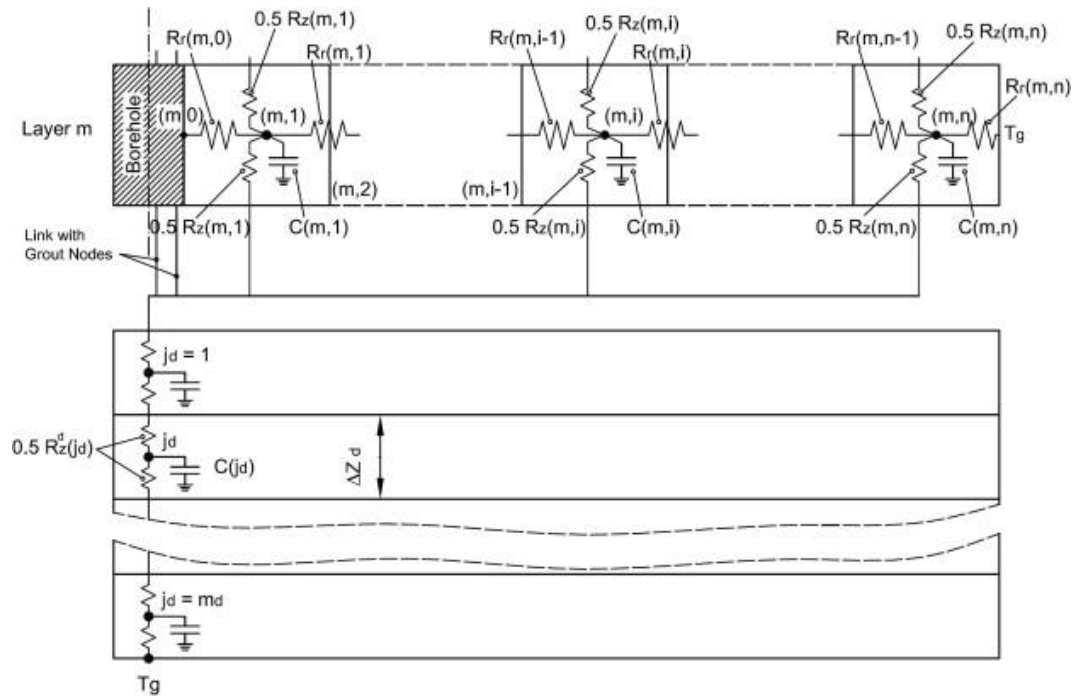


Figure 27. Deep zone modeling in CaRM-He model (Zarrella and De Carli 2013)

As mentioned before, the energy balance at the ground surface node is modeled as follows:

$$q_{conv} + q_{cond} + q_{solar} + q_{sky} = 0 \quad (31)$$

by means of

$$\begin{aligned} q_{conv} &= \pi \cdot r^2(n) \cdot h_{ext} \cdot (T_{ext} - T_{Se}) \\ q_{cond} &= \frac{T(1) - T_{Se}}{0.5 \cdot R_z^s(1)} \\ q_{solar} &= a \cdot I \cdot \pi \cdot r^2(n) \end{aligned} \quad (32)$$

$$q_{sky} = \varepsilon \cdot \sigma \cdot (T_{Se}^4 - T_{sky}^4) \cdot \pi \cdot r^2(n)$$

where q_{conv} is the heat exchange with the ambient due to convection, with h_{ext} being the convection heat transfer coefficient; q_{cond} is the 1-D vertical heat conduction with the ground surface node towards the interior of the ground; q_{solar} is the incident solar radiation (short-wave radiation); and q_{sky} is the heat exchange with the sky (long-wave radiation). The forth-power terms in the radiative exchange with the sky can be linearized with small error due to the small temperature difference involved in the process. The temperature of the sky is estimated by the Swinbank's correlation (Swinbank, 1963). See Zarrella and De Carli (2013) for further details.

The initial temperature distribution as a function of time of the year and depth is set via the Kusuda & Achenbach model (Kusuda & Achenbach, 1965) according to Equation (33). The accuracy of Equation (33) was investigated and the results are presented in APPENDIX D.

$$T(z, \tau) = T_m - A_T \cdot \exp\left(-z \cdot \sqrt{\frac{\pi}{\tau_y \cdot \alpha}}\right) \cdot \cos\left[\frac{2\pi}{\tau_y} \cdot \left(\tau - \tau_{shift} - \frac{z}{2} \cdot \sqrt{\frac{\tau_y}{\pi \cdot \alpha}}\right)\right] \quad (33)$$

where

- z = depth (ft)
- τ = time (days)
- τ_y = time span of a year (365 days)
- τ_{shift} = day of minimum surface temperature (days)
- T_m = annual average air temperature (°F)
- A_T = annual semi-amplitude of the monthly average air temperature (°F)
- α = ground thermal diffusivity (ft²/day)

Borehole Modeling

The borehole is the area of the ground that has been excavated and contains the helical GHE. Boreholes for helical GHEs are typically wider than those drilled for deep vertical U-tube probes. They can thus be backfilled with native soil instead of with a specialized thermal grout; although, the latter is still an option. Borehole modeling in the CaRM-He model is done in a similar way as the borehole zone of the ground. It is assumed that only axisymmetric heat conduction occurs within this area. Heat transfer in both axial and radial directions is considered.

The geometry of the borehole is discretized and approximated by splitting it into two zones: the core, between the internal straight radial pipe and the helical pipe; and the shell, between the helical pipe and the external wall of the borehole where it connects to the rest of the surrounded soil formation. According to this node network (Figure 28), a

set of energy balance equations can be set up. See reference (Zarrella and De Carli 2013) for detailed description.

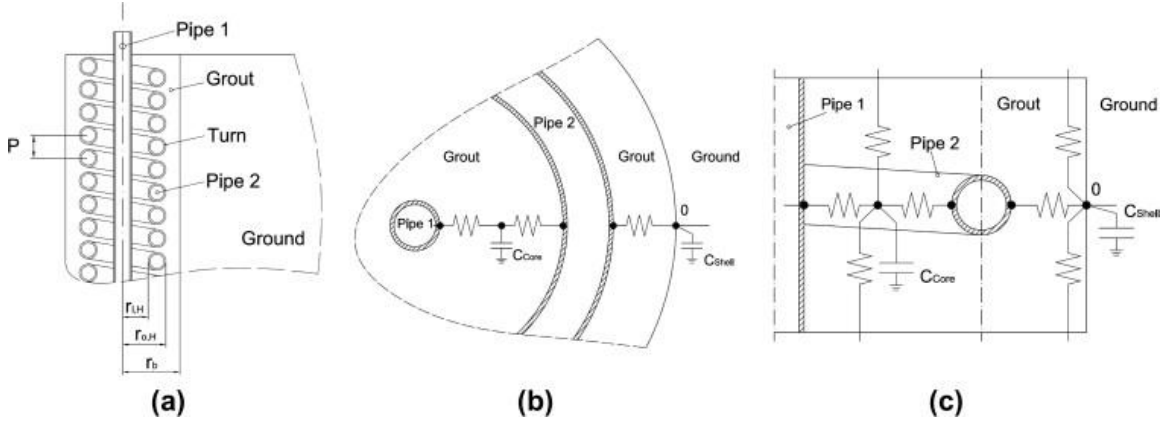


Figure 28. Borehole zone modeling in CaRM-He model (Zarrella and De Carli 2013). (a) General view. (b) Horizontal section. (c) Vertical section

Fluid Modeling

The last part of the geometry that needs to be modeled is the heat carrier fluid and how it interacts with the heat exchanger pipe wall. In this case, the heat transfer occurs not only by conduction but also by forced convection, and it is influenced by the flow regime in the pipe. The convective heat transfer coefficient can be approximated by the definition of the Nusselt number (Nu) and empirical correlations that depend on the flow regime. The Nusselt number is the ratio of the convective heat transfer coefficient to the conductive heat transfer coefficient (Equation (34)).

$$Nu = \frac{hL}{k} \quad (34)$$

where h is the convective heat transfer coefficient, L is the characteristic length and k is the thermal conductivity of the fluid. The empirical correlations used in the

CaRM model for each type of flow (laminar, transition and turbulent) depend on the Reynolds (Re) and Prandtl (Pr) numbers and the geometry of the pipe (Equation (35)).

$$\begin{aligned}
 Nu &= 1.61 \cdot Re \cdot \left(Pr \cdot \frac{D}{L} \right)^{\frac{1}{3}} \quad (Re < 2000) \\
 Nu &= 0.116 \cdot \left(Re^{\frac{3}{5}} - 125 \right) \cdot Pr^{\frac{1}{3}} \cdot \left[1 + \left(\frac{D}{L} \right)^{\frac{2}{3}} \right] \\
 &\quad (2000 < Re < 10000) \\
 Nu &= 0.023 \cdot Re^{0.8} \cdot Pr^{\frac{1}{3}} \quad (Re > 10000)
 \end{aligned} \tag{35}$$

The discretization of the system at this level takes into account the geometry of the helical heat exchanger (length, inner and outer diameter of the pipe, diameter of the helix and pitch between turns) at each slice of the borehole in the vertical (axial) direction. The energy balance also considers the conductive thermal resistance and the convective heat transfer coefficient, as well as the heat capacity, of the fluid and the filling or grouting material.

Influence of Adjacent Boreholes

It is common that a number of GHEs are needed for a specific geothermal energy HVAC application. If that is the case, a bore field must be implemented. Since availability of ground surface around the buildings can be short, it is possible that there is some interaction between adjacent GHE within the bore field depending on the spacing. To account for this interaction, the model makes use of an adiabatic (heat flux = 0) boundary condition located half way between the interacting GHEs (De Carli, Tonon et

al. 2010). Devising various types of possible configurations, any multi-borehole field can be modeled.

Running the Model

In the end, all the thermal nodes and heat balances considered in the model can be arranged in a linear system of equations that must be solved simultaneously. In matrix notation, the system of equations can be expressed:

$$[A]\{x\} = \{b\} \quad (36)$$

where $[A]$ is the matrix of coefficients, $\{b\}$ is a column vector of known constants, and $\{x\}$ is the column vector of unknowns, in this case, the temperatures of the nodes at any given time.

The output of the model is, as mentioned above, the temperature of each node at each time step of the simulation. This includes knowledge of the temperature of the fluid at the outlet of the heat exchanger. This information, together with the inlet temperature and mass flow rate, which were inputs of the model, facilitates the calculation of the net power exchanged at a particular time in the GHE.

Input Data from in-situ Experiments

The CarM-He model inputs are:

- Thermal Fluid: total mass flow rate, specific heat, thermal conductivity, kinematic viscosity, density

- Layout of the GHEs: type of probe, number of probes, spacing, depth of installation, length of probe, diameter of the borehole, type of ground surface
- Geometry and properties of the GHE: diameter of the helix, pitch between turns, total length of pipe, material of pipe, inside and outside diameter of pipe, density of material, thermal conductivity of the material
- Climate data of the location: air temperature, solar radiation, annual mean temperature, annual variation (semi-amplitude)
- Thermal properties of the ground and filling material: thermal conductivity, thermal capacity, density
- Operation mode: time span of the simulation, fluid supply (inlet to the GHE) temperature for each time step (alternatively, the heat load can be used as an input instead of the inlet temperature), mode of operation (on/off), type of connection of GHE (parallel, series).
- Mesh: time step of simulation and meshing parameters (number of slices/annular regions)

Input parameters from the experiments at the RMEH are summarized in Table 9.

Table 9. Input parameters from the in-situ tests performed at the RMEH

Description	Value (I-P Units)	Value (SI Units)
<u>Geothermal Fluid</u>		
Total mass flow rate in loop	lb _m /hr	kg/s
(Cooling Test I – Heating Test I)	4603 – 5000	0.58 – 0.63
(Cooling Test II – Heating Test II)	6032 – 5158	0.76 – 0.65
Specific heat	Btu/(lb _m ·°F)	J/(kg·K)
(Cooling Test I – Heating Test I)	0.953 – 0.951	3991.6 – 3979.7
(Cooling Test II – Heating Test II)	0.955 – 0.952	3998.2 – 3985.2
Thermal conductivity	Btu/(hr·ft·°F)	W/(m·K)
(Cooling Test I – Heating Test I)	0.30 – 0.287	0.518 – 0.497
(Cooling Test II – Heating Test II)	0.31 – 0.29	0.528 – 0.507
Kinematic viscosity	ft ² /s	m ² /s
(Cooling Test I – Heating Test I)	18.5·10 ⁻⁶ – 36.1·10 ⁻⁶	1.72·10 ⁻⁶ – 3.26·10 ⁻⁶
(Cooling Test II – Heating Test II)	14.3·10 ⁻⁶ – 25.2·10 ⁻⁶	1.33·10 ⁻⁶ – 2.34·10 ⁻⁶
Density	lb _m /ft ³	kg/m ³
(Cooling Test I – Heating Test I)	63.14 – 63.63	1011.5 – 1019.3
(Cooling Test II – Heating Test II)	62.85 – 63.42	1006.7 – 1015.9
<u>Layout and Geometry of Heat Exchanger</u>		
Type & Number of Probes	Helical – 3	
Spacing	11 ft	3.4 m
Depth of installation	6 ft	1.8 m
Length of probe	12 ft	3.7 m
Diameter of borehole	18 in	0.46 m
Diameter of helix	15 in	0.38 m
Pitch between turns	4.3 in	0.11 m
Total Length of pipe	131 ft	40 m
Material of pipe	PEXa	
Inside/outside diameter	0.875 / 1.125 in	22.2 / 28.6 mm
Density of material	58.68 lb _m /ft ³	940 kg/m ³
Thermal conductivity of material	0.24 Btu/(hr·ft·°F)	0.41 W/(m·K)
<u>Thermal Properties of the Ground</u>		
Estimated thermal conductivity	0.74 Btu/(hr·ft·°F)	1.28 W/(m·K)
Estimated specific heat capacity	0.261 Btu/(lb _m ·°F)	1090.9 J/(kg·K)
Estimated density	110 lb _m /ft ³	1760 kg/m ³
Backfilling material	Native soil (no grout)	
<u>Climate Data of Location</u>		
Location	Bozeman, MT	
Air temperature	From on-site data acquisition system	
Solar radiation	From EnergyPlus® database	
Annual mean temperature	44.6°F	7°C
Annual variation (semi-amplitude)	21.9°F	12.2°C

RESULTS

In this section, the results of the experiments are shown, followed by a performance analysis, the results of the model simulations, and finally the model is compared with experimental results.

Experimental Results

Cooling Test I

Cooling Test I ran for a period of four days in August 2014. Entering and leaving fluid temperatures (EWT and LWT) were measured at 30 second intervals. The measured temperature of the fluid coming into the GSHP (EWT) at the beginning of the test was 60.7°F (15.9°C). This can be an assumed estimation of the undisturbed temperature at that moment. At the end of the test, the same temperature had risen to 79.6°F (26.4°C).

EWT and LWT measurements were plotted together with the outdoor temperature against time (Figure 29). It can be observed that the temperature of the geothermal fluid increases rapidly during the first stages and more slowly as time passes. Of the roughly 18°F (10.6°C) net temperature difference, the temperature rose 68% (12.4°F [6.9°C]) of the total in the first day; whereas during the fourth day, only 6% (1.1°F [0.6°C]) was left to rise (Figure 30). This provides a gauge of how fast the system moves towards steady state.

Another interesting observation is the fact that the temperature difference (ΔT) between EWT and LWT was steadily decaying even towards the end of the experiment

(Figure 31). This indicates that the system did not reach steady state after four days under the conditions of the experiment.

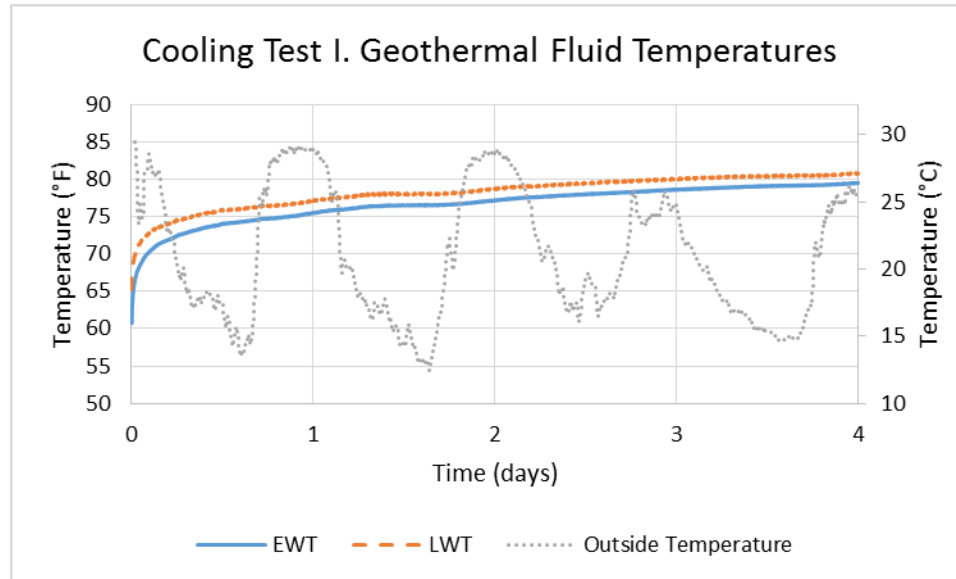


Figure 29. Cooling Test I

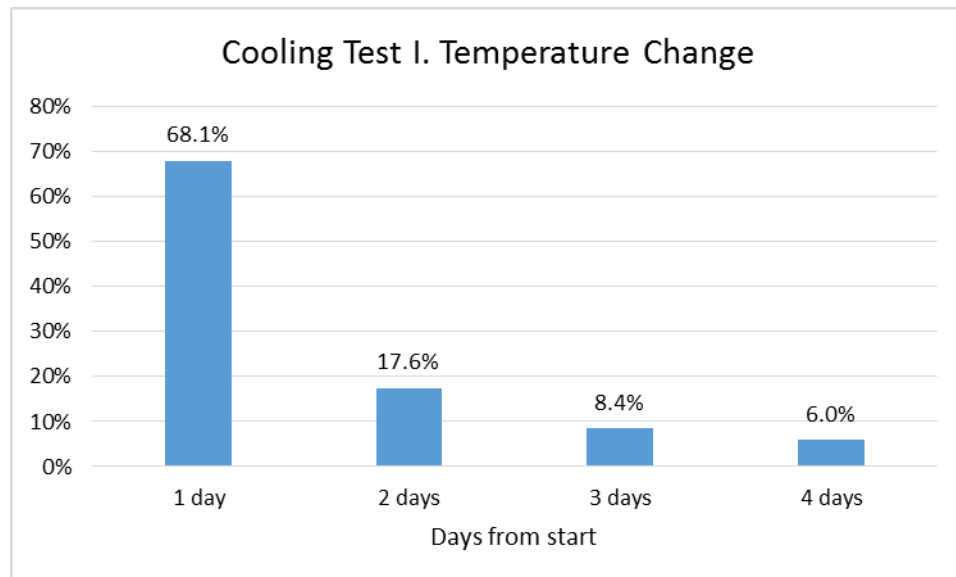


Figure 30. Study of the daily geothermal fluid temperature change with respect to the total net change during Cooling Test I

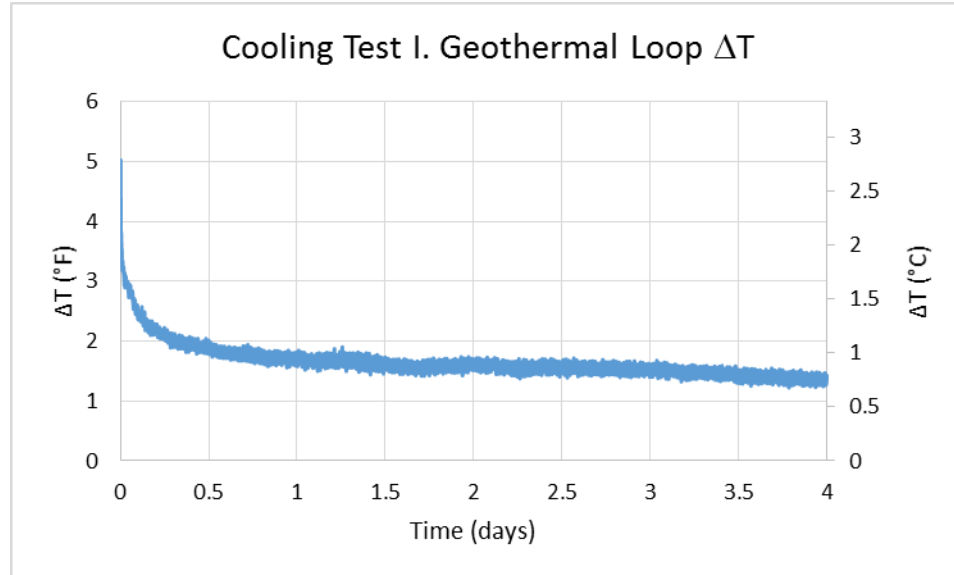


Figure 31. Analysis of temperature difference between inlet and outlet of the GHE (LWT – EWT) during Cooling Test I

The other parameter of interest for calculating the performance of the GHE is the flow rate of the fluid. Flow rate was also measured at 30 second intervals for the duration of the experiment. A plot of the flow rate against time is shown in Figure 32. The circulating pump was set to constant flow. The average flow rate was 8.97 gpm (0.57 l/s) with a standard deviation of 0.31 gpm (0.02 l/s).

The performance of the GHEs was calculated for each time step and the results are shown in Figure 33. It can be observed that the performance is very similar in shape to the ΔT analysis, since flow rate was constant and the rest of terms of the equation are assumed constant as well. In order to estimate a representative performance of the GHEs in this conditions, the average ΔT of 2.4 hours on the fourth day of the experiment was used ($\Delta T = 1.36^{\circ}\text{F}$ [0.76°C]). The average flow rate was 8.98 gpm (0.57 l/s) for the same period of time.

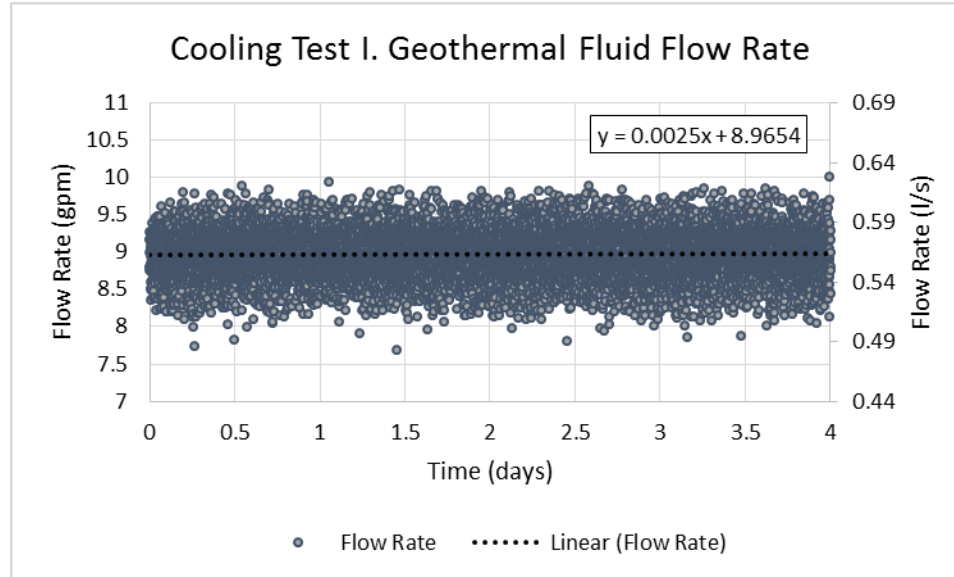


Figure 32. Volumetric flow rate of the geothermal fluid during Cooling Test I. The graphic shows a linear fit to the data with its equation

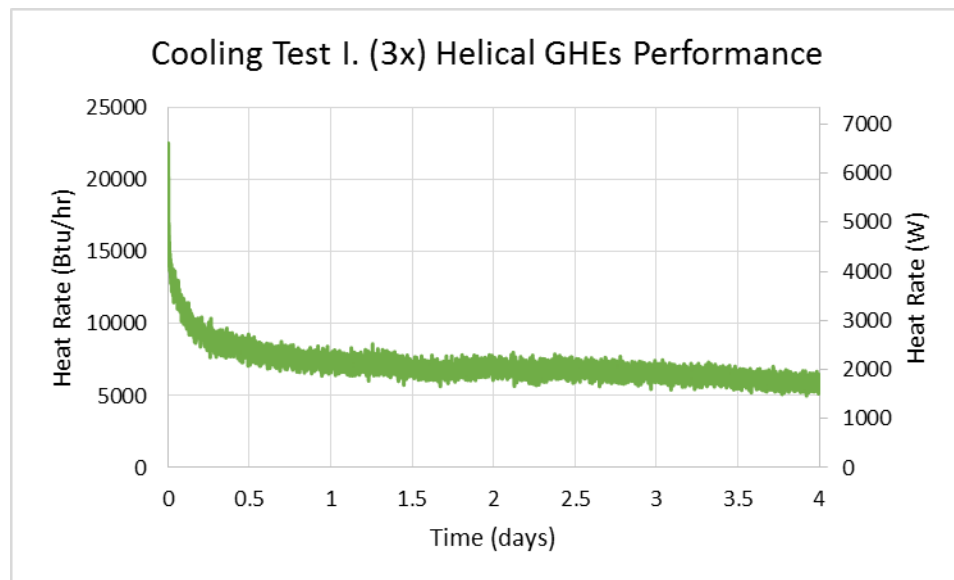


Figure 33. Performance analysis of the three helical GHEs at the RMEH during Cooling Test I

According to Equations (4) and (5), the performance of the GHE is:

$$q = \dot{m} \cdot c_p \cdot \Delta t$$

with

$$\dot{m} = 8.98 \frac{\text{gal}}{\text{min}} \cdot 0.1337 \frac{\text{ft}^3}{\text{gal}} \cdot 63.14 \frac{\text{lb}_m}{\text{ft}^3} \cdot 60 \frac{\text{min}}{\text{hr}} = 4548.5 \frac{\text{lb}_m}{\text{hr}}$$

Then

$$q = 4548.5 \frac{\text{lb}_m}{\text{hr}} \cdot 0.953 \frac{\text{Btu}}{\text{lb}_m \cdot ^\circ\text{F}} \cdot 1.36^\circ\text{F} = 5895.2 \frac{\text{Btu}}{\text{hr}}$$

This is the performance of the three helical GHEs in place, so the unitary representative performance for these conditions is:

$$\frac{5895.2 \frac{\text{Btu}}{\text{hr}}}{3 \text{ helix}} = 1965 \text{ Btu/hr} \cdot \text{helix} \left(576 \text{ W/helix} \right)$$

Cooling Test II

Cooling Test II spanned 42 days (6 weeks) during August and September 2015. Entering and leaving fluid temperatures (EWT and LWT) were measured every 5 minutes. The measured fluid temperature entering the GSHP (EWT) at the beginning of the test was 65.1°F (18.4°C). This is an estimation of the undisturbed temperature at that moment. At the end of the test, EWT had raised up to 96.4°F (35.8°C).

EWT and LWT measurements were plotted against time (Figure 34). A zoom of the first three days can be seen in Figure 35. The net temperature difference between the first and last day was approximately 31°F (17.2°C). EWT raised 75.5% (23.4°F [13°C]) of the total in the first week; whereas during the sixth week, EWT only increased 0.6% (0.2°F [0.1°C]) (Figure 36). It can be affirmed that the system reached near steady state at the end of the experiment.

One interesting observation revealed by such a long experiment is that there occurred subtle daily swings within the general EWT/LWT trend. This may be caused by the influence of the daily temperature change affecting the GHE, the trenched horizontal piping connecting the GHEs with the manifolds in the mechanical room or the heat losses from the uninsulated pipes in the mechanical room.

Temperature difference (ΔT) between EWT and LWT was virtually constant for the last 3 weeks, only changing in the range of one tenth of degree F (Figure 37). This confirms that the system did reach a quasi-steady state situation after 42 days under the conditions of the experiment. The average ΔT of the last week was equal to 0.89°F (0.5°C) with a standard deviation of 0.1°F (0.06°C).

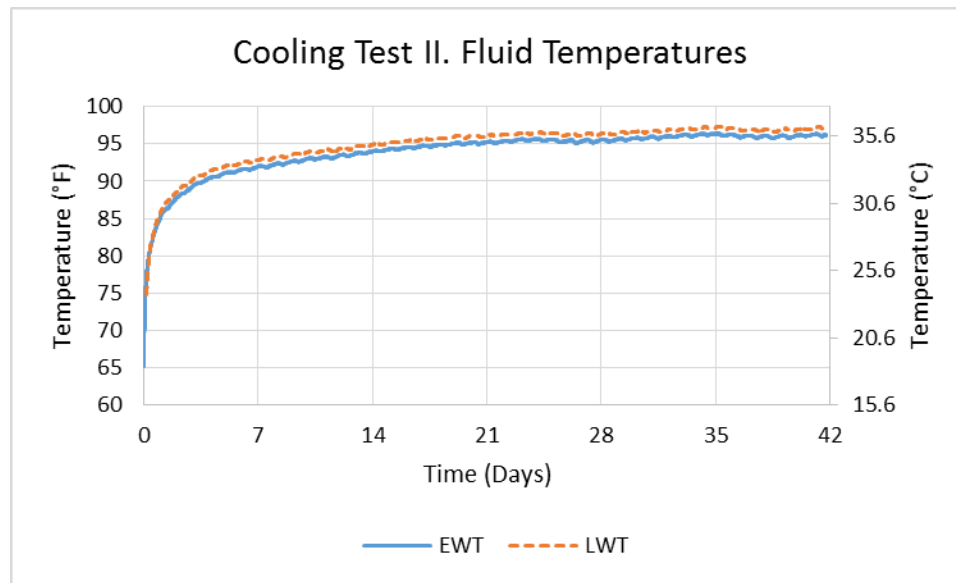


Figure 34. Cooling Test II

Volumetric flow rate was measured at 30 second intervals for the duration of the experiment. A plot of flow rate against time is shown in Figure 38. The circulating pump

was set to constant flow. Data showed that the flow rate was trending slightly upwards possibly due to fluid viscosity decreasing with temperature. Flow rate was averaged for the last week, resulting in 11.99 gpm (0.76 l/s) with a standard deviation of 0.22 gpm (0.014 l/s).

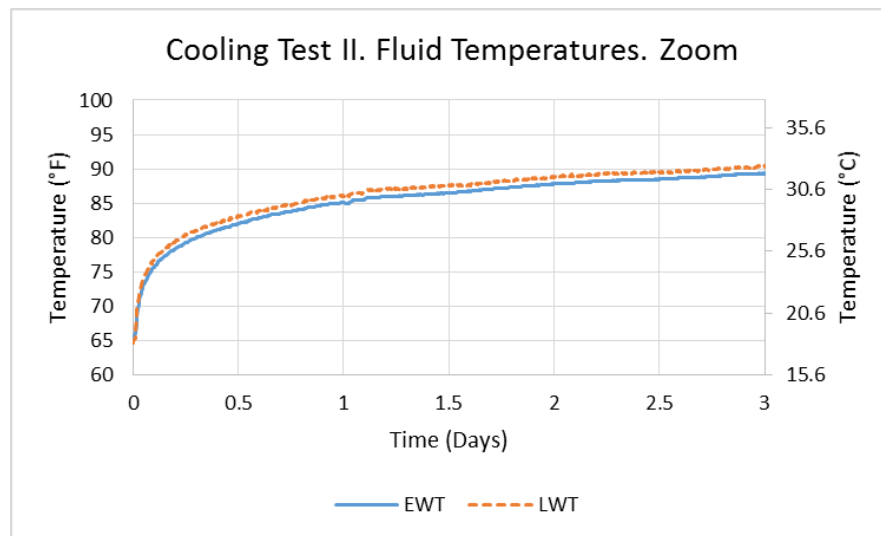


Figure 35. Zoom of first three days of Cooling Test II

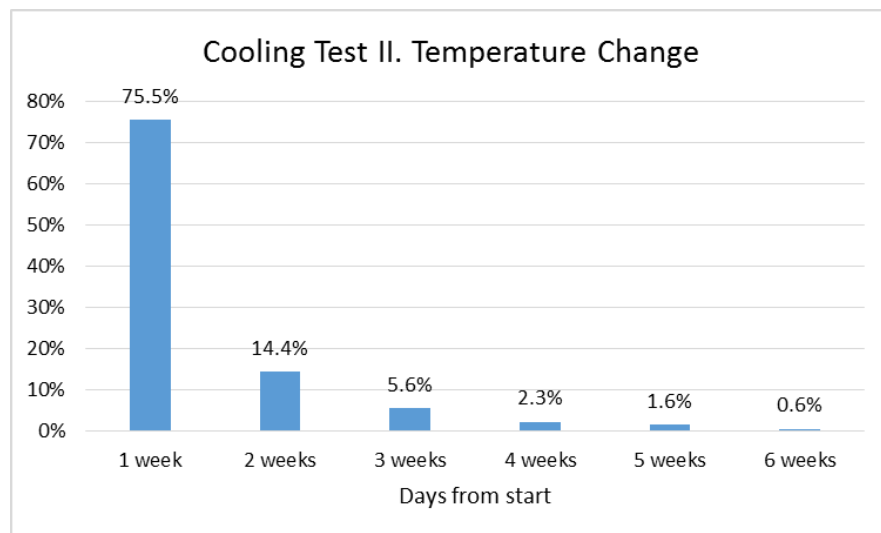


Figure 36. Study of the daily geothermal fluid temperature change with respect to the total net change during Cooling Test I

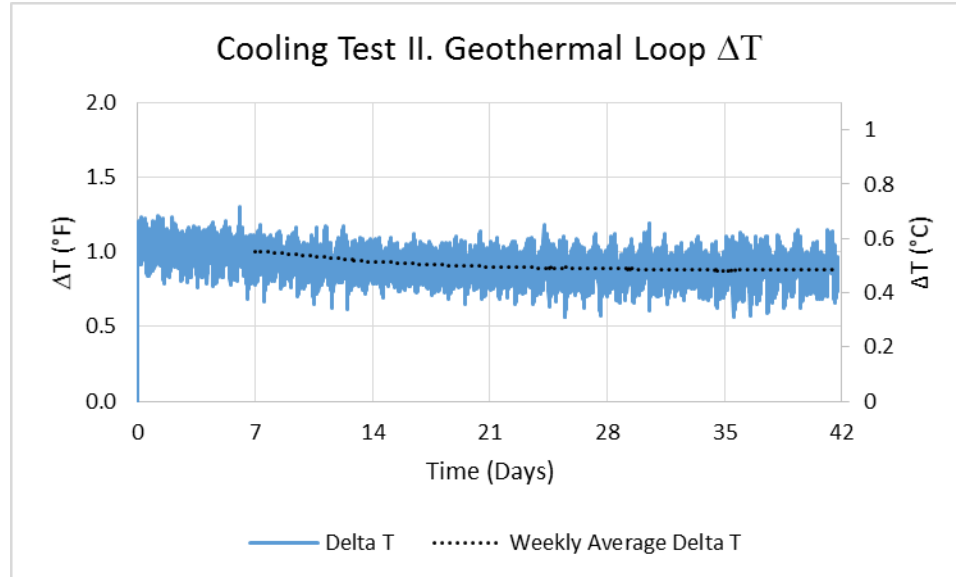


Figure 37. Analysis of temperature difference between inlet and outlet of the GHE (LWT – EWT) during Cooling Test II

Performance was calculated according to Equations (4) and (5). A plot of performance versus time is shown in Figure 39.

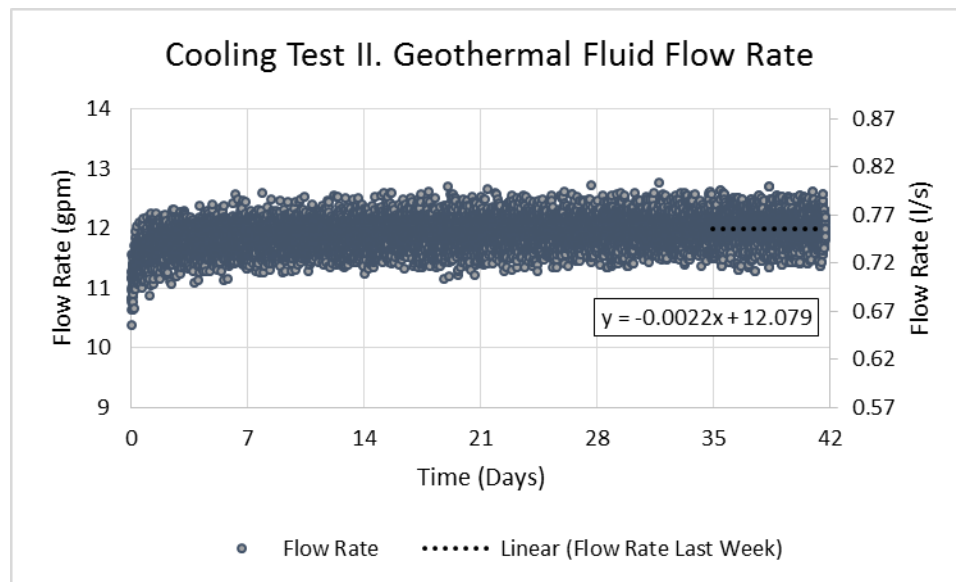


Figure 38. Volumetric flow rate of the geothermal fluid during Cooling Test II. The graphic shows a linear fit to the data for the last week with its equation

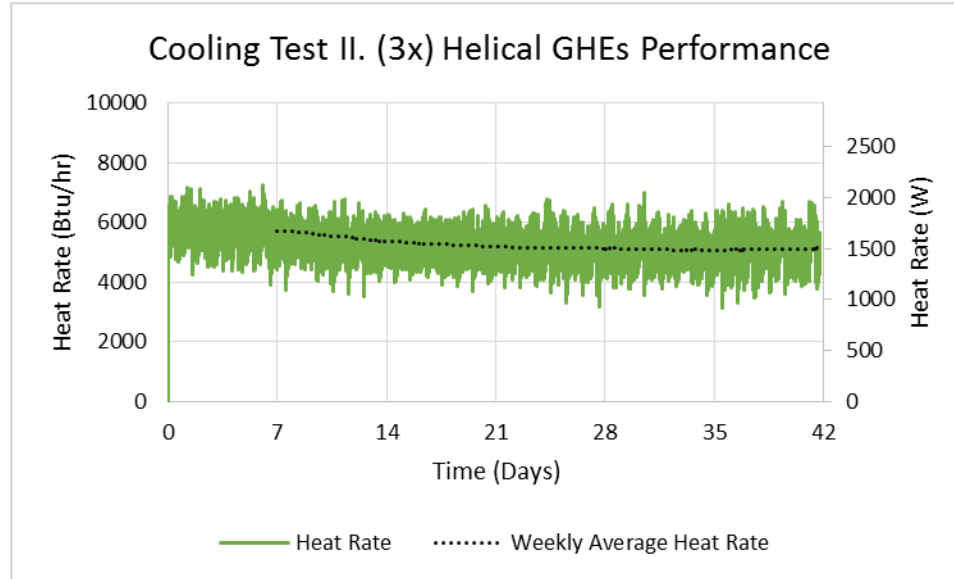


Figure 39. Performance analysis of the three helical GHEs at the RMEH during Cooling Test II

A representative performance calculation for Cooling Test II is:

$$\dot{m} = 11.99 \frac{\text{gal}}{\text{min}} \cdot 0.1337 \frac{\text{ft}^3}{\text{gal}} \cdot 62.85 \frac{\text{lb}_m}{\text{ft}^3} \cdot 60 \frac{\text{min}}{\text{hr}} = 6045.2 \frac{\text{lb}_m}{\text{hr}}$$

Then

$$q = 6045.2 \frac{\text{lb}_m}{\text{hr}} \cdot 0.955 \frac{\text{Btu}}{\text{lb}_m \cdot ^\circ\text{F}} \cdot 0.89^\circ\text{F} = 5138.1 \frac{\text{Btu}}{\text{hr}}$$

This corresponds to the performance of the three helical GHEs in place, so the unitary representative performance for these conditions is:

$$\frac{5138.1 \frac{\text{Btu}}{\text{hr}}}{3 \text{ helix}} = 1713 \text{ Btu/hr} \cdot \text{helix} \left(502 \text{ W/helix} \right)$$

Heating Test I

Heating Test I ran for a period of five days during February 2015. Entering and leaving fluid temperatures (EWT and LWT) were measured at 30 second intervals. The average temperature of the ground at the beginning of the test was 57.3°F (14.1°C). At the end of the test, the temperature decreased to 36.8°F (2.7°C).

EWT, LWT and outdoor temperature measurements were plotted against time (Figure 40). The first part of the experiment, in which the GSHP was cycling on and off can be clearly seen (Figure 41). After that, a steady load was applied in order to reach a steady state situation.

In this case, the temperature difference (ΔT) between EWT and LWT towards the end of the experiment was roughly maintained (Figure 42). However, both temperatures (EWT and LWT) were still trending down, as can be seen in Figure 40. This indicates that the system did not reach steady state after three days under the particular conditions of the experiment. A representative ΔT was taken at the end of the continuous load portion of the test with an average of the last 2.4 hr equal to 3.36°F (1.9°C).

The negative ΔT found between days 2 and 3 in Figure 42 is an anomaly caused by the GSHP going into cooling mode inadvertently; the GSHP's internal controls were not yet fully debugged during *phase I* of the experiments.

Flow rate was measured at 30 second intervals during the first 15.6 hours of the experiment. A plot of flow rate against time is shown in Figure 43. The circulating pump was set to constant flow. The average flow rate was 9.85 gpm (0.62 l/s) with a standard deviation of 0.24°F.

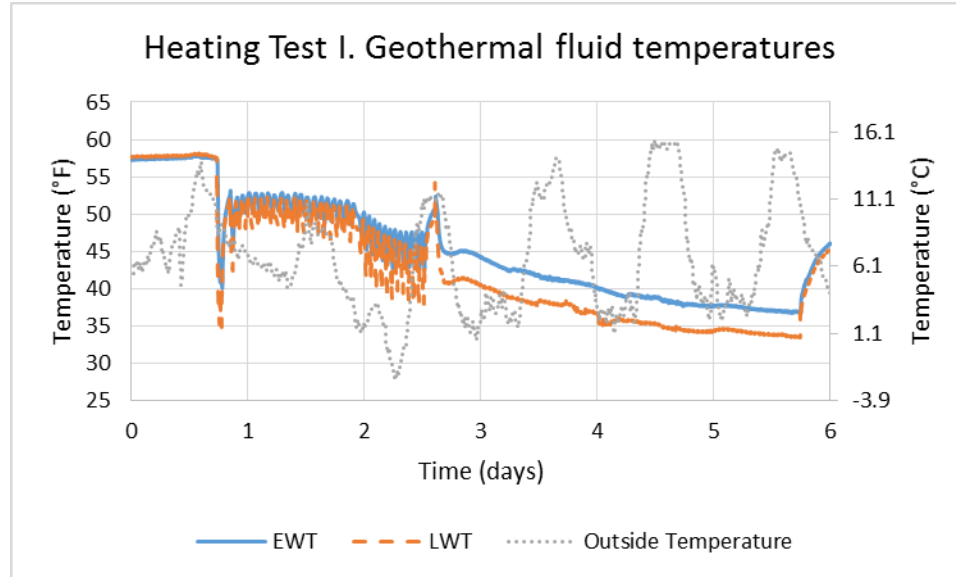


Figure 40. Heating Test I

Performance of the helical GHEs was calculated according to Equations (4) and (5), as for the previous experiment. A plot of performance versus time is shown in Figure 44.

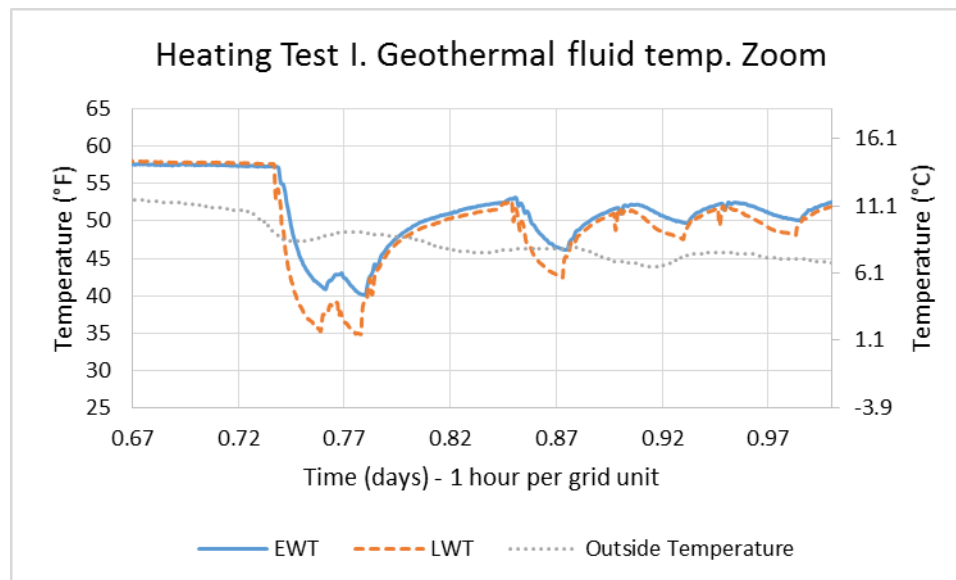


Figure 41. Heating Test I. Zoom of a period of 8 hours during the first day to note the loading and unloading cycles of the GSHP

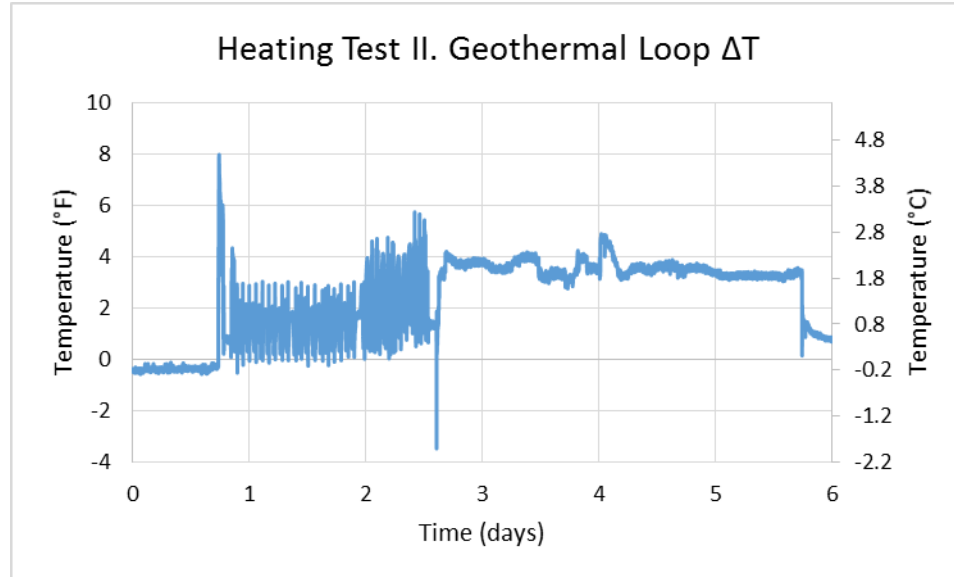


Figure 42. Analysis of temperature difference between inlet and outlet of the GHE (EWT – LWT) during Heating Test I

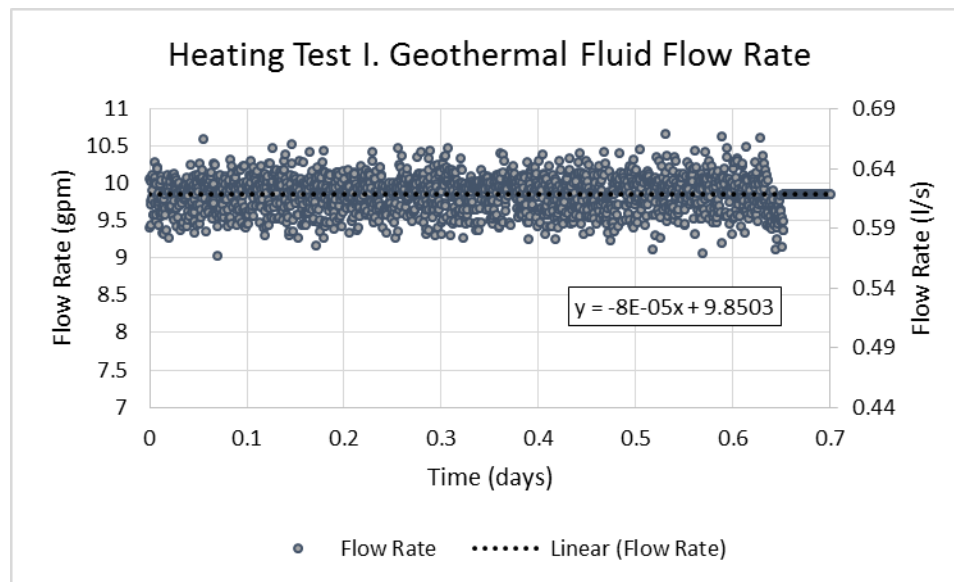


Figure 43. Volumetric flow rate of the geothermal fluid during Heating Test I. The graphic shows a linear fit to the data and its equation

A representative performance is calculated below using the same procedure as in previous experiments.

$$\dot{m} = 9.85 \frac{\text{gal}}{\text{min}} \cdot 0.1337 \frac{\text{ft}^3}{\text{gal}} \cdot 63.63 \frac{\text{lb}_m}{\text{ft}^3} \cdot 60 \frac{\text{min}}{\text{hr}} = 5027.8 \frac{\text{lb}_m}{\text{hr}}$$

Then

$$q = 5027.8 \frac{\text{lb}_m}{\text{hr}} \cdot 0.951 \frac{\text{Btu}}{\text{lb}_m \cdot ^\circ\text{F}} \cdot 3.36^\circ\text{F} = 16065.7 \frac{\text{Btu}}{\text{hr}}$$

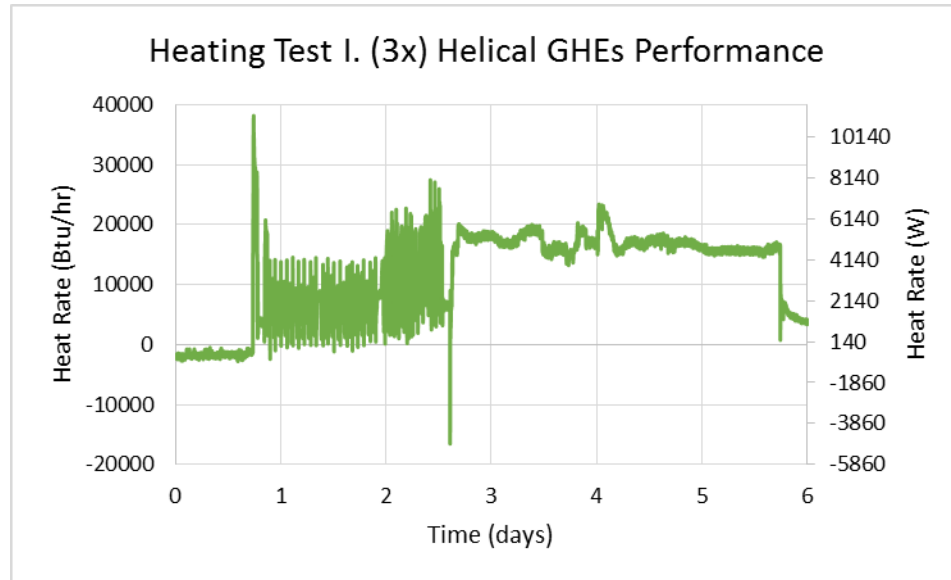


Figure 44. Performance analysis of the three helical GHEs at the RMEH during Heating Test I

This is the performance of the three helical GHEs in place, so the unitary representative performance for this conditions is

$$\frac{16065.7 \frac{\text{Btu}}{\text{hr}}}{3 \text{ helix}} = 5355 \frac{\text{Btu}}{\text{hr} \cdot \text{helix}} \left(1570 \frac{\text{W}}{\text{helix}} \right)$$

Heating Test II

Heating Test II ran for 29 days during October and November 2015. Entering and leaving fluid temperatures (EWT and LWT) were measured at 5 minute intervals. The

average temperature of the ground at the beginning of the test (average of two days previous to first GSHP startup) was 72.7°F (22.6°C). At the end of the test, the temperature decreased to a minimum of 31.1°F (-0.5°C).

EWT and LWT measurements were plotted against time (Figure 45). The first part of the experiment is zoomed in Figure 46 to show the daily cycles with loading stages during the cold hours of the day and resting periods when the ground “recharges” and drifts back towards the undisturbed temperature. Only by adjusting the scale to a few hours can the temperature difference be clearly noticed. For instance, during one of the mornings of the testing period, the GSHP had to work continuously for approximately four hours at the maximum allowed capacity which would not trip the GSHP’s low suction pressure safety measures (Figure 47). During that period, a good grasp of the maximum performance at which the helical GHE can work was obtained.

Temperature difference between EWT and LWT was analyzed and plotted against time (Figure 48). The negative values are anomalies. They may be caused by the relatively large time step (5 minutes), which could have not fully captured some of the rapidly changing temperatures occurring during on/off cycles. Also, the heat contributed by the wet rotor circulating pump might have had an influence.

Volumetric flow rate was measured at 5 minute intervals for the duration of the experiment. A plot of flow rate against time is shown in Figure 49. The circulating pump was set to constant flow. However, this graphic revealed an interesting fact: the volumetric flow rate measured by the flow sensors showed a “swinging” behavior, especially towards the end of the experiment. Furthermore, the swings matched with the

swings in EWT/LWT measurements caused by the GSHP on/off daily cycles. This was most likely caused by the geothermal fluid becoming more viscous as the temperature decreased, thus affecting the flow rate. These oscillations can be up to 20°F (11.1°C) or more in one day (see Figure 45).

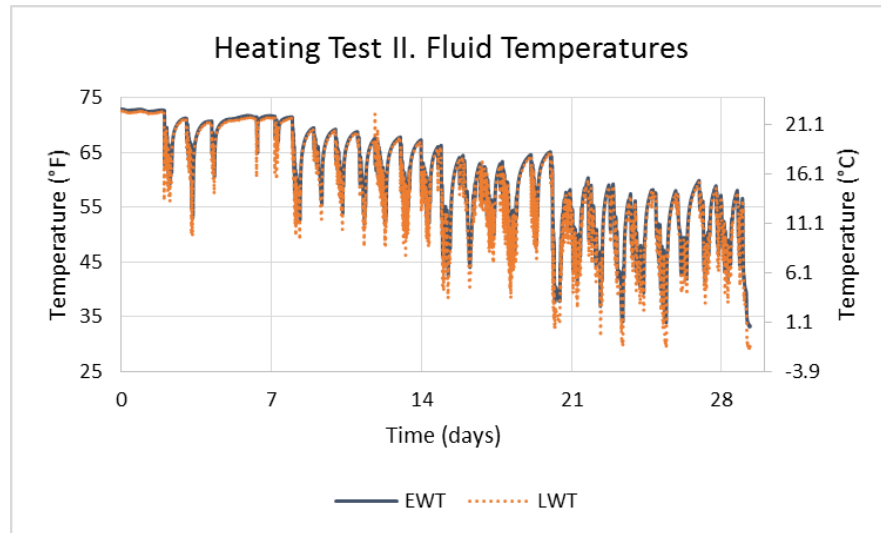


Figure 45. Heating Test II

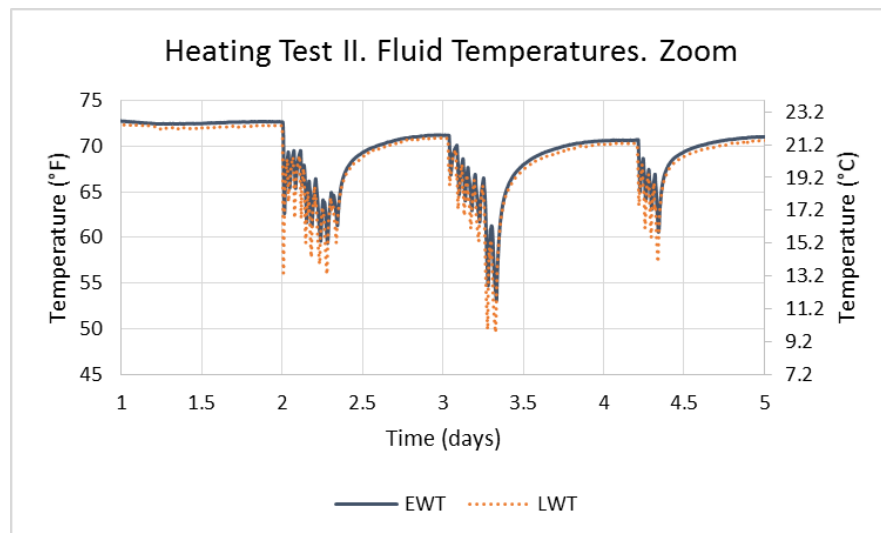


Figure 46. Heating Test II. Zoom of the geothermal fluid temperatures during the first 5 days

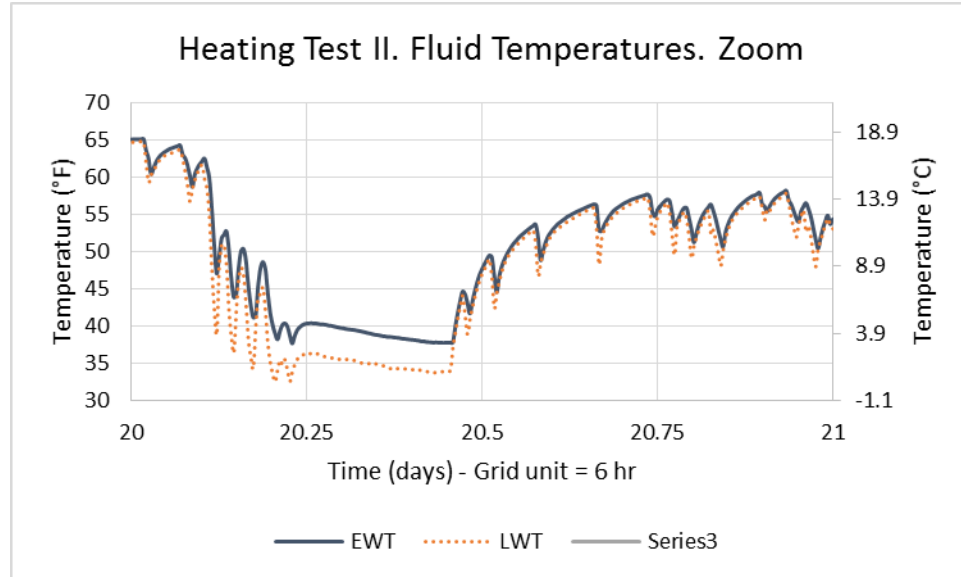


Figure 47. Heating Test II. Zoom of the geothermal fluid temperatures during the 20th day of the experiment

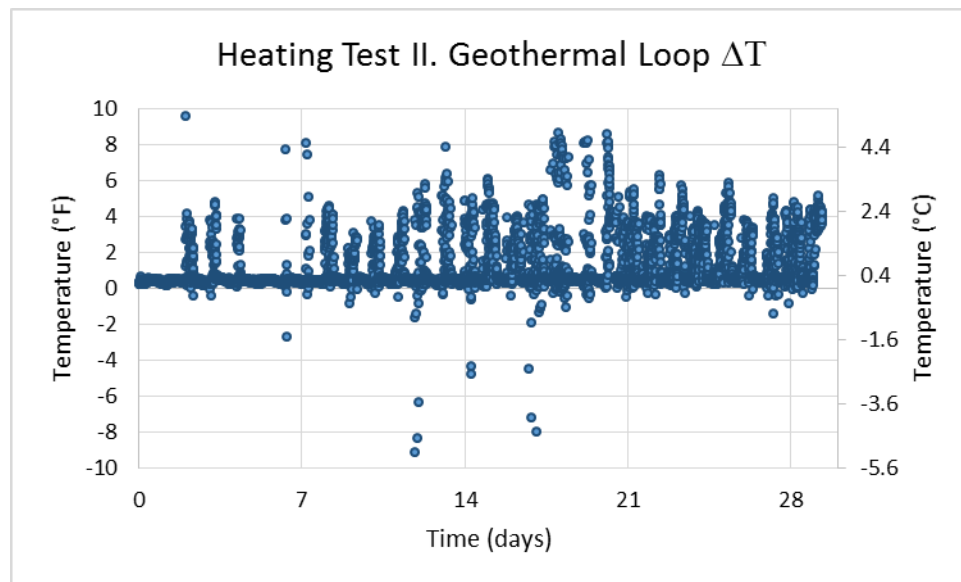


Figure 48. Analysis of temperature difference between inlet and outlet of the GHE (EWT – LWT) during Heating Test II

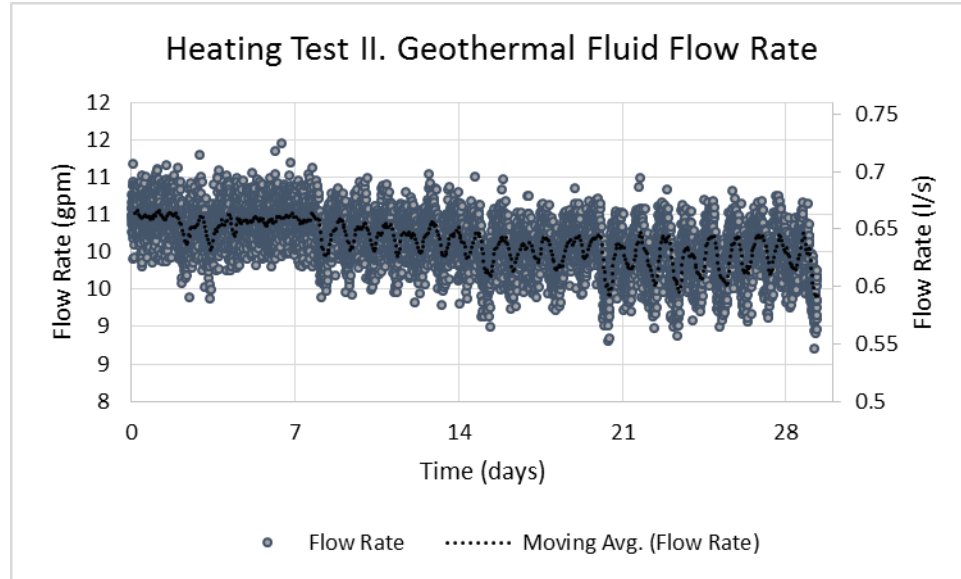


Figure 49. Volumetric flow rate of the geothermal fluid during Heating Test II. The graphic shows the raw measurement and a moving average fit (in dotted black).

As mentioned earlier, in order to obtain a gauge of the performance of the helical GHEs under these particular conditions, day 20 of the experiment was selected (Figure 47). During that morning the GSHP cycle was the longest of this experiment (roughly 5 hours). The average ΔT of that cycle was 4.03°F (2.2°C) and the average flow rate was 9.45 gpm (0.596 l/s). Performance calculations are then:

$$\dot{m} = 9.45 \frac{\text{gal}}{\text{min}} \cdot 0.1337 \frac{\text{ft}^3}{\text{gal}} \cdot 63.42 \frac{\text{lb}_m}{\text{ft}^3} \cdot 60 \frac{\text{min}}{\text{hr}} = 4807.7 \frac{\text{lb}_m}{\text{hr}}$$

Thus,

$$q = 4807.7 \frac{\text{lb}_m}{\text{hr}} \cdot 0.952 \frac{\text{Btu}}{\text{lb}_m \cdot ^{\circ}\text{F}} \cdot 4.03^{\circ}\text{F} = 18445.2 \frac{\text{Btu}}{\text{hr}}$$

This corresponds to the performance of the three helical GHEs in place, so the unitary representative performance for these conditions is:

$$\frac{18445.2 \frac{Btu}{hr}}{3 \text{ helix}} = 6148 \frac{Btu}{hr \cdot \text{helix}} \left(1802 \frac{W}{\text{helix}} \right)$$

A plot of the instantaneous performance during day 20 of the experiment is shown in Figure 50.

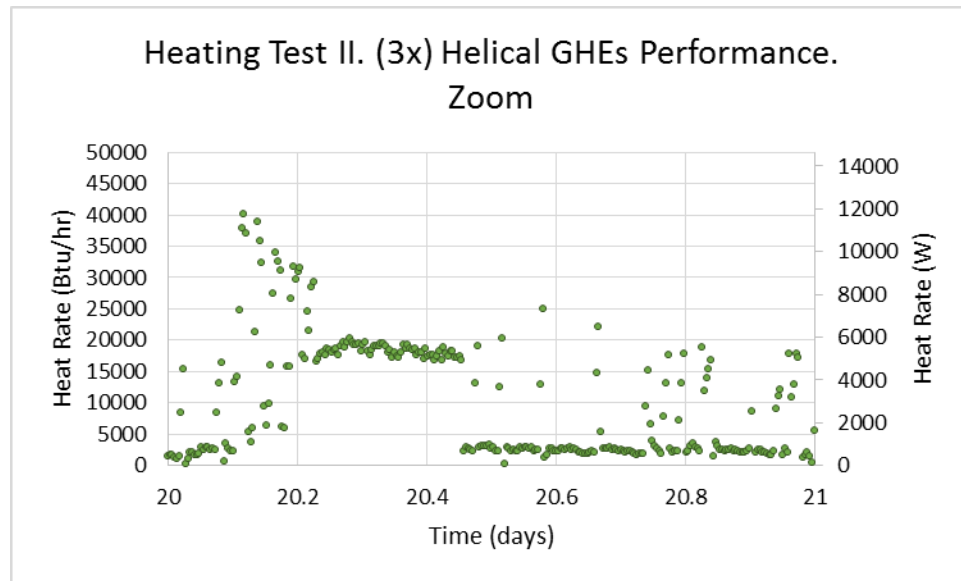


Figure 50. Performance analysis of the three helical GHEs at the RMEH during Cooling Test II

Results from the additional experiments Cooling Test III and Heating Test III can be found in APPENDIX C.

Performance Summary

A summary of experimental results is shown in Table 10. Results are summarized in a per unit helical GHE basis and grouped by cooling/heating test type. Other useful information, like rate of heat transfer per feet of excavated borehole depth and total length of pipe is also presented for practical comparison with other GHE systems.

Table 10. Performance Summary of the helical GHEs during the tests

Test ID	Test Date	Unitary Heat Absorbed/Rejected by Helical GHE ¹			
		(Btu/ hr·helix)	(W/helix)	(Btu/ hr·ft-depth)	(Btu/ hr·ft-pipe)
Cooling Test I	8/1 to 8/5, 2014	1965	576	164	15
Cooling Test II	8/11 to 9/22, 2015	1713	502	143	13
Cooling Test III	7/15 to 7/20, 2015	4643	1361	387	35
Heating Test I	2/9 to 2/14, 2015	5355	1570	446	41
Heating Test II	10/13 to 11/11, 2015	6148	1802	512	47
Heating Test III	12/11, 2015	2229	653	186	17

1 Values are considered positive for both heat absorbed/rejected

Simulation Results and Model Validation

In this section, results from the application of the in-situ experimental data to the CaRM model are presented, along with a comparison between experimental and simulation results. Two indicators of the model's degree of accuracy were used:

1. Mean bias error (MBS)
2. Root mean squared error (RMSE)

MBS is calculated according to Equation (37):

$$MBE = \frac{1}{n} \sum_{i=1}^n \frac{x_i - x_{true}}{x_{true}} \quad (37)$$

where

n = number of measurements
 x_i = calculated value
 x_{true} = measured value

RMSE is calculated according to the following Equation (38):

$$RMSE = \left[\frac{1}{n} \sum_{i=1}^n \left(\frac{x_i - x_{true}}{x_{true}} \right)^2 \right]^{1/2} \quad (38)$$

Cooling Test I

A comparison between measured and simulated EWT during Cooling Test I can be seen in Figure 51. The accuracy indicators MBE and RMSE are 0.41°F (0.23°C) and 0.49°F (0.27°C) respectively. The difference between measured and calculated performance was 23%.

Cooling Test II

Measured and simulated EWT during Cooling Test II can be seen in Figure 52. MBE and RMSE are -0.03°F (-0.01°C) and 0.16°F (0.09°C) respectively. The difference between measured and calculated performance was 4%.

Heating Test I

A comparison between measured and simulated EWT during Heating Test I can be seen in Figure 53. MBE and RMSE are -2.3°F (-1.28°C) and 2.56°F (1.42°C) respectively. The difference between measured and calculated performance was 89%.

Heating Test II

Measured and simulated EWT during Heating Test II can be seen in Figure 54. MBE and RMSE are -1.43°F (-0.79°C) and 1.58°F (0.88°C) respectively. The difference between measured and calculated performance was 66%.

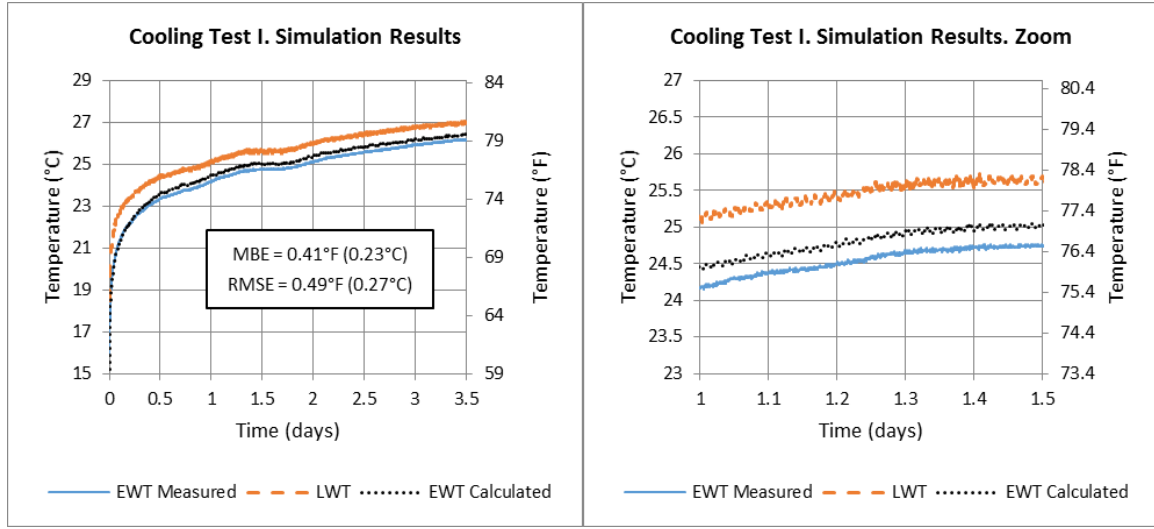


Figure 51. Comparison between measured and simulated EWT during Cooling Test I. Left: whole test; right: zoom of first half of second day of experiment. Mean bias error (MBE) and root mean squared error (RMSE) are stated in the box

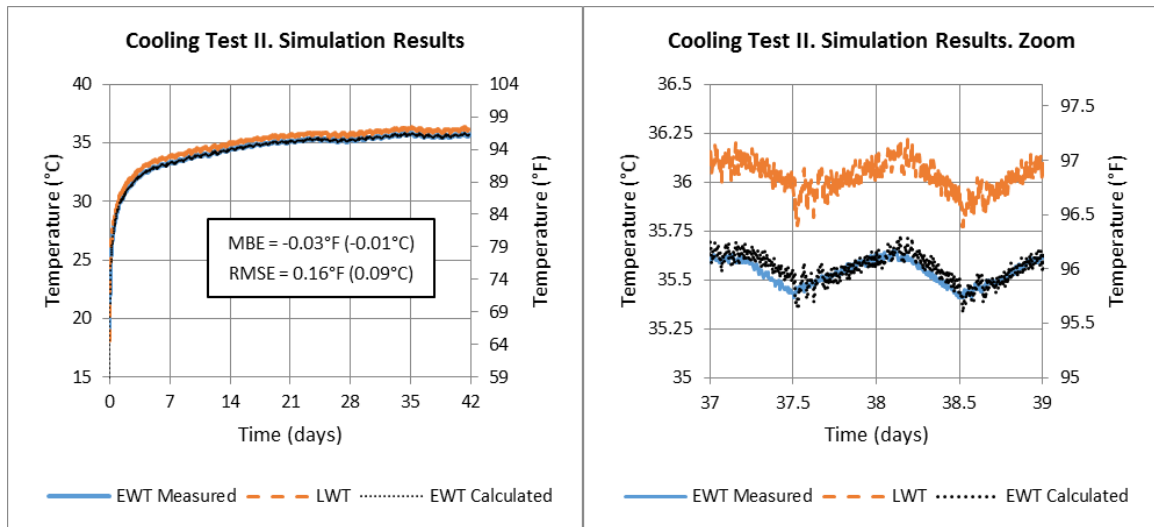


Figure 52. Comparison between measured and simulated EWT during Cooling Test II. Left: whole test; right: zoom of days 38 and 39 of experiment. Mean bias error (MBE) and root mean squared error (RMSE) are stated in the box

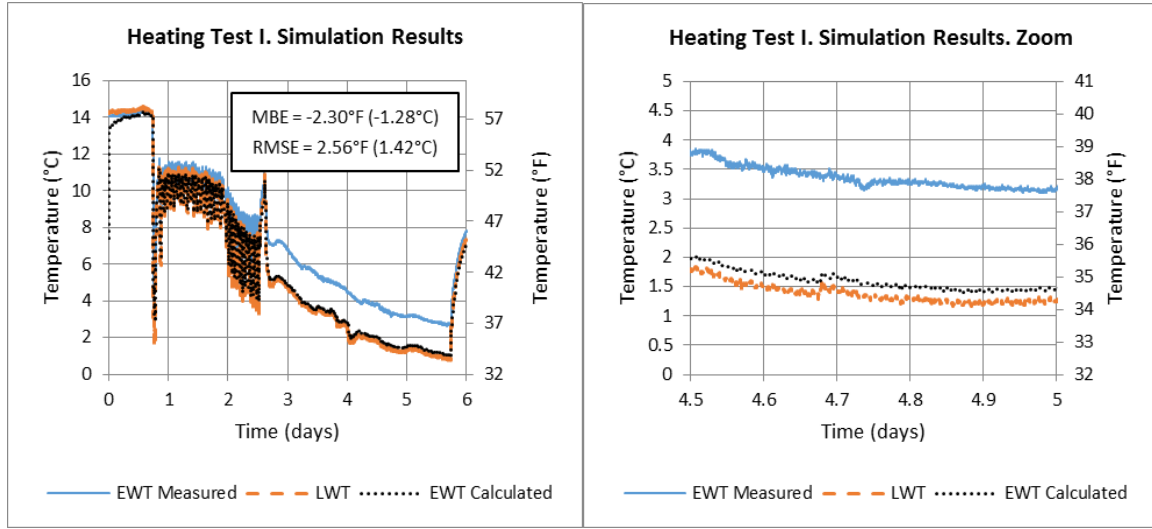


Figure 53. Comparison between measured and simulated EWT during Heating Test I. Left: whole test; right: zoom of second half of fifth day of experiment. Mean bias error (MBE) and root mean squared error (RMSE) are stated in the box

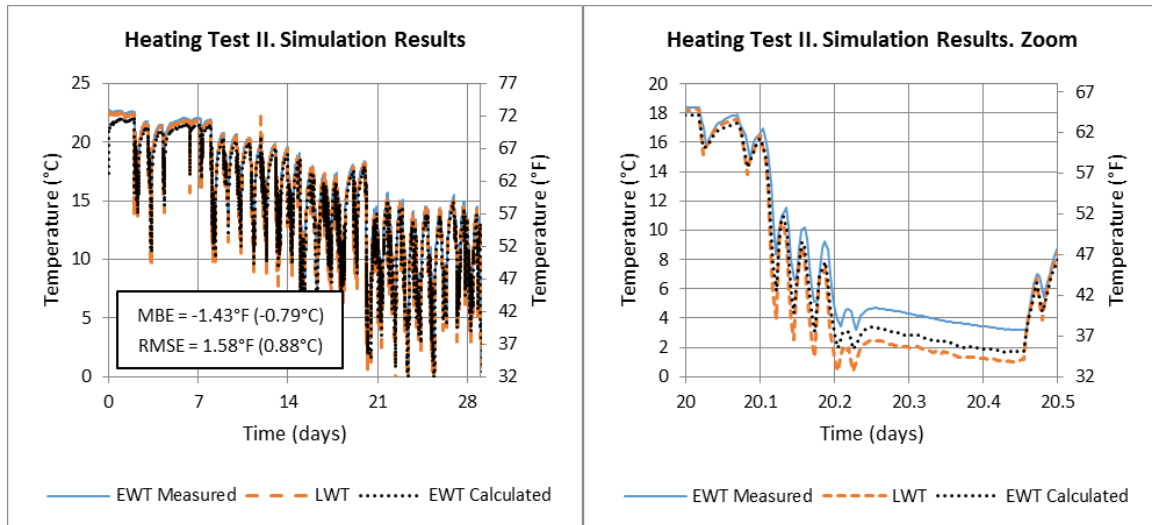


Figure 54. Comparison between measured and simulated EWT during Heating Test II. Left: whole test; right: zoom of first half of day 20. Mean bias error (MBE) and root mean squared error (RMSE) are stated in the box

Results from the additional experiments Cooling Test III and Heating Test III can be found in APPENDIX C.

Summary of Results

A summary of experimental and simulation results is presented in this section. For each test, the model errors indicators (MBE and RMSE) are shown, along with a comparison between measured versus predicted representative performance (Table 11).

Table 11. Summary of experimental versus simulated results and model validation parameters

Test ID	Test Date	Degree of Closeness		Unitary Heat Absorbed/Rejected by Helical GHE				
		MBE °F (°C)	RMSE °F (°C)	Measured		Predicted		% diff.
Cooling Test I	8/1 to 8/5, 2014	0.41 (0.23)	0.49 (0.27)	1965	576	1522	446	23
Cooling Test II	8/11 to 9/22, 2015	-0.03 (-0.01)	0.16 (0.09)	1713	502	1638	480	4
Cooling Test III	7/15 to 7/20, 2015	-0.29 (-0.16)	1.06 (0.59)	4643	1361	3911	1146	16
Heating Test I	2/9 to 2/14, 2015	-2.3 (-1.28)	2.56 (1.42)	5355	1570	580	170	89
Heating Test II	10/13 to 11/11, 2015	-1.43 (-0.79)	1.58 (0.88)	6148	1802	2111	618	66
Heating Test III	12/11, 2015	-1.68 (-0.93)	1.74 (0.97)	2229	653	-246	-72	111

DISCUSSION

Experimental Uncertainty

One of the aspects that first stands out from the outcome of this thesis is the high degree of experimental uncertainty. The uncertainty in the results for the GHE performance analysis was found to be in the order of 23%. This was mainly caused by the high uncertainty level in one of the measured variables: the temperature difference between inlet and outlet of the GHE or ΔT as it has been typically referred to throughout this thesis. In turn, this high level of uncertainty has two main combined causes. One is that ΔT was typically very small (sometimes in the order of 1°F [0.6°C]). Flow rates on the geothermal loop were selected based on obtaining a turbulent flow in the pipes that enhanced the heat transfer. This turned out to be counter-productive because the circulating pump was working hard and introduced a significant amount of residual heat into the loops. This was clearly a source of error because the assumption was no external losses or gains in the geothermal loops other than those produced by the GSHP. Selecting smaller flow rates would have been beneficial since the overall ΔT would have increased considerably along with a decrease in pumping power. On the other hand, small flow rates would have produced a laminar flow in the pipes, and a corresponding decrease in heat exchange efficiency. It would be interesting to investigate the performance of the GHE at lower flow rates. The other main cause of high uncertainty in the measurement of ΔT was the selected sensors. Type-T thermocouples in the conditions of the experiments were found to have an error of $\pm 0.2^\circ\text{F}$ (0.1°C). This may seem like a low level of uncertainty, but compared to 1°F, it is a 20% error. A combination of smaller flow rates

and more precise temperature sensors would have drastically decreased the uncertainty of the experimental results.

Model Input Data Uncertainty

A comprehensive analysis of uncertainty in the application of the CaRM tool was outside of the scope of this thesis. However, it is important to point out the sources of error of the input data used by the simulation tool.

- **Thermal Fluid.** The composition of the heat carrier fluid was determined by a simple handheld refractometer test. The percentage of propylene glycol in the mixture is visually determined by eye sight on a scale with a $\pm 5\%$ resolution and is affected by the temperature of the sample. In addition, the model uses average thermal fluid properties as input, like density, specific heat and kinematic viscosity; however, all these properties are temperature dependent and the temperature of the fluid is not typically constant throughout the test.
- **Ground properties.** Ground properties like thermal conductivity, thermal diffusivity, density and heat capacity were estimated based on tables from ASHRAE (ASHRAE, 2015) and a drilling log from the on-site drilling works. However, there is no indication of the water content. Water content can have a significant influence in the soil properties and therefore in the heat conduction process when coupled to a GHE. There is a high degree of uncertainty with respect to the true thermal properties of the ground.

- Climate conditions. Outdoor air temperature was determined with a high degree of accuracy since there are sensors in-situ. However, solar radiation had to be estimated via statistical records from a nearby weather station. This would be a correct practice for design purposes, but not ideal when the intention is to calibrate or validate a model.

Experimental Versus Predicted Results

From the analysis of the two indicators used to measure the degree of closeness of the model predictions with respect to the experimental data (MBE and RMSE), it can be affirmed that there are two different tendencies (Table 11). The three cooling tests showed smaller errors than the three heating tests. In the cooling tests, mean bias error (MBE) ranged between -0.29 and 0.41°F (-0.16 and 0.23°C); whereas, root mean squared error (RMSE) ranged between 0.16 and 1.06°F (0.09 and 0.59°C). That certainly looks like a small error for a scale of typical fluid temperatures in cooling mode (around 50 to 90°F [10 to 32.2°C]), but it is not that small if compared with the ranges of temperature differences which occurred during the cooling mode tests, 0.9 to 2.6°F (0.5 to 1.4°C). Looking at the differences in performance (Table 11), it can be observed that the model always under-predicted the performance ranging from 4 to 23%.

In heating mode the differences are more pronounced. MBE ranged between -2.3 and -1.43°F (-1.28 to -0.79°C) and RMSE ranged between 1.58 and 2.56°F (0.88 to 1.42°C). These negative bias errors consistently indicate that the model is under-estimating the performance of the GHE in heating mode. This is confirmed with the tendency in the measured versus predicted performance calculations (Table 11).

Predicted performance is very small compared to the actual measurements; for example, 2111 Btu/hr (618 W) vs 6149 Btu/hr (1802 W) in Heating Test II.

The Importance of Sources of Error and Assumptions

A possible explanation for the mismatch between model prediction and actual measurements could be the combination of three factors:

- As mentioned earlier in the Sources of Error section, some of the pipes in the mechanical room at the RMEH are not insulated. A rough estimation of the heat gains from the mechanical room to the cold fluid during heating mode operation yielded around 220 Btu/hr (65 W) added to the GHE loop. In cooling mode operation, the temperatures of the heat carrier fluid are closer to room temperature, thus significantly attenuating the potential heat losses.
- It has also been mentioned that the circulating pump was set to high flow rates during the experiments, and this introduced a significant amount of residual heat into the geothermal fluid loops. It was estimated that the power consumption of the pump was around 290 W (890 Btu/hr). To put it in perspective, product literature estimates the performance of one helical GHE in average operation conditions to 400 W/helix (1365 Btu/hr·helix). The peculiarity of this source of error is that it works as an extra load in cooling mode and as credit in heating mode. In other words, the system is working as if it had almost only two helix probes in cooling mode and four helix probes in heating mode.

- Lastly, one of the assumptions of this research was that the horizontal loops connecting the helical GHEs with the mechanical room were insulated. However, the performance of the system, especially in heating mode, is remarkably higher than what would have been expected for the assumed configuration alone. This finding suggests that maybe the horizontal loops are not insulated; thus, there is an equivalent horizontal GHE in series with each helical GHE loop. According to literature (Reuss & Sanner), an estimation for a two pipe loop in a 6 foot deep trench on average soil conditions could be around 20 to 30 W/m-trench (224 Btu/hr·ft-trench). For a 128 ft (39 m) trench, taking 25 W/m gives 3360 Btu/hr (985 W) per helix loop.

In heating mode, these three contributions add up to:

$$3360 \times 3 \text{ helices} + 890 + 220 = 11190 \text{ Btu/hr}$$

And the difference between the actual measurements and the model prediction in the case of Heating Test II is:

$$6149 \times 3 - 2111 \times 3 = 12115 \text{ Btu/hr}$$

The results are surprisingly close, supporting the hypothesis of the horizontal loop influence being actually true. Unfortunately, further investigations to confirm this end are outside the scope of this thesis.

CONCLUSIONS

Achieved Objectives

This thesis provided practical insight about the operation and testing of a system of helical GHEs coupled with a GSHP. The facilities and experiments were described in detail and the experimental results were presented and analyzed. Also, data from these experiments was used as input for a previously reported simulation tool, a capacity-resistance based model (CaRM) to predict performance of various types of GHE, including the helical one.

The purpose of validating the model was not totally fulfilled due to the uncertainties in both the experimental setup and the input data required by the model. Specifically, the horizontal runs that connected the helical GHEs with the mechanical room at the RMEH were suspected to be non-insulated, unlike initially assumed, having a significant influence on the overall performance of the system. This version of the CaRM is built in such a way that the influence of connecting horizontal runs is not taken into account. In fact, it was the significant differences between the model and the measurements, especially in heating mode, which triggered the discovery of the possibly erroneous assumption.

Despite the fact stated in the previous paragraph, it is clear that the helical GHE is an alternative to the traditional horizontal and deep borehole GHE installations. It is the responsibility of the geothermal designer to be aware of novelty alternatives to solve GSHP projects, which, in a case-by-case basis can provide the customer with the most advantageous installation possible.

Recommendations and Future Work

This section is divided in recommendations to improve the experimental setup, recommendations to improve the model, and a list of other interesting research lines.

Improving the Experimental Setup

- An investigation on the actual configuration of the horizontal runs that connect the GHEs with the mechanical room would enable for the GHE geometry to be more accurately modeled.
- The influence of the heat originated by the pumping operation would be largely decreased if smaller flow rates were selected to run in the helical GHE loops. An additional benefit would be the increase in temperature difference between inlet and outlet of the GHE for the same power input, therefore equalizing the uncertainty levels relative to the measured variables, i.e. temperature difference and flow rate. The only drawback of this measure would be a reduction of the convective heat transfer from the fluid to the pipe walls due to a less turbulent (or even laminar) flow. Care must be also taken while reducing the flow rates since the volumetric flow rates used in the experiments are already in the lower end of the recommended operational range of the flow meters.
- Selection of sensors with narrower uncertainty levels would decrease the overall uncertainty of the performance calculations.
- For research purposes, the geothermal loops in the mechanical room should be completely insulated. Even the thermoplastic (PEXa) pipes have

significant losses when the temperature gradient is large. In heating mode, temperatures on the geothermal loops can easily reach 30 to 40°F (-1.1 to 4.4°C), which can lead to important heat losses/gains to or from the mechanical room air (typically at 70°F [21°C] or more). This would also prevent condensation related problems.

- Some of the model inputs are physical and thermal properties of the ground. For an accurate characterization of the ground, not only a detailed composition is needed, but also a determination of the water content, since this can have a significant influence over the performance of the heat exchanger.
- Another model input, the global horizontal solar radiation, was retrieved from the nearest weather station average hourly statistics provided by EnergyPlus[®], a building energy simulation program from the U.S. Department of Energy (EnergyPlus, 2015). Actual local solar data would improve the accuracy of the model, especially for model calibration purposes.

Improving the Model

- Including the possibility to model horizontal runs in the CaRM would improve the overall versatility of the model since it would allow for more realistic systems to be simulated.
- Also, the possibility to account for extraneous heat inputs like the residual heat from the pumping operation would be interesting.

- Another improvement would be to add accountability for seasonal soil moisture variations and other soil water content related effects.

Other Interesting Research Lines

Additional lines of research available at the RMEH that would be interesting to explore are, among others:

- The study of the ground heat exchanger is only a piece of the puzzle. GHEs are coupled with the GSHP equipment and in turn, the latter is coupled with the building through its heating and cooling systems. Individual performance variations affect the efficiency of the whole system. A holistic approach to energy modeling should be undertaken to evaluate the overall efficiency. In this sense, there are already some advances in this field (Zarrella et al., 2013), as mentioned in the LITERATURE REVIEW section.
- The potential benefit of “solar recharge” in GSHP applications can be substantial. The RMEH has a “heat sink” as one of its cutting-edge technologies. The heat sink is designed to accumulate the excess of solar radiation during the summer from the solar thermal panels and re-inject that energy back into the geothermal loops during the heating season. However, this feature has not yet been investigated. There are also several sunny days in the winter when the solar panels are warm but still not warm enough to be used for domestic hot water heating. The special configuration of the HVAC systems at the RMEH would allow use of this

“low-grade” heat to aid the GSHP’s low temperature heating systems such as radiant floors, increasing the overall efficiency.

- Further investigation on how different flow patterns (laminar vs. turbulent; fixed vs. variable) affect the overall efficiency of the system would also be interesting.

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APPENDICES

APPENDIX A

EQUIPMENT SPECIFICATIONS

AAON INC.
2425 S. YUKON TULSA, OKLAHOMA
VENDOR: AAON PART NUMBER
TO BE ON CARTON OR PART ITSELF.
SOLIDEDGE MODEL # _____

REV.	BY	ECN NUMBER	DATE
NEW	BA	1543AK	09/15/08

DWG BY:	S. HALL	PCN #	R 63010	REV.
DATE:	10/02/08			NEW
SHEET	1 / 1	APPD BY:		

COMPONENT: _____ COMP ZPD34K5E-PFV-130 208/230V

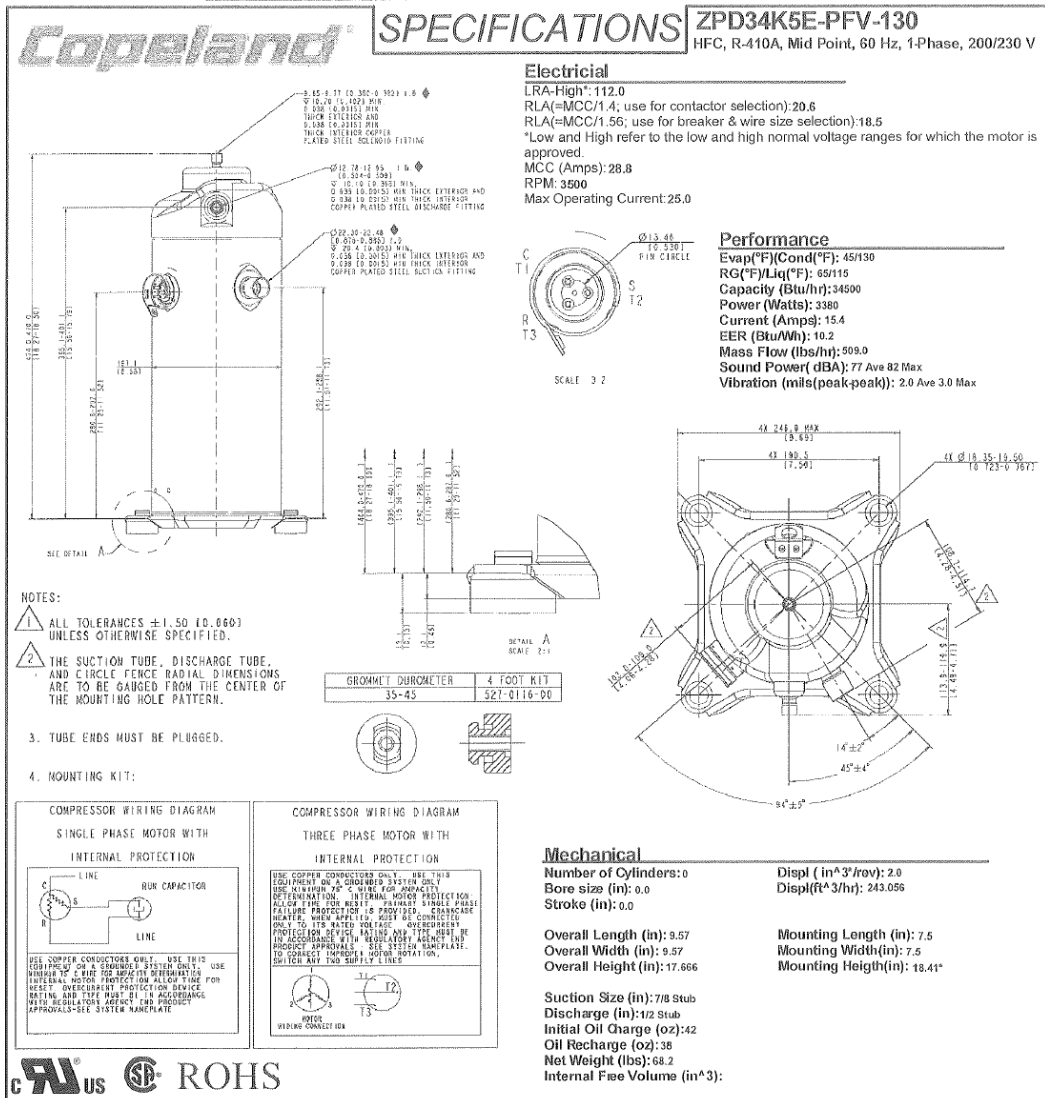
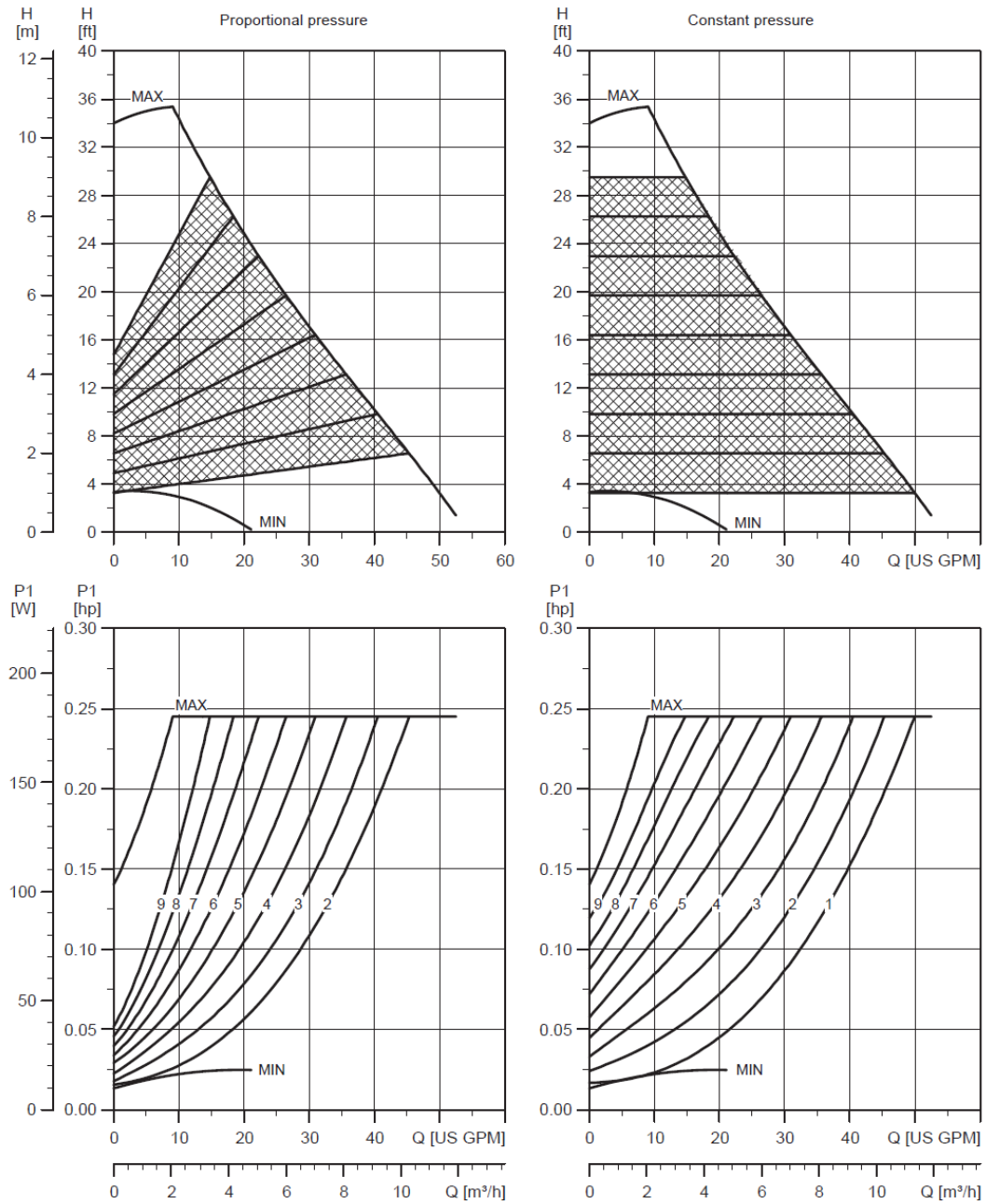


Figure 55. Specification sheet of the compressors in the GSHP at the RMEH

MAGNA 32-100 F

TMD4 9409 4110

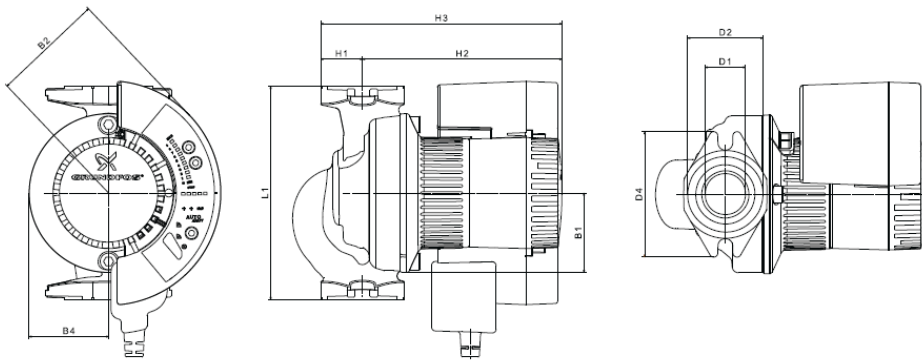
Figure 56. Specification sheet of primary water loop circulation pump (1 of 2)

MAGNA 32-100 F

Electrical data

U_n [V]		P_1 [W]	$I_{1/1}$ [A]
1 x 208 - 230	Min.	10	0.1 A
	Max.	180	1.37 - 1.23 A

MAGNA 32-100 F is also available with stainless-steel housing, type N.



TM04 9443 5110

Dimensions

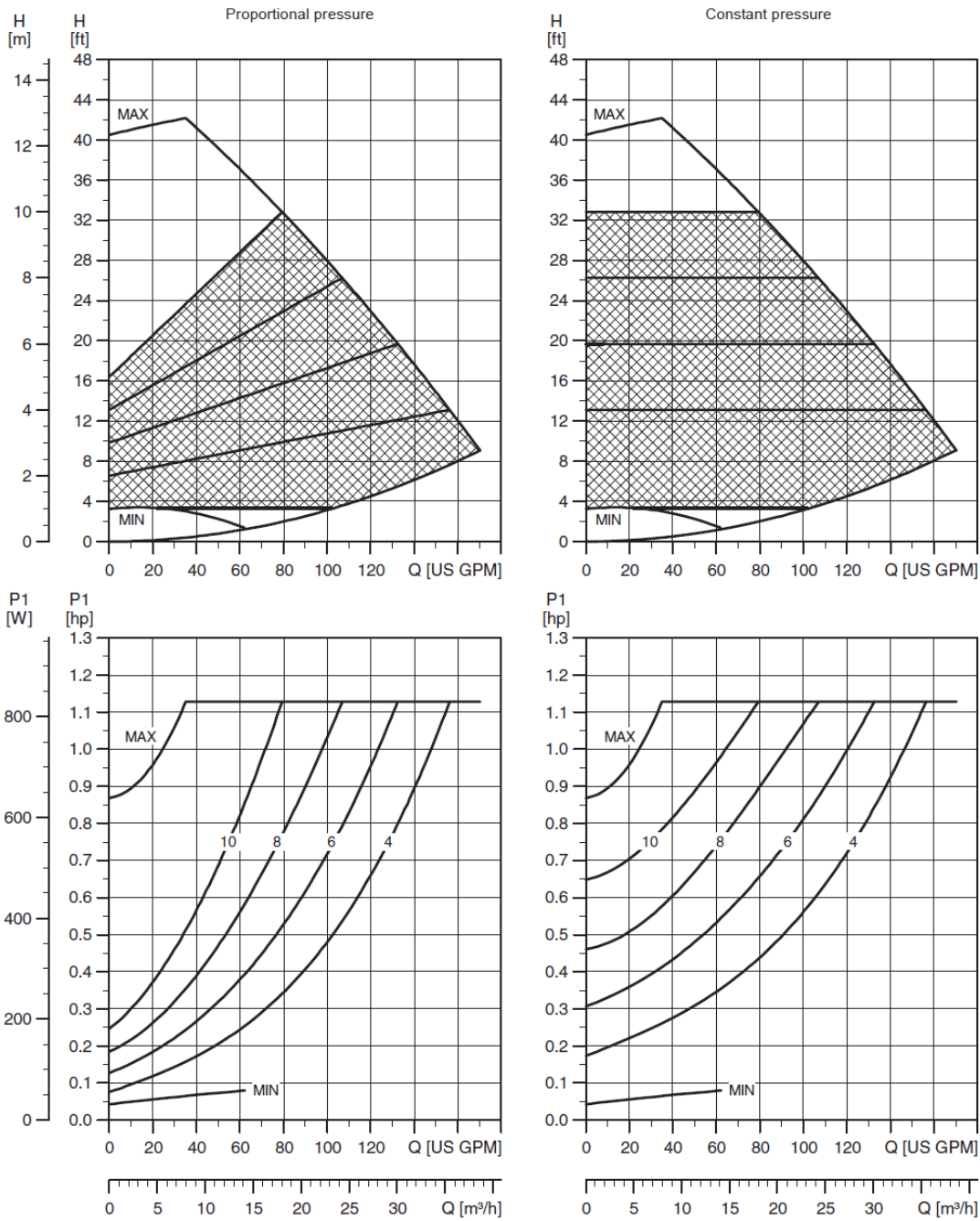
Pump type	Dimensions										
		L1	B1	B2	B4	H1	H2	H3	D1	D2	D4
MAGNA 32-100 F	[in]	6 1/2	2 3/8	3 3/8	2 3/8	1 1/4	6 1/8	7 3/8	1 1/4	2 1/4	3 7/8
	[mm]	165	61.5	86.9	61.5	32.3	154.5	186.8	31.1	58.5	97.1

Weights

Pump type	Weights [lb (kg)]
	Gross
MAGNA 32-100 F	13 (6)

Figure 57. Specification sheet of primary water loop circulation pump (2 of 2)

MAGNA 65-120 F

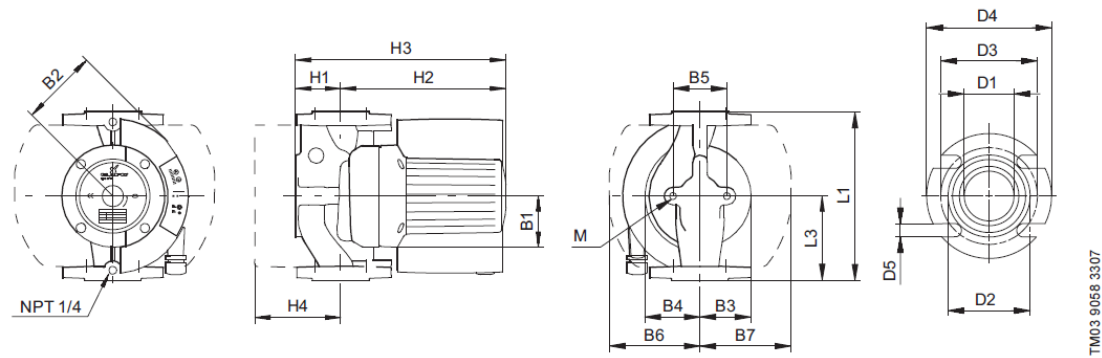


Electrical data

Figure 58. Specification sheet of geothermal loop circulation pump (1 of 2)

U_n [V]		P_1 [W]	$I_{1/1}$ [A]
1 x 230 V	Min.	35	0.28
	Max.	900	3.9

MAGNA 65-120 F is also available with stainless-steel housing, type N.



Dimensions and weights

Pump type	Dimensions															
	L1	L3	B1	B2	B3	B4	B5	B6	B7	H1	H2	H3	H4	D1	D2	D3
MAGNA 65-120 F	[in]	11 1/2	5 3/4	3 1/16	4 15/16	3 7/16	4 1/8	3 3/4*	5 1/2	4 5/16	3 1/4	9 15/16	13 1/8	4 3/16	2 1/2	3 15/16
	[mm]	292	146	77	125	88	104	96*	140	110	82	252	334	107	63	100

Pump type	Dimensions		
	D4	D5	M
MAGNA 65-120 F	[in]	6	5/8
	[mm]	152	16

* Stainless steel pump housing 3 1/8 [in]/80 [mm]

Weights

Pump type	Weights [lb (kg)]	
	Net	Gross
MAGNA 65-120 F	56 (25.5)	60.5 (27.5)

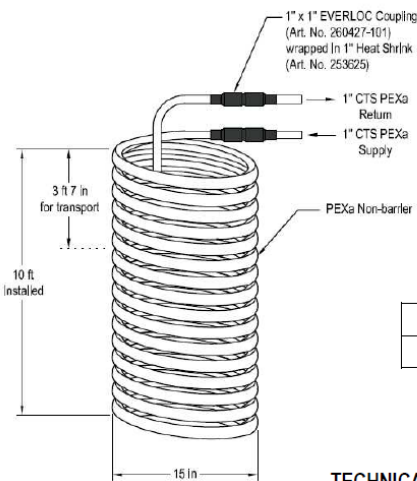
Figure 59. Specification sheet of geothermal loop circulation pump (2 of 2)

Construction
Automotive
Industry



PRODUCT SUBMITTAL 153

Product: RAUGEO™ Helix
Date: 5-August-10



Article No.	Description	Length of Pipe	Weight
630406-001	1" RAUGEO Helix Probe	131 ft (39.9 m)	16.5 lbs (7.5 kg)

TECHNICAL DESCRIPTION OF RAUGEO® PEXa PIPE

Specification	English	SI	Standard
Minimum Density	58 lb/ft³	926 kg/m³	ASTM F876 DIN 53479
Degree of Cross Linking	70-89%	70-89%	ASTM F876 DIN 16892
Thermal Conductivity	0.24 BTU/(hr ft °F)	0.41 w/(m°K)	DIN 52612
Linear Exp.	9.33 x 10-4 in/ft°F @ 68°F 1.33 x 10-3 in/ft°F @ 212°F	0.14 mm/(m°C) @ 20°C 0.2 mm/(m°C) @ 100°C	DIN 42328
Modulus of Elasticity	87,000-130,500 psi @ 68°F 43,500-58,000 psi @176°F	600-900 N/mm2 @ 20°C 300-400 N/mm2 @ 80°C	DIN 53457

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Before using, the user will determine suitability of the information for user's intended use and shall assume all risk and liability in connection therewith. © 2010 REHAU

Figure 60. Specification sheet of helical ground heat exchanger (1 of 2)

Specification	English	SI	Standard
Tensile Strength	4194-4355 psi @ 68°F 2610-2900 psi @ 176°F	26-30 N/mm2 @ 20°C 18-20 N/mm2 @ 80°C	DIN 53455
IZOD Impact Res.	No Break	No Break	DIN 53453
Temperature Working Range	-40°F to 200°F	-40°C to 93°C	N/A
Roughness	e=0.00028 in	e=0.007 mm	N/A
Max. Short-term Exposure	150 psig @ 210°F (48 hr)	1035 kPa @ 99°C (48 hr)	ASTM F876
UV Rating	15 days	15 days	ASTM F2657

FUNCTIONAL DESCRIPTION

RAUGEO Helix probes are transported in 3 ft 7 in. lengths that expand to 10 ft during installation. Each Helix comes preassembled with a color-coded supply and return tail for easy installation.

RAUGEO pipes are recognized by the International Ground Source Heat Pump Association (IGSHPA) *Design and Installation Standards* guide as an accepted piping system for ground loop heat exchangers.

RAUGEO pipes are manufactured using REHAU's high-pressure peroxide method for crosslinked polyethylene.

RAUGEO PEXa pipe is rated for continuous operation of 160 psi @ 73.4°F (1105 kPa @ 23°C), 100 psi @ 180°F (690 kPa @ 82.2°C). It is compatible with RAUPEX compression-nut fittings and compression-sleeve fittings certified to ASTM F2080 and CSA B137.5, and electro-fusion fittings certified to ASTM F1055.

PRESSURE LOSS TABLE

Head Loss for Fluids at 32°F (0°C)*						
Flow Rate (gpm)	Water		20% Ethylene Glycol		20% Propylene Glycol	
	(ft head loss per 100 ft of pipe)	(ft head loss per Helix)	(ft head loss per 100 ft of pipe)	(ft head loss per Helix)	(ft head loss per 100 ft of pipe)	(ft head loss per Helix)
1.0	0.19	0.25	0.33	0.43	0.44	0.58
1.2	0.41	0.54	0.39	0.51	0.53	0.69
1.4	0.53	0.69	0.46	0.60	0.61	0.80
1.6	0.67	0.88	0.52	0.68	0.70	0.92
1.8	0.81	1.06	0.59	0.77	0.79	1.03
2.0	0.97	1.27	1.17	1.53	0.88	1.15
2.2	1.15	1.51	1.38	1.81	0.96	1.26
2.4	1.33	1.74	1.59	2.08	1.05	1.38
2.6	1.53	2.00	1.83	2.40	1.99	2.61
2.8	1.74	2.28	2.07	2.71	2.25	2.95
3.0	1.96	2.57	2.33	3.05	2.53	3.31
3.2	2.19	2.87	2.60	3.41	2.82	3.69
3.4	2.43	3.18	2.89	3.79	3.13	4.10
3.6	2.68	3.51	3.18	4.17	3.45	4.52
3.8	2.95	3.86	3.49	4.57	3.78	4.95
4.0	3.22	4.22	3.81	4.99	4.12	5.40
4.2	3.5	4.59	4.15	5.44	4.48	5.87
4.4	3.8	4.98	4.49	5.88	4.85	6.86
4.6	4.11	5.38	4.85	6.35	5.24	6.86
4.8	4.42	5.79	5.22	6.84	5.63	7.38
5.0	4.75	6.22	5.60	7.34	6.04	7.91

* IGSHPA guidelines recommend a pressure loss less than 4 ft of head per 100 ft of pipe

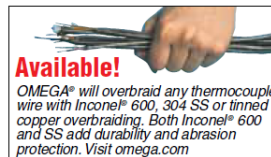
Figure 61. Specification sheet of helical ground heat exchanger (2 of 2)

Thermocouple Wire

Duplex Extension Grade

ANSI
color
code
shown

To order
IEC color
code visit
omega.com



JX Extension Grade
Iron/Constantan
Duplex
ANSI Type JX



ANSI Color Code: Positive Wire, White; Negative Wire, Red; Overall, Black

To Order visit omega.com/exgg_jx_wire for Pricing and Details									
Insulation	AWG No.	Model Number	Type Wire	Insulation		Max Temp*		Nominal Size mm (inch)	Wt [†] . kg/300 m (lb/1000')
				Conductor	Overall	°C	°F		
Glass	16	EXGG-J-16	Solid	Glass braid	Glass braid	200	392	2.2 x 3.3 (0.085 x 0.130)	9 (18)
PFA	16	EXTT-J-16	Solid	PFA	PFA	200	392	2.2 x 3.9 (0.086 x 0.152)	8 (16)
Neoflon	20	EXTT-J-20	Solid	PFA	PFA	200	392	1.7 x 2.9 (0.068 x 0.116)	4 (9)
PFA	24	EXTT-J-24	Solid	PFA	PFA	200	392	1.4 x 2.3 (0.056 x 0.092)	3 (8)
Polyvinyl	14	EXPP-J-14	Solid	Polyvinyl	Polyvinyl	105	221	3.4 x 7.3 (0.134 x 0.288)	13 (29)
	16	EXPP-J-16	Solid	Polyvinyl	Polyvinyl	105	221	3.1 x 5.3 (0.123 x 0.207)	10 (22)
	16	EXPP-J-16S	7 x 24	Polyvinyl	Polyvinyl	105	221	5.5 x 5.7 (0.139 x 0.226)	10 (22)
	20	EXPP-J-20	Solid	Polyvinyl	Polyvinyl	105	221	2.4 x 3.8 (0.095 x 0.151)	6 (12)
	20	EXPP-J-20S	7 X 28	Polyvinyl	Polyvinyl	105	221	3.1 x 4.4 (0.123 x 0.175)	6 (12)
	24	EXPP-J-24S	7 x 32	Polyvinyl	Polyvinyl	105	221	2.1 x 3.4 (0.082 x .0134)	4 (9)
PFA	16	EXFF-J-16	Solid	FEP	FEP	200	392	2.2 x 3.9 (0.086 x 0.152)	8 (16)
Neoflon	20	EXFF-J-20	Solid	FEP	FEP	200	392	1.7 x 2.9 (0.068 x 0.116)	4 (9)
FEP	24	EXFF-J-24	Solid	FEP	FEP	200	392	1.4 x 2.3 (0.056 x 0.092)	3 (6)

* Maximum temperature is for extension-grade wire or insulation, whichever is lower.

† Weight of spool and wire rounded to the next highest kg/lb (does not include packing material).

‡ To order special limits of error wire, add "-SLE" to model number before spool length.

Ordering Example: EXTT-J-20-25, 25' spool of extension-grade thermocouple wire, Type J, 20 gage, PFA over PFA insulation.

TX Extension Grade
Copper/Constantan
Duplex
ANSI Type TX



ANSI Color Code: Positive Wire, Blue; Negative Wire, Red; Overall, Blue

To Order <i>visit omega.com/exgg_jx_wire for Pricing and Details</i>									
Insulation	AWG No.	Model Number	Type Wire	Insulation		Max Temp*		Nominal Size mm (inch)	Wt [†] . kg/300 m (lb/1000')
				Conductor	Overall	°C	°F		
PFA	16	EXTT-T-16	Solid	PFA	PFA	100	212	2.2 x 3.9 (0.086 x 0.152)	8 (16)
Neoflon	20	EXTT-T-20	Solid	PFA	PFA	100	212	1.7 x 2.9 (0.068 x 0.116)	4 (9)
PFA	24	EXTT-T-24	Solid	PFA	PFA	100	212	1.4 x 2.3 (0.056 x 0.092)	3 (6)
Polyvinyl	14	EXPP-T-14	Solid	Polyvinyl	Polyvinyl	100	212	3.4 x 7.3 (0.134 x 0.288)	13 (29)
	16	EXPP-T-16	Solid	Polyvinyl	Polyvinyl	100	212	3.1 x 5.3 (0.123 x 0.207)	10 (22)
	20	EXPP-T-20	Solid	Polyvinyl	Polyvinyl	100	212	2.4 x 3.8 (0.095 x 0.151)	6 (12)
	20	EXPP-T-20S	7 x 28	Polyvinyl	Polyvinyl	100	212	3.1 x 4.4 (0.123 x 0.075)	6 (12)
	24	EXPP-T-24	Solid	Polyvinyl	Polyvinyl	100	212	2.1 x 3.4 (0.082 x .0134)	2 (4)
	24	EXPP-T-24S	7 x 32	Polyvinyl	Polyvinyl	100	212	2.1 x 3.4 (0.082 x .0134)	4 (9)
PFA	16	EXFF-T-16	Solid	FEP	FEP	100	212	2.2 x 3.9 (0.086 x 0.152)	8 (16)
Neoflon	20	EXFF-T-20	Solid	FEP	FEP	100	212	1.7 x 1.2 (0.068 x 0.116)	4 (9)
FEP	24	EXFF-T-24	Solid	FEP	FEP	100	212	1.4 x 2.3 (0.056 x 0.092)	3 (6)

* Maximum temperature is for extension-grade wire or insulation, whichever is lower.

† Weight of spool and wire, rounded to next highest kg/lb (does not include packing material).

‡ To order special limits of error wire, add "-SLE" to model number before spool length.

Ordering Example: EXTT-T-20-50, 50' spool of extension-grade thermocouple wire, type TX, 20 gage, PFA Neoflon insulation.

Discount Schedule (1000' Spools Only)	
3 to 4 spools	10%
5 to 9 spools	15%
10 to 19 spools	20%

H-34

Figure 62. Specification sheet of type-T thermocouple wire

ANSI and IEC Color Codes[†] for Thermocouples, Wire and Connectors

All OMEGA™ Thermocouple Wire, Probes and Connectors are available with either **ANSI** or **IEC** Color Codes. In this Handbook, model numbers in the To Order tables reflect the **ANSI** Color-Coded Product.

Connectors					Connectors					
ANSI Code	ANSI/ASTM E-230 Color Coding		Alloy Combination		Comments Environment Bare Wire	Maximum T/C Grade Temp. Range	EMF (mV) Over Max. Temp. Range	IEC 584-3 Color Coding		IEC Code
	Thermocouple Grade	Extension Grade	+ Lead	- Lead				Thermocouple Grade	Intrinsically Safe	
J			IRON Fe (magnetic)	CONSTANTAN COPPER-NICKEL Cu-Ni	Reducing, Vacuum, Inert. Limited Use in Oxidizing at High Temperatures. Not Recommended for Low Temperatures.	-210 to 1200°C -346 to 2193°F	-8.095 to 69.553			J
K			CHROMEGLA™ NICKEL-CHROMIUM Ni-Cr	ALOMEGA™ NICKEL-ALUMINUM Ni-Al (magnetic)	Clean Oxidizing and Inert. Limited Use in Vacuum or Reducing. Wide Temperature Range, Most Popular Calibration	-270 to 1372°C -454 to 2501°F	-6.458 to 54.886			K
T			COPPER Cu	CONSTANTAN COPPER-NICKEL Cu-Ni	Mild Oxidizing, Reducing Vacuum or Inert. Good Where Moisture is Present. Low Temperature & Cryogenic Applications	-270 to 400°C -454 to 752°F	-6.258 to 20.872			T
E			CHROMEGLA™ NICKEL-CHROMIUM Ni-Cr	CONSTANTAN COPPER-NICKEL Cu-Ni	Oxidizing or Inert. Limited Use in Vacuum or Reducing. Highest EMF Change Per Degree	-270 to 1000°C -454 to 1832°F	-9.835 to 76.373			E
N			OMEGA-P™ NICHROSIL Ni-Cr-Si	OMEGA-N™ NISIL Ni-Si-Mg	Alternative to Type K. More Stable at High Temps	-270 to 1300°C -450 to 2372°F	-4.345 to 47.513			N
R	NONE ESTABLISHED		PLATINUM-13% RHODIUM Pt-13% Rh	PLATINUM Pt	Oxidizing or Inert. Do Not Insert in Metal Tubes. Beware of Contamination. High Temperature	-50 to 1768°C -58 to 3214°F	-0.226 to 21.101			R
S	NONE ESTABLISHED		PLATINUM-10% RHODIUM Pt-10% Rh	PLATINUM Pt	Oxidizing or Inert. Do Not Insert in Metal Tubes. Beware of Contamination. High Temperature	-50 to 1768°C -58 to 3214°F	-0.236 to 18.693			S
R/SX	NONE ESTABLISHED		COPPER Cu	COPPER-LOW NICKEL Cu-Ni	Extension Grade Connecting Wire for R & S Thermocouples. Also Known as RX & SX Extension Wire.					R/SX
U*	NONE ESTABLISHED		COPPER Cu	COPPER Cu	Uncompensated for use with RTDs and Thermistors					U
B	NONE ESTABLISHED		PLATINUM-30% RHODIUM Pt-30% Rh	PLATINUM-6% RHODIUM Pt-6% Rh	Oxidizing or Inert. Do Not Insert in Metal Tubes. Beware of Contamination. High Temp. Common Use in Glass Industry	0 to 1820°C 32 to 3308°F	0 to 13.820			B
G* (W)	NONE ESTABLISHED		TUNGSTEN W	TUNGSTEN-26% RHENIUM W-26% Re	Vacuum, Inert, Hydrogen. Beware of Embrittlement. Not Practical Below 399°C (750°F). Not for Oxidizing Atmosphere	0 to 2320°C 32 to 4208°F	0 to 38.564	NO STANDARD USE ANSI COLOR CODE		G (W)
C* (W5)	NONE ESTABLISHED		TUNGSTEN-5% RHENIUM W-5% Re	TUNGSTEN-26% RHENIUM W-26% Re	Vacuum, Inert, Hydrogen. Beware of Embrittlement. Not Practical Below 399°C (750°F). Not for Oxidizing Atmosphere	0 to 2320°C 32 to 4208°F	0 to 37.066	NO STANDARD USE ANSI COLOR CODE		C (W5)
D* (W3)	NONE ESTABLISHED		TUNGSTEN-3% RHENIUM W-3% Re	TUNGSTEN-25% RHENIUM W-25% Re	Vacuum, Inert, Hydrogen. Beware of Embrittlement. Not Practical Below 399°C (750°F). Not for Oxidizing Atmosphere	0 to 2320°C 32 to 4208°F	0 to 39.506	NO STANDARD USE ANSI COLOR CODE		D (W3)

* Not official symbol or standard designation

† JIS color code also available.

H-3

Figure 63. ANSI/ASTM E-320 color codes for thermocouple wires

Thermocouple Tolerances (Reference Junction at 0°C)

American Limits of Error ASTM E230-ANSI MC 96.1

ANSI Code		Standard Limits [†]		Special Limits [†]	
J	Temp Range	>0 to 750°C	>32 to 1382°F	0 to 750°C	32 to 1382°F
	Tolerance Value	2.2°C or 0.75%	4.0°F or 0.75%	1.1°C or 0.4%	2.0°F or 0.4%
K	Temp Range	>0 to 1250°C	>32 to 2282°F	0 to 1250°C	32 to 2282°F
	Tolerance Value	2.2°C or 0.75%	4.0°F or 0.75%	1.1°C or 0.4%	2.0°F or 0.4%
	Temp. Range*	-200 to 0°C	-328 to 32°F		
T	Temp Range	>0 to 350°C	>32 to 662°F	0 to 350°C	32 to 662°F
	Tolerance Value	1.0°C or 0.75%	1.8°F or 0.75%	0.5°C or 0.4%	1°F or 0.4%
E	Temp. Range*	-200 to 0°C	-328 to 32°F		
	Tolerance Value	1.0°C or 1.5%	1.8°F or 1.5%		
N	Temp Range	>0 to 900°C	>32 to 1652	0 to 900°C	32 to 1652°F
	Tolerance Value	1.7°C or 0.5%	3°F or 0.5%	1.0°C or 0.4%	1.8°F or 0.4%
	Temp. Range*	-200 to 0°C	-328 to 32°F		
R S	Temp Range	>0 to 1300°C	>32 to 2372°F	0 to 1300°C	32 to 2372°F
	Tolerance Value	2.2°C or 0.75%	4.0°F or 0.75%	1.1°C or 0.4%	2.0°F or 0.4%
B	Temp. Range*	-270 to 0°C	-454 to 32°F		
	Tolerance Value	2.2°C or 2.0%	4.0°F or 2.0%		
G* C* D*	Temp Range	0 to 1450°C	32 to 2642°F	0 to 1450°C	32 to 2642°F
	Tolerance Value	1.5°C or 0.25%	2.7°F or 0.25%	0.6°C or 0.1%	1°F or 0.1%
B	Temp Range	800 to 1700°C	1472 to 3092°F	Not Established	
	Tolerance Value	0.5%	0.5%		
G* C* D*	Temp Range	0 to 2320°C	32 to 4208°F	Not Established	
	Tolerance Value	4.5°C or 1.0%	9°F or 1.0%		

* Not official symbol or standard designation † Whichever value is greater.

Note: Material is normally selected to meet tolerances above 0°C. If thermocouples are needed to meet tolerances below 0°C, the purchaser shall state this as selection of material is usually required.

IEC Tolerance Class EN 60584-2; JIS C 1602

IEC Code		Class 1	Class 2	Class 3 ^{††}
J	Temp Range	-40 to 375°C	-40 to 333°C	Not Established
	Tolerance Value	±1.5°C	±2.5°C	
	Temp. Range	375 to 750°C	333 to 750°C	
K N	Tolerance Value	±0.4% Reading	±0.75% Reading	
	Temp Range	-40 to 375°C	-40 to 333°C	-167 to 40°C
	Tolerance Value	±1.5°C	±2.5°C	±2.5°C
T	Temp. Range	375 to 1000°C	333 to 1200°C	-200 to -167°C
	Tolerance Value	±0.4%	±0.75% Reading	±1.5% Reading
	Temp Range	-40 to 125°C	-40 to 133°C	-67 to 40°C
E	Tolerance Value	±0.5°C	±1°C	±1°C
	Temp. Range	125 to 350°C	133 to 350°C	-200 to -67°C
	Tolerance Value	±0.4% Reading	±0.75% Reading	±1.5% Reading
R S	Temp Range	-40 to 375°C	-40 to 333°C	-167 to 40°C
	Tolerance Value	±1.5°C	±2.5°C	±2.5°C
	Temp. Range	375 to 800°C	333 to 900°C	-200 to -167°C
B	Tolerance Value	±0.4% Reading	±0.75% Reading	±1.5% Reading
	Temp Range	0 to 1100°C	0 to 600°C	600 to 800°C
	Tolerance Value	±1°C	±1.5°C	+4°C
B	Temp. Range	1100 to 1600°C	600 to 1600°C	800 to 1700°C
	Tolerance Value	±[1 + 0.3% x (Rdg-1100)]°C	±0.25% Reading	±0.5% Reading
	Temp. Range	Not Established	600 to 1700°C	800 to 1700°C
	Tolerance Value		±0.25% Reading	±0.5% Reading

†† Material is normally selected to meet tolerances above -40°C. If thermocouples are needed to meet limits of Class 3, as well as those of Class 1 or 2, the purchaser shall state this, as selection of material is usually required.

H-4

Figure 64. Thermocouple tolerances (Reference Junction at 0°C) ASTM E-320 – ANSI MC96.1. American Limits of error

SPECIFICATIONS

Wetted Materials (except tees)

- See Technical Brief for material specifics.

Sensor Sleeve and Hex Adapter for 228BR, 228CB,

- Sleeve: Admiralty Brass, UNS C44300;
Hex Adapter: Valve Bronze, UNS C83600

Sensor Sleeve and Hex Adapter for 228SS

- Series 300 Stainless Steel

Tee for 228B

- Cast Bronze, Class 125 Per ASME B16.15 and
Copper Coupling

Tee for 228SS

- Cast 316 Stainless, Class 150

Tee for 228CB

- Cast Iron, Class 125 Per ASME B16.4

Tee for 250B

- Cast valve bronze, UNS C83600

Temperature Ratings

- Standard Electronics - 221°F (105°C)
- Irrigation Electronics - 150°F (66°C)

Pressure Ratings

	At 100°F	At 221°F
250BR	400 psi	325 psi
228B	200 psi	165 psi
228CB	175 psi	140 psi
220SS	400 psi	325 psi

Recommended Design Flow Range

- 0.5 to 30 ft/sec for 228
- 0.3 to 15 ft/sec for 250B

Accuracy

- $\pm 1.0\%$ of full scale over recommended design
flow range for 228
- $\pm 1.0\%$ of rate for 250

Figure 65. Specification sheet of the flow meter (1 of 2)

250 and 228 Metal Tee Impeller Flow Sensors

Repeatability

- $\pm 0.3\%$ of full scale over recommended design flow range for all except 250B
- $\pm 0.7\%$ of full scale over recommended design flow range for 250B

Linearity

- $\pm 0.2\%$ of full scale over recommended design flow range for all except 250BR
- $\pm 0.7\%$ of full scale over recommended design flow range for 250BR

Transducer Excitation

- Quiescent current 600uA@8.0VDC to 35.0VDC max.
- Quiescent voltage (Vhigh) Supply Voltage -(600uA*Supply impedance)
- ON state (VLow) Max. 1.2VDC@40mA current limit (15 Ω +0.7VDC)

Output Frequency

- 3.2 Hz to 200 Hz for all except 250BR
- 0.8 Hz to 80 Hz for 250BR

Output Pulse Width

- 5 msec $\pm 25\%$
- Square wave between 100Hz to 200Hz limit

Electrical Cable for Standard Sensor Electronics

- 20 feet of 2-conductor 20 AWG shielded U.L. type PTLC wire provided for connection to display or analog transmitter unit. Rated to 105°C. May be extended to a maximum of 2000 feet with similar cable and insulation appropriate for application.

Electrical Cable for IR Sensor Electronics

- 48 inches of U.L. Style 116666 copper solid AWG 18 wire with direct burial insulation. Rated to 105°C.

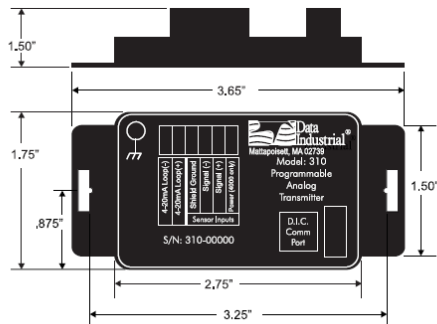
Figure 66. Specification sheet of the flow meter (2 of 2)

Badger® Model 310	Programmable Loop-Powered Analog Transmitter	Technical Brief
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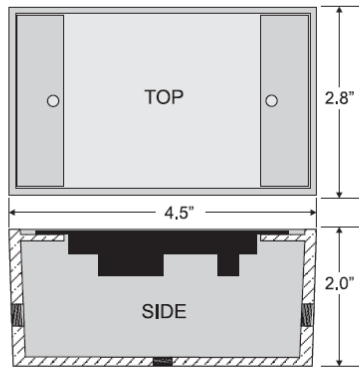
The Badger Model 310 is a loop powered, programmable transmitter capable of converting the signal from Badger Meter flow sensors to a linear 4-20mA analog signal. In addition to our standard square wave signal, it can also accept a sine wave making it a versatile transmitter for numerous applications.

With an onboard microcontroller and digital circuitry, the Badger Model 310 is programmed from a computer eliminating the need to adjust potentiometers to produce precise, accurate and drift free signals. This will save both time and money by lowering overall maintenance times. This model also has an integral filter that the user can specify as 0 (to show true sensor readings) or 10 (for maximum dampening).

The compact cast epoxy body measures 1.75" (44mm) x 2.75" (70mm) x 1" (25mm) and can easily be mounted to panels, DIN rails or enclosures. With multiple inputs, ease of use and a variety of enclosures, the Badger Model 310 is a powerful and competitive transmitter for many of today's demanding applications.



Transmitter Only



Optional Enclosure (Ver. 310-02 and 310-03)

EXAMPLE: 310 - xx			
SERIES			
Programmable Analog Transmitter	310		
OPTIONS			
Transmitter Only			00
W / NEMA 4X Enclosure			01
W / Metal Weathertite Enclosure			02
W / Plastic Weathertite Enclosure			03
W / DIN rail Mounting Clips			04

Badger® Model 310 Ordering Matrix

- SPECIFICATIONS**
- Power Requirements:**
Loop Input Voltage 9-35Vdc
- Input Frequency:**
0.4 to 10 KHz
- Load Resistance**
Max 750Ω@24Vdc
- Output Response Time**
Varies with filter
- Temperature (operating):**
-29°C to 70°C
-20°F to 158°F
- Temperature (storage):**
-40°C to 85°C
-40°F to 185°F
- Accuracy**
± 0.04% of reading over entire span
- Linearity**
0.1% of full scale

Figure 67. Specification sheet of the flow meter's analog transmitter

Last Revised: 2014-11-06 07:14:15.0

NI 9215

±10 V, Simultaneous Analog Input, 100 kS/s, 4 Ch Module



- 4 differential channels, 100 kS/s per channel sample rate
- ±10 V measurement range, 16-bit resolution
- 250 Vrms channel-earth, CAT II (screw terminal), or 60 VDC channel-earth, CAT I (BNC) isolation
- 10-position screw-terminal or BNC connectors available
- -40 °C to 70 °C operating, 5 g vibration, 50 g shock

Overview

The NI 9215 module for use with NI CompactDAQ and CompactRIO chassis includes four simultaneously sampled analog input channels and successive approximation register (SAR) 16-bit analog-to-digital converters (ADCs). The NI 9215 contains NIST-traceable calibration, a channel-to-earth ground double isolation barrier for safety and noise immunity, and high common-mode voltage range.

Recommended Accessories

-NI 9927 strain relief and operator protection (for screw-terminal variant)

Optional Accessories

-NI 9936 extra screw-terminal block (for screw-terminal variant)
-NI 9980 spring terminal block (for screw-terminal variant)
Note: The NI 9980 is not compatible with the NI 9927 and must be used with low- or non-hazardous voltages or installed in a properly rated enclosure.
-BNC male to BNC male cables, 4 pack (2 m; for BNC variant)

Box Contents

-1 NI 9215 C Series module
-1 NI 9215 Operating Instructions and Specifications manual
-1 NI 9936 10-position screw-terminal connector (for screw-terminal variant)

Figure 68. Overview of DAQ’s NI 9215 ±10V analog input module

Accuracy		
Measurement Conditions	Percent of Reading (Gain Error)	Percent of Range (Offset Error) ³
Calibrated max (–40 to 70 °C)	0.2%	0.082%
Calibrated, typ (25 °C, ±5 °C)	0.02%	0.014%
Uncalibrated max (–40 to 70 °C)	1.05%	0.82%
Uncalibrated typ (25 °C, ±5 °C)	0.6%	0.38%

Figure 69. DAQ’s NI 9215 ±10V analog input module accuracy specifications

Last Revised: 2014-11-06 07:14:43.0

NI 9211
±80 mV Thermocouple Input, 14 S/s, 4 Ch Module



- 4 thermocouple or ±80 mV analog inputs
- -40 to 70 °C operating range
- 24-bit resolution; 50/60 Hz noise rejection
- Hot-swappable operation
- NIST-traceable calibration

Overview

The NI 9211 C Series thermocouple input module for use with NI CompactDAQ and CompactRIO chassis includes a 24-bit delta-sigma ADC, antialiasing filters, open-thermocouple detection, and cold-junction compensation for high-accuracy thermocouple measurements. It contains NIST-traceable calibration and a channel-to-earth ground double-isolation barrier for safety, noise immunity, and high common-mode voltage range.

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Requirements and Compatibility

OS Information

- Real-Time OS
- Windows

Driver Information

- NI-DAQmx
- NI-RIO

Software Compatibility

- LabVIEW
- LabWindows/CVI
- Measurement Studio
- SignalExpress
- Visual Basic
- Visual C#
- Visual C++
- Visual Studio
- Visual Studio .NET

Figure 70. Overview of DAQ's NI 9212 ±80mV thermocouple input module

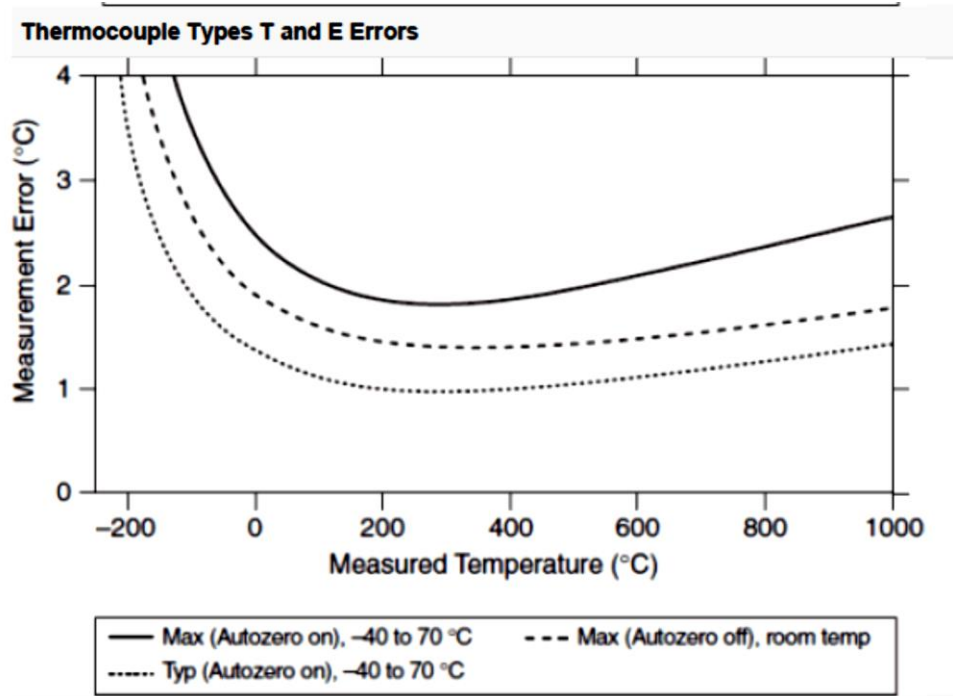


Figure 71. DAQ's NI 9212 $\pm 80\text{mV}$ thermocouple input module error specification

APPENDIX B

UNCERTAINTY ANALYSIS

Experimental Uncertainty

In uncertainty analysis, two basic categories of errors can be found:

- Systematic errors
- Random errors

This uncertainty analysis is based on the *root of the sum of the squares* (RSS) method to compute the propagation of uncertainties (Wheeler & Ganji, 2010).

The method states that for a result R that is a function of n measured variables x_i , the uncertainty in the result w_R is, Equation (39):

$$w_R = \sqrt{\sum_{i=1}^n \left(w_{x_i} \frac{\partial R}{\partial x_i} \right)^2} \quad (39)$$

The measured variables need to have their own uncertainties already determined.

The methodology was applied to the computation of the performance of the GHE (q) in order to achieve a result in the form:

$$q = A \pm w_q \quad (40)$$

where q is the rate of heat transfer, according to Equations (4) and (5), ($q = \dot{m} \cdot c_p \cdot \Delta T$), with $\dot{m} = \dot{Q} \cdot \rho$, of the Performance Calculations section, A is the value of the calculated result and w_q is the estimated uncertainty. For this calculation, it will be assumed that c_p and ρ are constants. Then the heat rate q takes the form:

$$q = C \cdot \dot{Q} \cdot \Delta T \quad (41)$$

where C is a constant.

The first individual measured variable is:

$$\dot{Q} = A_{\dot{Q}} \pm w_{\dot{Q}} \quad (42)$$

where $A_{\dot{Q}}$ is the measured value for flow rate and $w_{\dot{Q}}$ is the uncertainty in the measurement of flow rate. In the process of measuring the flow rate, three components contribute to the total uncertainty: measurement errors from the sensing element, errors from the transmitter and errors from the DAQ equipment (see Experimental Apparatus section). The manufacturers of each of these components state their accuracy specifications in their respective specification sheets (see APPENDIX A), and they are summarized in Table 12.

Table 12. Chain of errors in the flow rate measurement system

Element (Model)	Type of error	% Value	Actual value
Flow sensor (Badger Meter® Data Industrial 228BR2005)	Accuracy	± 1.0% of full scale	± 2.9 gpm
	Repeatability	± 0.3% of full scale	± 0.9 gpm
	Linearity	± 0.2% of full scale	± 0.6 gpm
Analog transmitter (Badger Meter® Model 310)	Accuracy	± 0.04% of reading	± 0.005 gpm
	Linearity	± 0.1% of full scale	± 0.05 gpm
±10V analog input (NI® 9215)	Gain error	0.2% of reading	± 2.4 gpm
	Offset error	0.082% of range	± 0.041 gpm

To combine all the errors in Table 12, some assumptions are made. First, for the flow sensor, it is assumed that the accuracy error contains the other two (repeatability and linearity). 1% of full scale as stated in the equipment's specification sheet is equal to 2.9 gpm (0.18 l/s). For the flow rate measured during Cooling Test II, 12 gpm (0.76 l/s), this error corresponds to 24.2% of reading. Secondly, the transmitter accuracy error is 0.04% of reading, or 0.005 gpm ($3.2 \cdot 10^{-4}$ l/s) for 12 gpm (0.76 l/s). The linearity error is 0.1% of full scale, or 0.05 gpm (0.0032 l/s) of 50 gpm (3.15 l/s), equivalent to 0.42% of reading in

this case. Thus, the total uncertainty in transmitter measurement is calculated by the root of the sum of the squared values as follows:

$$w_{Transmitter} = \sqrt{0.42^2 + 0.04^2} = 0.42\% \quad (43)$$

Finally, the DAQ system module contributes with a 0.2% of reading and 0.082% of range gain and offset errors respectively. The offset error is equivalent to 0.34% of a reading of 12 gpm (0.76 l/s). Combining the two types of error using the root of the sum of the squares, the total error of the DAQ is:

$$w_{DAQ} = \sqrt{0.2^2 + 0.34^2} = 0.39\% \quad (44)$$

Once all the contributors are determined, the total uncertainty in flow rate measurement ($w_{\dot{Q}}$) is calculated by applying once more the root of the sum of the squared values:

$$w_{\dot{Q}} = \sqrt{24.2^2 + 0.42^2 + 0.39^2} = 24.2\% \quad (45)$$

For the particular case of Cooling Test II, the measurement of flow rate is represented, to a confidence level of 95%, by:

$$\dot{Q} = 12 \pm 2.9 \text{ gpm} (0.76 \pm 0.18 \text{ l/s}) \quad (46)$$

This error estimation is probably too big; it is not realistic. The problem is that the sensors installed in the geothermal loops are working on the lowest part of their range.

An alternative statistical approach was taken instead for comparison. From a sufficiently large sample of actually acquired data for the flow rate, the average and standard deviation was calculated. For the specific case of Cooling Test II, the average (mean) flow was found to be 12 gpm (0.76 l/s), with a standard deviation of 0.22 gpm (0.014 l/s). Since the sensors are calibrated, it can be assumed that the errors are only random errors,

and the scatter of the measured data over the mean can be considered to follow a normal distribution. For a confidence level of 95%, the interval of a measurement is approximately the mean ± 2 standard deviations. In this case, the uncertainty in sensor reading is $0.22 \times 2 = 0.44$ gpm (0.028 l/s), which corresponds to a 3.7% error of reading. According to this approach, the measurement of flow rate would be represented, to a confidence level of 95%, by:

$$\dot{Q} = 12 \pm 0.4 \text{ gpm } (0.76 \pm 0.03 \text{ l/s}) \quad (47)$$

The second individual measured variable is the temperature difference, as in Equation (48):

$$\Delta T = A_{\Delta T} \pm w_{\Delta T} \quad (48)$$

where $A_{\Delta T}$ is the measured value for temperature difference and $w_{\Delta T}$ is the uncertainty in the measurement of temperature difference. In the process of measuring the temperature, only two subsystems are in place: the thermocouple wire and the DAQ system. Table 13 summarizes the error originated in each of these subsystems.

Table 13. Chain of errors in the temperature measurement system

Element (Model)	Type of error	Value
Thermocouple wire (Omega [®] extension grade type-T)	Special limits of error	greater of 1°F (0.6°C) or 0.4%
± 80 mV Thermocouple Input (NI [®] 9211)	Maximum error at room temperature ¹	3.1°F (1.7°C)

¹ Room Temperature = 59°F (15°C) to 95°F (35°C). Error includes gain errors, offset errors, differential and integral nonlinearity, quantization errors, noise errors, and isothermal errors (see APPENDIX A. Equipment Specifications)

A root of the sum of the squares over these two errors gives an overall error for the temperature measurement system of 3.3°F (1.8°C). For the very narrow range of

temperature differences occurring in these thesis' experiments (sometimes less than 1°F [0.6°C]), it would not have been advised to measure temperature with this type of sensor, because the error in measurement is greater than the temperature difference itself. However, an observation of actual measurements acquired by the DAQ shows a very stable behavior of the measurements, suggesting that the accuracy of the sensors might have been higher. As with the case of the flow rate measurement, a statistical approach was conducted.

During the last week of Cooling Test II, the average ΔT (mean) was 0.89°F (0.49°C) with a standard deviation of 0.1°F (0.06°C). Similarly, as in the previous example, the normal distribution can be assumed, and the uncertainty level set to double the standard deviation. Consequently, to a level of confidence of 95%, the GHE loop fluid temperature difference can be represented in Equation (49). The error thus calculated is equivalent to a 22.5% of measurement.

$$\Delta T = 0.89 \pm 0.2 \text{ } ^\circ F (0.5 \pm 0.1 \text{ } ^\circ C) \quad (49)$$

Table 14 summarizes the uncertainty levels found thus far for the individual measured variables.

Table 14. Summary of uncertainty level for individual measurements affecting GHE performance calculation

Measured variable	Uncertainty (%)	Confidence level (%)
Flow rate	± 3.7	95
Temperature	± 22.5	95

Now the RSS method can be completed. Equation (39) takes the form:

$$w_q = \sqrt{\left(w_{\dot{Q}} \frac{\partial q}{\partial \dot{Q}}\right)^2 + \left(w_{\Delta T} \frac{\partial q}{\partial \Delta T}\right)^2} \quad (50)$$

The partial derivatives are:

$$\begin{aligned} \frac{\partial q}{\partial \dot{Q}} &= C \cdot \Delta T = C \cdot 0.89^\circ F \text{ (} 0.49^\circ C \text{)} \\ \frac{\partial q}{\partial \Delta T} &= C \cdot \dot{Q} = C \cdot 12 \text{ gpm (} 0.76 \text{ l/s)} \end{aligned} \quad (51)$$

and

$$\begin{aligned} w_q &= \sqrt{(w_{\dot{Q}} C \Delta T)^2 + (w_{\Delta T} C \dot{Q})^2} \\ &= C \cdot \sqrt{(0.44 \cdot 0.89)^2 + (0.2 \cdot 12)^2} \\ &= C \cdot 2.43 \frac{\text{gal} \cdot ^\circ F}{\text{min}} \end{aligned} \quad (52)$$

In the example (Cooling Test II):

$$\begin{aligned} C &= 0.1337 \frac{\text{ft}^3}{\text{gal}} \cdot 63.92 \frac{\text{lb}_m}{\text{ft}^3} \cdot 60 \frac{\text{min}}{\text{hr}} \cdot 0.964 \frac{\text{Btu}}{\text{lb}_m \cdot ^\circ F} \\ &= 494.3 \frac{\text{min} \cdot \text{Btu}}{\text{gal} \cdot \text{hr} \cdot ^\circ F} \end{aligned} \quad (53)$$

Finally, substituting Equation (53) into (52) yields

$$w_{q(\text{three helix})} = 494.3 \cdot 2.43 = 1201 \text{ Btu/hr (} 352 \text{ W)} \quad (54)$$

As this was calculated for the set of three helical GHEs, the unit error per helix is:

$$w_q = \frac{1201}{3} = 400 \text{ Btu/hr (} 117 \text{ W)} \quad (55)$$

Performance calculation of GHE for Cooling Test II yielded 1760 Btu/(hr·helix) (516 W/helix) (A in Equation (40)); therefore, the global uncertainty for the performance calculation of helical GHE under the conditions of Cooling Test II is represented by:

$$q = 1760 \pm 400 \text{ Btu/hr} \quad (516 \pm 117 \text{ W}) \quad (56)$$

This represents an error of roughly 23%, which is not acceptable for industry standards.

See DISCUSSION section.

APPENDIX C

ADDITIONAL EXPERIMENTS

In this Appendix, two additional experiment and simulation results are presented:

Cooling Test III and Heating Test III.

Cooling Test III

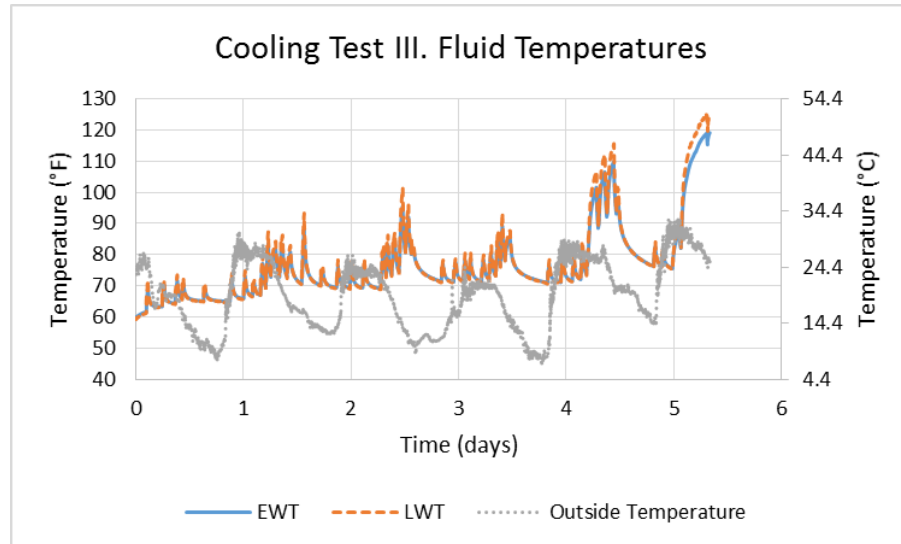


Figure 72. Cooling Test III. Geothermal fluid temperatures and outdoor air temperature

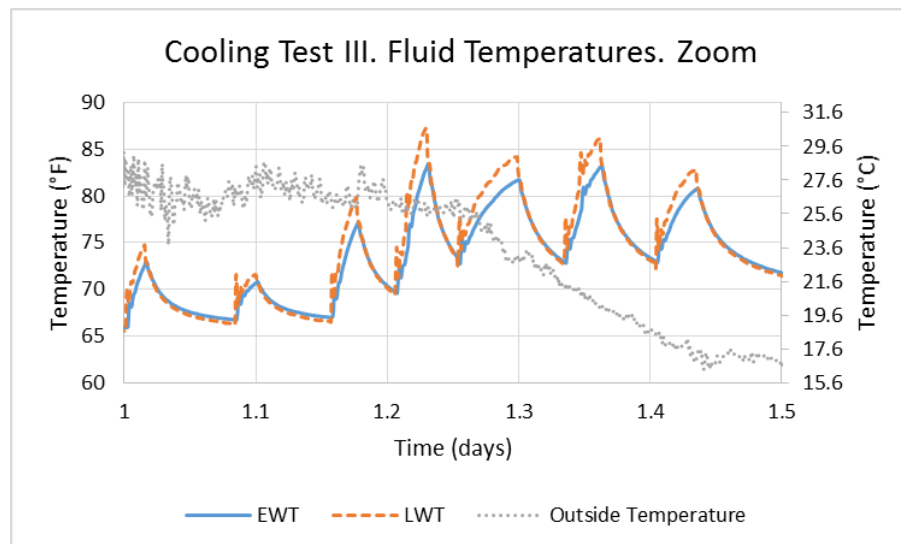


Figure 73. Cooling Test III. Geothermal fluid temperatures and outdoor air temperature. Zoom of the first half of the second day

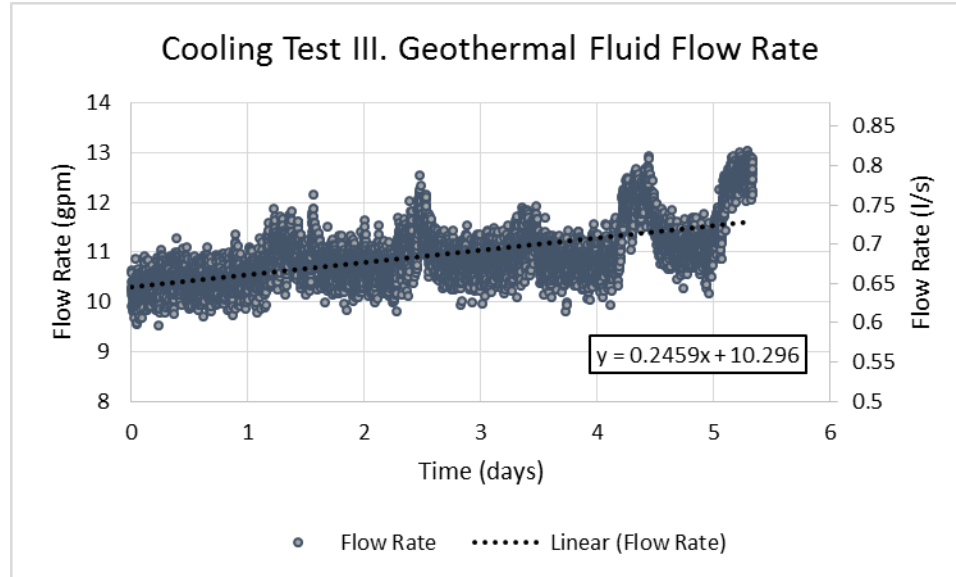


Figure 74. Cooling Test III. Volumetric flow rate. The graphic shows a linear fit to the data and its equation

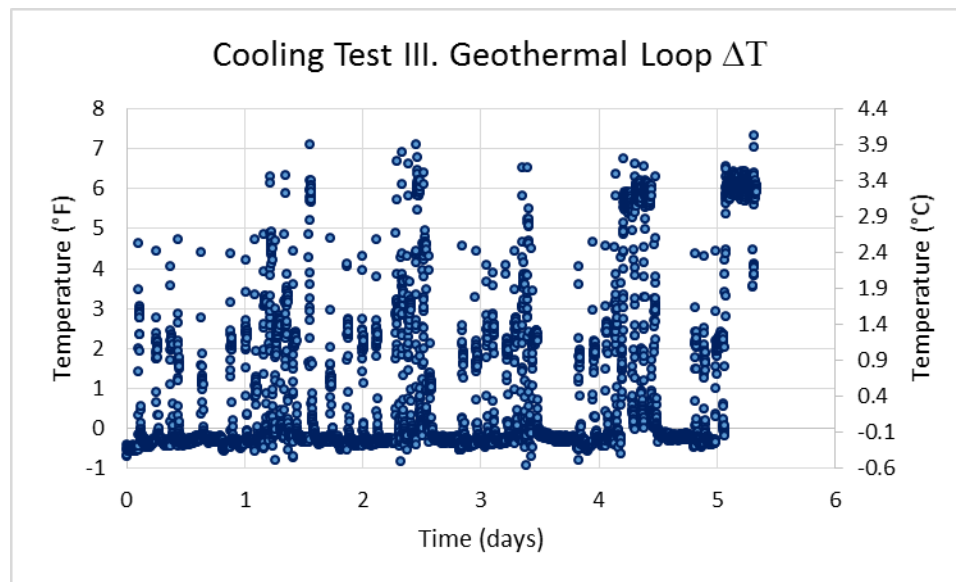


Figure 75. Cooling Test III. Geothermal fluid temperature difference (ΔT)

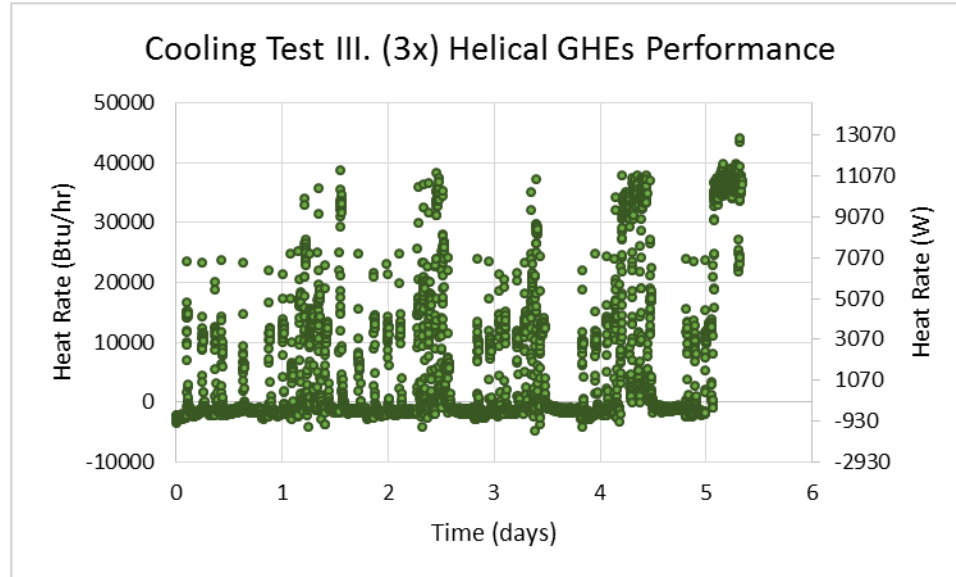


Figure 76. Cooling Test III. Performance analysis of the three helical GHEs at the RMEH

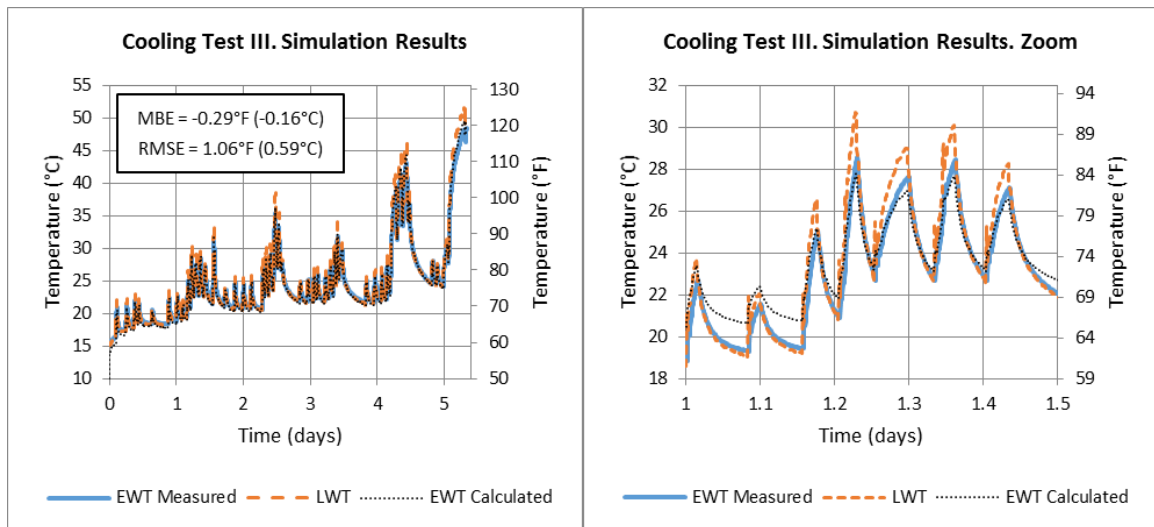


Figure 77. Comparison between measured and simulated EWT during Cooling Test III. Left: whole test; right: zoom of the first half of the 2nd day of experiment. Mean bias error (MBE) and root mean squared error (RMSE) are stated in the box

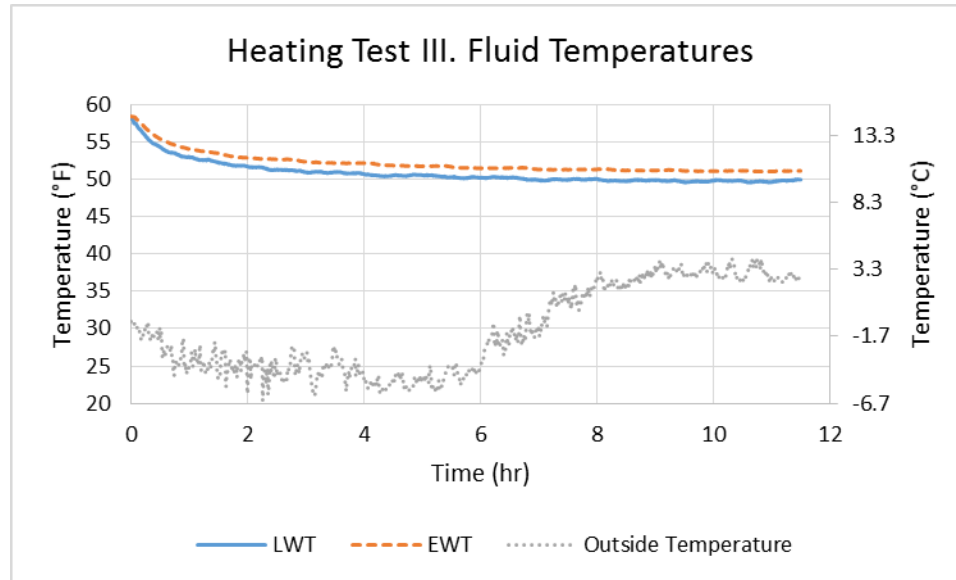
Heating Test III

Figure 78. Heating Test III. Geothermal fluid temperatures and outdoor air temperature

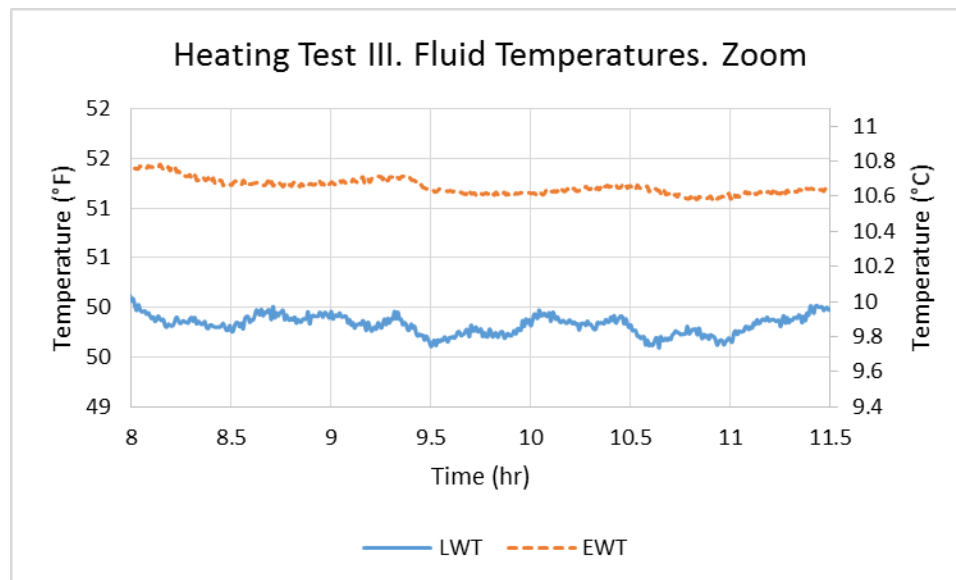


Figure 79. Heating Test III. Geothermal fluid temperatures. Zoom of the last 3.5 hr

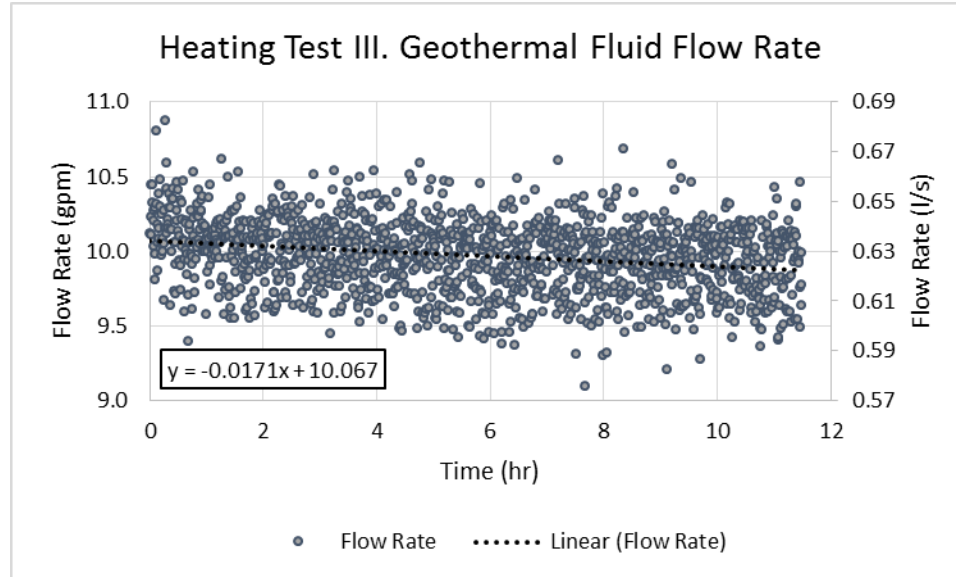


Figure 80. Heating Test III. Volumetric flow rate. The graphic shows a linear fit to the data and its equation

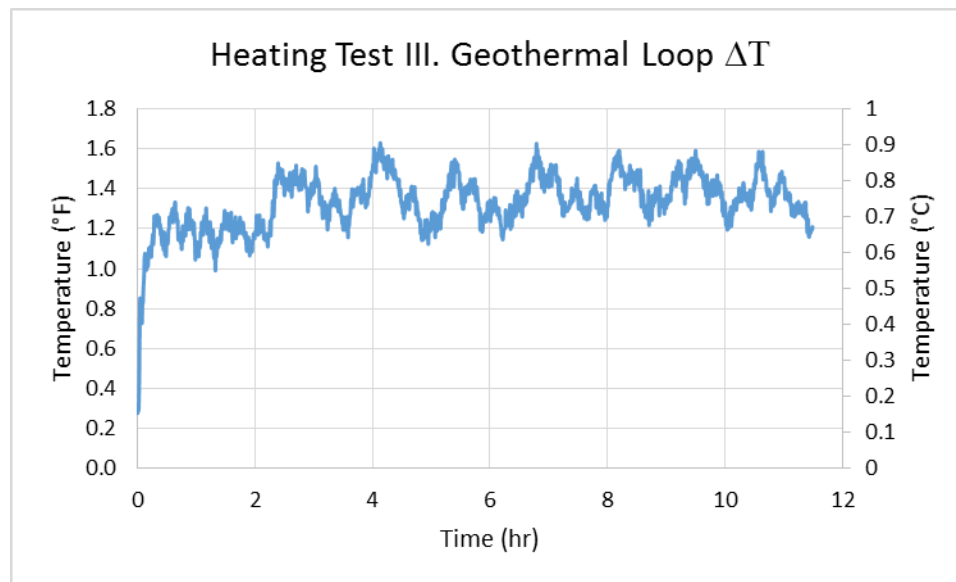


Figure 81. Heating Test III. Geothermal fluid temperature difference (ΔT)

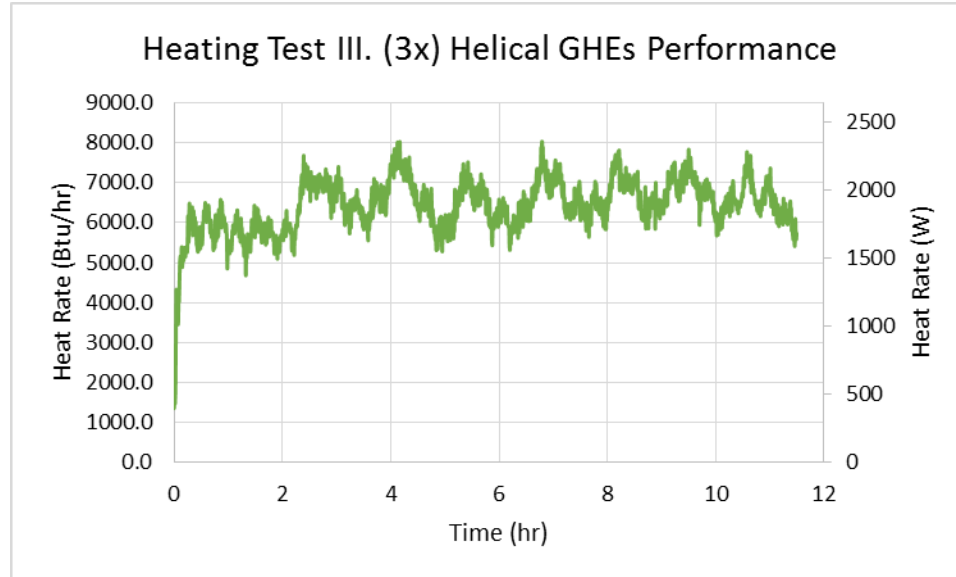


Figure 82. Heating Test III. Performance analysis of the three helical GHEs at the RMEH

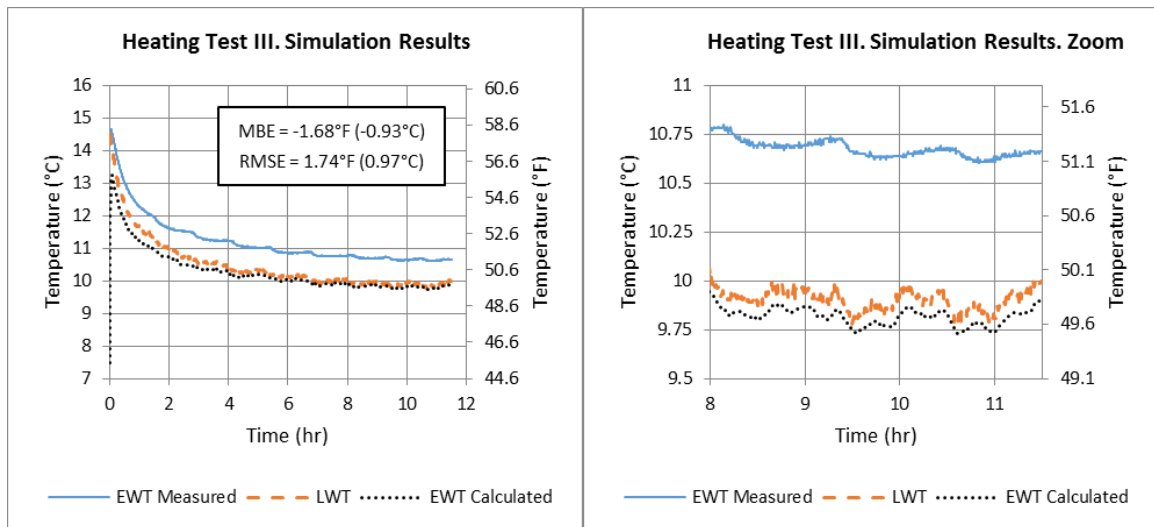


Figure 83. Comparison between measured and simulated EWT during Heating Test III. Left: whole test; right: zoom of the last 3.5 hr of experiment. Mean bias error (MBE) and root mean squared error (RMSE) are stated in the box

APPENDIX D

SOIL TEMPERATURE VALIDATION

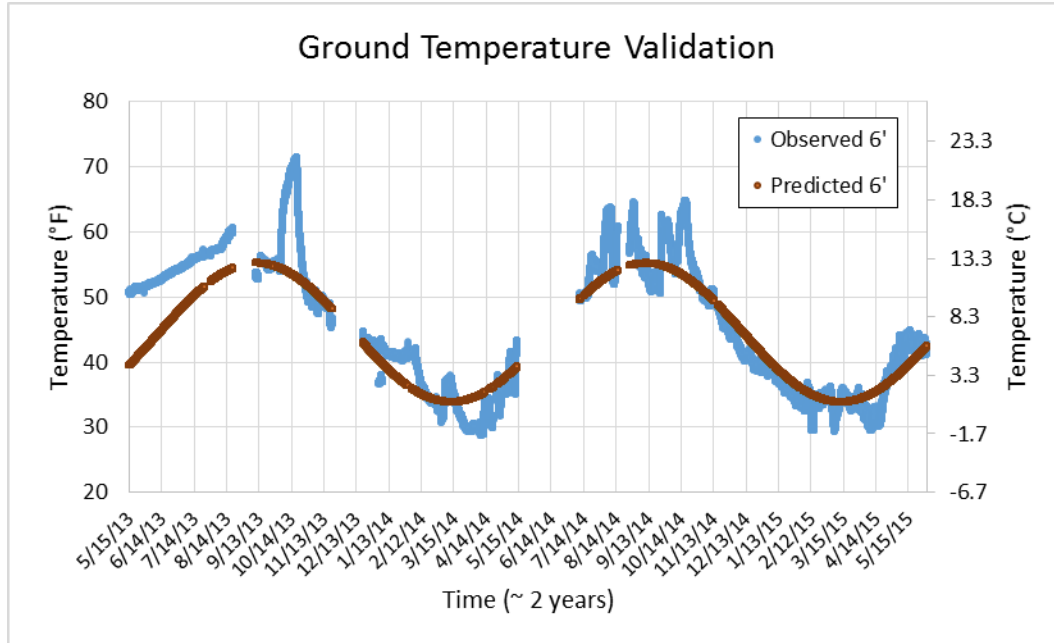


Figure 84. Ground temperature validation. Comparison of predicted soil temperatures at 6 ft of depth by the Kusuda and Achenbach relation (Kusuda & Achenbach, 1965) and observed values as recorded by the DAQ system in place at the RMEH

APPENDIX E

SOLAR AND CLIMATE DATA

Table 15. Monthly normals for the station at Montana State University – Bozeman, Montana (NOAA, 2015)

Month	Monthly Normal (x10) (°F) (NOAA)	Average Temp (°F)	Average Temp (°C)
Jan	251	25.1	-3.8
Feb	277	27.7	-2.4
Mar	355	35.5	1.9
Apr	434	43.4	6.3
May	517	51.7	10.9
Jun	593	59.3	15.2
Jul	671	67.1	19.5
Aug	661	66.1	18.9
Sep	566	56.6	13.7
Oct	456	45.6	7.6
Nov	325	32.5	0.3
Dec	234	23.4	-4.8
Average		44.5	6.9
Max		67.1	19.5
Min		23.4	-4.8
Amplitude		43.7	24.3
Semi-amplitude		21.9	12.1

Table 16. Average Hourly Statistics for Global Horizontal Solar Radiation Wh/m² from Energy Plus[®] (EnergyPlus, 2015) used as solar input data for CaRM model

[illegible]