

ISTANBUL TECHNICAL UNIVERSITY ★ GRADUATE SCHOOL OF SCIENCE
ENGINEERING AND TECHNOLOGY

**THE USE OF GEOTHERMAL HEAT EXCHANGER PILES FOR
SUSTAINABLE DESIGN**

Ph.D. THESIS

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Department of Civil Engineering

Soil Mechanics and Geotechnical Engineering Program

SEPTEMBER 2015

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SEPTEMBER 2015

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**SÜRDÜRÜLEBİLİR TASARIM İÇİN JEOTERMAL ENERJİ
KAZIKLARININ KULLANIMI**

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To my beloved wife and my parents,

FOREWORD

The use of heat exchanger piles by utilizing shallow geothermal energy has recently gained increasing interest due to sustainability aspects and reduced installation costs by offsetting additional cost of borehole drilling. This innovative technology that combines geothermal heat exchange and structural foundation support needs special attention in a geotechnical point of view. However, to this date, little knowledge is available on the thermo-mechanical behavior and long-term performance of heat exchanger piles. This research was initiated in response to the limited understanding of thermo-mechanical behavior of heat exchanger piles and was intended to develop efficient analysis methods to shed light on the topic.

I dedicate all my work to my beloved wife, Nazlım Çalışkan Özüdoğru, for putting up with me during my long years of research. She was always there for me with endless love and support, and she has done above and beyond what I imagined assuring a calm environment for me to focus and work. All my gratitude goes to my beloved parents for their unconditional love.

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ABBREVIATIONS

1D	: One dimensional
2D	: Two dimensional
3D	: Three dimensional
APG	: Auger pressure grouted
APGE	: Auger pressure grouted energy
ASHP	: Air-source heat pump
ASHRAE	: American Society of Heating, Refrigerating and Air-Conditioning Engineers
ASTM	: American Society for Testing and Materials
ATES	: Aquifer thermal energy storage
BHE	: Borehole heat exchanger
BTES	: Borehole thermal energy storage
CHS	: Cylindrical heat source
COP	: Coefficient of performance
DST	: Duct storage
EER	: Energy efficiency ratio
EPFL	: École Polytechnique Fédérale de Lausanne
FD	: Finite difference
FE	: Finite element
FHWA	: Federal Highway Administration
FLS	: Finite line source
GCHP	: Ground-coupled heat pump
GHE	: Geothermal heat exchanger
GSHP	: Ground-source heat pump
GWHP	: Ground-water heat pump
HDPE	: High density polyethylene
ILS	: Infinite line source
PEX	: Cross-linked polyethylene
SWHP	: Surface water heat pump
TCT	: Thermal conductivity test
TRT	: Thermal response test
VT	: Virginia Tech – Virginia Polytechnic Institute and State University

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THE USE OF GEOTHERMAL HEAT EXCHANGER PILES FOR SUSTAINABLE DESIGN

SUMMARY

There is a developing trend around the world to explore alternative energy sources. The main driving forces are growing global energy demand, depleting natural resources and the adverse effects of greenhouse gas emissions from fossil fuel consumption. Geothermal energy is one of the promising renewable sources that can be utilized to offset such trends. Ground temperature remains constant after a depth of about 6 to 10 m as the near surface soils act as a thermal insulator as the ambient temperatures vary seasonally. In most regions this constant temperature is about 10 to 24°C. The relatively constant temperature and the heat storage capacity of near-surface soils in any region represent a significant potential for stored thermal energy which in turn can be harvested with ground-source heat pump (GSHP) systems. These systems exemplify a highly efficient renewable energy technology for space heating and cooling.

Over the past 20 years, the ground coupling concept has been expanded from geothermal borehole systems to the use of building foundation elements as heat exchangers. Heat exchanger pile (also mentioned as energy pile) in particular is an innovative technology that combines geothermal heat exchange and structural foundation support. In this hybrid system, geothermal circulation loops are integrated into the deep foundation elements, such as piles, piers, or drilled shafts that are already designed to provide structural support. The heat energy is fed into the ground for cooling in the summer and withdrawn from the ground for heating in the winter. The temperature differential between the ground and the outside temperature acts as an energy pathway to harvest stored ground energy for bringing the building temperature to comfort zone levels. As an added benefit, the additional cost of geothermal borehole drilling for loop placement is offset by this combined use and installation costs of geothermal heat exchangers are significantly reduced.

Thermal operation of a heat exchanger pile induces temperature changes in the piles. Such temperature changes create thermal stresses and displacements along the pile that, if ignored in the design process, can result on overstressing the piles and/or unacceptable building deformations. Numerous numerical and analytical approaches were developed to approximate the thermally induced stresses and displacements in heat exchanger piles. All the techniques for analyzing the thermal effects on the mechanical behavior of the piles rely on accurate estimates of the temperature changes in the pile.

Sustainable operation of heat exchanger piles or any other thermo-active foundation element relies on maintaining the constant temperature of the ground. Ground temperature changes can directly affect the heat exchange capacity and efficiency of operations. It is critical to evaluate how the heat exchanger pile operation over the life cycle of the structure will change the temperature gradients around the pile. This is

particularly important at regions where the respective heating and cooling energy demands are not balanced.

The objectives of this dissertation can be listed as follows: (1) estimating the thermal properties of the ground accurately, (2) assessing the thermo-mechanical behavior of heat exchanger piles, and (3) predicting the long-term performance of heat exchanger piles and pile groups. The pile subjected to heating and cooling cycles will eventually expand and/or contract. It is thought that the volume changes and the constraints of the pile, i.e., the degree of fixity at the pile head and the pile toe will have an impact on the thermo-mechanical behavior. The amount of slip displacements at the pile-soil interface and the magnitudes of changes in pile axial stress, shear resistance and radial pressures on the pile soil-interface are the main points of concern. The efficiency and sustainability of the system will be investigated for long-term seasonal operation using actual energy demands for locations with extreme climatic conditions, i.e., hot, moderate and cold. Single pile and pile groups with numerous grid sizes will be investigated. Long-term thermo-mechanical behavior of single piles in each location will be assessed.

In order to meet the above mentioned research objectives, in the scope of this study, robust and competent finite element analysis tools will be developed for modeling geothermal heat exchangers. The numerical tools with different levels of complexity will be utilized to analyze various problems related to the operation of GHEs. These include modeling of field thermal conductivity testing, full scale thermo-mechanical testing of heat exchanger piles and long-term seasonal operation of heat exchanger piles and pile groups. The models are planned to be built extensible in nature so that modifications for considering various aspects such as the geothermal gradient or ambient temperature changes could be easily incorporated. Validation and calibration of the proposed numerical models are to be performed using available analytical methods and field testing data.

The dissertation includes literature review on the theory of heat transfer with the academic background needed to study the thermal processes occurring in the geothermal heat exchangers. The concept of ground-source heating/cooling, types of commonly used geothermal heat exchangers, detailed discussion on field thermal conductivity testing and currently available analytical and numerical analysis methods are presented. Three numerical modeling techniques are developed for simulating the thermal processes in GHEs by utilizing finite elements. The validation of the models is performed via an analytical method, and a detailed comparison of the analysis results is presented. Several methods to improve the modeling and analysis of the field TCT are also discussed. Case studies of field TCTs performed in two different sites are analyzed. The concept of thermo-mechanical behavior of heat exchanger piles subjected to temperature changes is explained. Thermo-mechanical numerical models are developed and utilized to assess the change in the stresses in the pile, contact pressures and the behavior of the pile-soil interface by performing parametric analyses with respect to soil strength and pile fixity. The proposed thermo-mechanical numerical model is utilized to study the long-term performance of the heat exchanger piles. This study includes analyses in three different cities, which have hot, moderate and cold climates. Finally, the conclusions of the research is presented along with the discussion of recommendations for future research.

SÜRDÜRÜLEBİLİR TASARIM İÇİN JEOTERMAL ENERJİ KAZIKLARININ KULLANIMI

ÖZET

Dünyada hızla artan nüfus ve refah seviyesi enerji tüketimini yoğun olarak arttırmıştır. Enerjiye olan talep her geçen gün artarken, bu yoğun kullanım sebebiyle bu kısıtlı kaynağın maliyeti de artmaktadır. Ayrıca, mevcut enerji kaynakları tükenme tehlikesi altında olduğundan yakın gelecekte enerji tedarikinde sorunlar oluşacağı öngörülmektedir. Fosil-temelli enerji kaynaklarının kullanımı sonucu çevre kirliliği oluşmakta ve atmosfere salınan karbondioksit (CO₂) ve diğer sera gazları küresel ısınmayı tetiklemektedir.

Fosil-temelli enerji kaynaklarının kullanımını asgari düzeye indirebilmek için temiz, yenilenebilir ve verimli alternatif enerji kaynakları arayışı sürmektedir. İlk aşamada özellikle ulaşım ve ısıtma-soğutma sistemlerinin temiz ve yenilenebilir enerjiye dayalı olarak tasarlanması öncelikli ve acil konular olarak öne çıkmaktadır. Yenilenebilir temiz enerji kaynağı olarak başta güneş enerjisi olmak üzere; rüzgâr, su, jeotermal enerji ve enerji depolama sistemleri gibi çeşitli olanakların araştırıldığı ve geliştirildiği bu süreçte yeryüzü içinde mevcut bulunan jeotermal enerjiden daha geniş ölçekte yararlanılması gerekmektedir.

Klasik ve bilinen tanımıyla jeotermal enerji, yer içindeki doğal ısı enerjisinin ulaşılabilir derinliklere çıktığı noktalarda sıvı ya da gaz ile taşınarak kullanılabilen bir enerji türüdür. Buna ek olarak, yer yüzeyinin hemen altındaki kesimler hava sıcaklığından etkilenirken, belirli bir derinliğin altındaki seviyeler mevsimsel değişimlerden en az düzeyde etkilenmektedir. Örnek olarak İstanbul şehri ele alındığında, yaklaşık olarak 5 metre derinliğin altında bu etki en aza inerken, 10 metreden sonra yer sıcaklığı neredeyse değişmemektedir. Yer sıcaklığı benzer şekilde dünyanın birçok bölgesinde 6-10 m derinlikten sonra 10~24°C olarak sabitlenmektedir. Yer sıcaklığının sığ derinliklerde sabit olması sayesinde, genel anlamda bilinen jeotermal enerji kaynaklarına ihtiyaç duyulmaksızın, dünyanın her yerinde ve her mevsimde, yeraltı tabakalarından yenilenebilir bir ısıtma ve soğutma kaynağı olarak faydalanmak mümkün olmaktadır. Dolayısıyla, yer içinde bulunan bu potansiyel jeotermal enerji, jeotermal sondaj kuyuları benzeri elemanlara bağlı bir ısı pompası yardımıyla yenilenebilir ve temiz bir enerji kaynağı olarak çok büyük bir olanak sunmaktadır.

Son yıllarda jeotermal enerji, sondaj kuyuları dışında binaların yapısal elemanları aracılığıyla da kullanılmaya başlanmıştır. Enerji kazıkları (veya ısı değişim kazıkları), yenilenebilir enerji alanında sürdürülebilirlik açısından yenilikçi ve geri dönüşümü olan bir teknolojidir. Bu bağlamda, enerji kazıkları, sabit yeraltı sıcaklığı ve zeminin ısı tutma kapasitesinden faydalanılarak binaların etkin olarak ısıtma ve soğutmasında kullanılabilir. Bu gelişen yeni yaklaşımda, binayı desteklemek için mevcut olarak kullanılan derin temel yapı elemanları aynı zamanda jeotermal ısıtma/soğutma elemanları olarak da kullanılmaktadır. Burada, kazıklar ısı transferini gerçekleştirmek

üzere dolaşım (devridaim) boruları içermektedir. Üst yapıdaki ısı enerjisi bir ısı pompası kullanılarak, antifriz katkılı su ve bu borular vasıtasıyla zemine iletilmektedir. Böylece, yaz mevsiminde binanın soğutulması, kış mevsiminde ise ısıtılması amacıyla ısı enerjisi sırasıyla yeraltına verilir ve yeraltından alınır. Temel sistemlerinde kullanılan kazıklar, genel olarak, jeotermal enerjiden sürekli olarak yararlanabilmek için yeterli uzunluğa sahiptir. Böylece, hâlihazırda imal edilecek olan bu kazıklı temel sistemine dolaşım boruları yerleştirilerek jeotermal enerji kullanılabilir. Bu şekilde, temiz ve yenilenebilir bir enerji kaynağı olan jeotermal enerjiden, her yerde ve her mevsimde yararlanmak oldukça düşük ek maliyetlerle mümkün olabilecektir.

Termo-aktif elemanlar olarak enerji kazıklarının ısı değişim mekanizmasının belirlenmesi yüksek önem taşımaktadır. Zemin bünyesindeki ısı transferinin temel mekanizması konduksiyon (ısı iletimi) olmakla beraber, yeraltı su akımı olduğu durumlarda konveksiyon (ısı döngüsü) da etkili olabilmektedir. Enerji kazıklarının ısı değişim performansı birçok parametreye bağlıdır. Bunlar; zeminin ısı iletkenliği, kazık malzemesi ve çapı, dolaşım borusu çapı, dolaşım sıvısı akış hızı ve dolaşım sıvısındaki karışım oranı, yeraltı su seviyesi gibi faktörlerdir. Bu parametrelerin etkilerini değerlendirmek için laboratuvar ve gerçek ölçekli arazi deneylerinin yanı sıra üç-boyutlu sonlu elemanlar analizleri gibi yöntemler kullanılmaktadır.

Isıtma ve soğutma çevrimlerinin yük altındaki kazık davranışına olan etkileri konusundaki araştırmalar ise sınırlıdır. Özellikle de, enerji kazıklarının uzun vadeli performansı ile ilgili verilerin mevcut olmaması ve ayrıca deney ve tasarım standartlarının bulunmaması bu konudaki belirsizliklerin ana unsurlarıdır. Enerji kazıklarının veya diğer termo-aktif temel elemanlarının sürdürülebilirliği zemin sıcaklığının sabit tutulmasına bağlıdır. Zemin sıcaklığındaki değişiklikler ısı değişim kapasitesini ve verimliliği direkt olarak etkilemektedir. Bu yüzden, yapının proje ömrü boyunca enerji kazıklarının performansının nasıl etkileneceğini analiz etmek kritik tasarım noktalarından biridir. Bu durum özellikle ısıtma ve soğutma arzlarının dengesiz olduğu bölgelerde daha fazla önem kazanmaktadır.

Enerji kazıklarının yaygın bir şekilde kullanılması önündeki önemli engellerden birisi de enerji kazıklarının termo-mekanik davranışlarının yeterince aydınlatılmamış olmasıdır. Kazıkların hem servis yükü, hem de ısıtma-soğutma çevrimlerine maruz kalması durumunu ifade eden termo-mekanik yükler altındaki davranışları halen araştırma konusudur. Özetle, yapılan araştırmalar sonucunda termal yüklerin kazıklarda fazladan gerilmelere neden olduğu ve kazıkların sürtünme direncini azalttığı görülmektedir. Bu tür termal etkiler, enerji kazıklarının çalışma programına göre önceden belirlenmeli ve kazık tasarımı bu bilgiler ışığında yapılmalıdır.

Enerji kazıklarının ısıtma ve soğutma çevrimlerinin zemine olan doğrudan etkisi görece daha bilinen bir konudur. Zemindeki ani sıcaklık artışı drenajsız durumlarda boşluk suyu basıncının artmasına ve efektif gerilmelerin azalmasına neden olabilir. Bunun sonucunda kayma mukavemeti kaybı ve kazıklı temelde oturmalar meydana gelebilmektedir. Bu bağlamda, enerji kazıklarının ısı değişimi davranışı ve ısıtma-soğutma döngüleri sırasında oluşan zemin-kazık etkileşimi büyük önem arz etmektedir. Bu bahsedilen konuların açıklığa kavuşturulması ve enerji kazıklarının davranışının daha iyi anlaşılması sayesinde yeni tasarım yöntemlerinin geliştirilmesi mümkün olacaktır.

Yer ısısına dayalı bu tip enerji aktarım sistemlerinin giderek yaygınlaşması durumunda, komşu yapıların jeotermal temelleri arasındaki etkileşimini anlayabilmek için enerji kazık gruplarının davranışlarının incelenmesinde fayda vardır. Ayrıca,

enerji kazıklarının uzun süreli termo-mekanik performansının çok iyi bilinmemesi sebebiyle, tasarım standartlarının oluşturulabilmesi açısından, bu konu da araştırılması gereken başlıklar arasında yer almaktadır. Enerji kazıklarının daha yaygın kullanılmasının önündeki engellerin kaldırılması için yukarıda özetlenen sorulara cevap aranması gerekmektedir. Enerji kazıklarının davranışına yönelik sorulara cevap aranan bu çalışma sonucunda bu son derece önemli uygulama daha yaygın kullanılabilir ve yüksek miktarda enerji tasarrufu sağlanabilecektir. Enerji fiyatlarının son derece yüksek olduğu ülkemiz özelinde bu daha da önem taşıyan bir etkiye sahiptir.

Bu tez çalışmasının amaçları şu şekilde listelenebilir: (1) zemin tabakalarının ısısal özelliklerinin hassas olarak belirlenmesi, (2) enerji kazıklarının termo-mekanik davranışlarının analizi ve (3) enerji kazıklarının ve kazık gruplarının uzun vadeli performanslarının tayin edilmesi.

Tekrarlı ısıtma ve soğutma çevrimlerinin etkileri sonucunda; enerji kazıklarının uzun süreli performansı, kazıktaki eksenel yükteki değişimler (ek gerilmeler), kazıkta tekrarlı dairesel şekil değiştirme ve zemin-kazık etkileşim arayüzünde yorulma ve gerilme rahatlaması gibi konular aydınlatılmıştır.

Tez çalışmasının içeriğinde ısı iletim teorisi ve jeotermal ısı değişim sistemleri hakkında literatür taraması bulunmaktadır. Yeraltı kaynaklı ısıtma/soğutma kavramı, yaygın olarak kullanılan jeotermal ısı değişimi elemanları ve arazi ısı iletkenlik deneyi ve analiz yöntemleri hakkında detaylı bilgi verilmiştir. Jeotermal ısı değişim elemanlarında meydana gelen termal süreçleri modellemek amacıyla sonlu elemanlar yöntemi kullanılarak üç tip nümerik model geliştirilmiştir. Bu modellerin doğrulaması analitik yöntem ve arazi deney verileri kullanılarak gerçekleştirilmiştir. Analiz sonuçları ve karşılaştırmaları detaylı bir şekilde tartışılmıştır. Arazi ısı iletkenlik deneyinin modellenmesini ve analizini geliştirmek amacıyla çeşitli yöntemler ortaya konmuştur. Ayrıca, bu deney ile ilgili iki farklı sahada yapılan vaka analizleri sunulmuştur. Sıcaklık değişimlerine maruz kalan enerji kazıklarının termo-mekanik davranışıyla ilgili kavramlar açıklanmıştır. Geliştirilen termo-mekanik nümerik model kullanılarak, enerji kazığında oluşması beklenen eksenel gerilme değişimlerini ve kazık-zemin arayüzünün davranışını incelemek amacıyla zemin rijitliğine ve kazığın mesnetlenme koşullarına bağlı olarak parametrik bir çalışma yapılmıştır. Benzer şekilde tekil kazıkların ve kazık gruplarının uzun vadeli performansını belirlemek amacıyla nümerik analizler sunulmuştur. Bu çalışma, sıcak, orta ve soğuk iklim olmak üzere üç farklı şehirde gerçekleştirilmiştir. Son olarak, gerçekleştirilen araştırma sonucunda elde edilen sonuçlar ile birlikte araştırmanın ileri aşamaları için tavsiyeler verilmiştir.

1. INTRODUCTION

There is a developing trend around the world to explore alternative energy sources. The main driving forces are growing global energy demand, depleting natural resources and the adverse effects of greenhouse gas emissions from fossil fuel consumption. Geothermal energy is one of the promising renewable sources that can be utilized to offset such trends. Yet, the use of geothermal energy has been limited mainly to certain hot spot areas where it is used either for district heating and/or electricity generation. On the other hand, ground temperature remains constant after a depth of about 6 to 10 *m* as the near surface soils act as a thermal insulator as the ambient temperatures vary seasonally. The value of the ground temperature depends on the regional climate and reflects the mean annual temperature at that specific location (Kusuda and Achenbach, 1965). In most regions this constant temperature is in the range of 10 to 24°C. The relatively constant temperature and the heat storage capacity of near-surface soils in any region represent a significant potential for stored thermal energy which in turn can be harvested with ground-source heat pump (GSHP) systems. These systems exemplify a highly efficient renewable energy technology for space heating and cooling (Sanner et al., 2003; Omer, 2008; Lund et al., 2011).

Added benefits of these systems include reduction in carbon dioxide (CO₂) emission and energy use for heating and cooling of the buildings. Electrical consumption and maintenance requirements of GSHP systems are lower than those required by conventional systems (i.e. air source heat pumps – ASHP), and consequently these systems have lower annual operating costs (Yu et al., 2002). Therefore, using GSHP systems results in significantly lower heating and cooling costs compared to conventional ASHP systems. GSHP systems also have a much higher coefficient of performance (COP) compared to ASHP systems. A higher COP can be achieved by a GSHP system because the heat source/sink temperature is relatively constant compared to air temperature and provides a more favorable baseline temperature compared to ambient air temperature when heating or cooling is needed in the winter and summer respectively. Generally these heat pump systems offer better levels of

comfort, reduced noise levels, lower greenhouse gas emissions and reasonable environmental safety compared to the other conventional technologies (Bandos et al., 2011). The above mentioned benefits enabled geothermal energy to receive considerable attention recently.

GSHP systems consist of embedded loop of pipes, buried in the ground and connected to a heat pump (Bose et al., 1991). A vertical borehole configuration is usually preferred over horizontal trench systems because it requires smaller footprint and can reach to deeper levels less affected by the near-surface seasonal temperature variations. Traditionally, geothermal boreholes have utilized this concept for space heating and cooling. In this system, one or more U-shaped tubing functioning as circulation loop is placed in a small-diameter borehole typically extending to a depth of 50-200 m. The hole is then backfilled with a mixture of sand, bentonite and/or cement. The loop is connected to a geothermal heat pump and the carrier fluid inside the loop is circulated. The heat energy is injected into the ground for cooling in the summer and extracted from the ground for heating in the winter. Large geothermal heat exchanger (GHE) systems consist of an array of boreholes, in which each borehole containing one or more circulation loops. GHEs of this type are also used in large thermal storage schemes in addition to building heating and cooling systems (Rees and He, 2013).

Over the past 20 years, the ground coupling concept has been expanded from geothermal borehole systems to the use of building foundation elements as heat exchangers. Heat exchanger pile (also mentioned as energy pile) in particular is an innovative technology that combines geothermal heat exchange and structural foundation support (Brandl, 2006; Abdelaziz et al., 2011; Kramer et al. 2015). Heat exchanger piles were first developed in Austria in the 1980s, but have only recently received attention and common usage worldwide (Brandl, 2006). As of 2000, a total of 80 countries have used some form of geothermal energy (Hepbasli, 2003). The number of installations worldwide have reached a million as indicated by several researchers (Spitler, 2005; Clarke et al., 2008). It is obvious that the applications of heat exchanger piles will increase over the next decade with further development of this technology.

In this hybrid system, geothermal circulation loops are integrated into the deep foundation elements, such as piles, piers, or drilled shafts that are already designed to provide structural support. Fluid is circulated through the tubes in the piles. The heat

energy is fed into the ground for cooling in the summer and withdrawn from the ground for heating in the winter. The temperature differential between the ground and the outside temperature acts as an energy pathway to harvest stored ground energy for bringing the building temperature to comfort zone levels.

As an added benefit, the additional cost of geothermal borehole drilling for loop placement is offset by the combined use since the heat exchanger piles are installed at sites where pile foundations are already required and installation costs of geothermal heat exchangers are significantly reduced. Low maintenance, long lifetime, less variation in energy supply compared to solar and wind power, and environmental friendliness have been cited as additional advantages. Case studies suggest that they can significantly lower heating/cooling costs and reduce carbon footprint. Energy cost savings for typical structures utilized with heat exchanger piles could be as much as 70 percent (Abdelaziz, 2013).

1.1 Research Motivation

The key parameters in the design and dimensioning of the ground-source heating and cooling systems are the thermal characteristics and the heat exchange capacity of the in-situ soil and rock formations. The magnitude and rate of thermal energy that can be harvested with a GHE depends mainly on the thermal properties of the ground, in particular thermal conductivity (Wagner and Clauser, 2005). Proper design and sizing of GHE systems for space heating/cooling requires a good estimate of the ground thermal properties. An in-situ testing method, namely field thermal conductivity test (TCT) was developed in the mid-80s and has been refined since then to measure the ground thermal properties on site. A variety of mobile equipment and test setups have been built by different researchers at several countries (Eklöf and Gehlin, 1996; Austin, 1998; Kavanaugh et al., 2001). The main purpose of a field TCT is to evaluate the thermal properties of the GHE and the subsurface. These parameters are required for the design of GSHP systems. Estimated properties are the thermal conductivity of the ground and the thermal resistance of the heat exchanger. These properties are used to calculate the required length of the GHE for a given energy demand, which is the main cost associated with a GSHP system. Therefore, an accurate estimation of thermal properties from TCT has a direct impact on the efficiency and cost of GSHP installations (Marcotte and Pasquier, 2008).

Over the years, various analytical and numerical models of varying complexities have been developed and used as design tools for GHEs. A number of design tools based on finite element or finite volume programs were used to develop fully discretized borehole heat exchanger models to include transient effects, as well as the complicated borehole geometries. Some of the most noteworthy models include the studies by Al-Khoury et al. (2005), Al-Khoury and Bonnier (2006), Signorelli et al. (2007), Marcotte and Pasquier (2008), Lamarche et al. (2010) and He (2012). Some of the models were limited to 2D discretization because of the number of small elements needed for the sufficient discretization of the borehole cross section (Austin, 2000; Yavuztürk, 1999; Yavuztürk et al., 1999). On the other hand, several hybrid models have also been developed to provide feasible analytical modeling alternatives (Eskilson, 1987; Yavuztürk and Spitler, 1999). Such models are used for calculating the ground temperature response functions numerically.

Analytical models typically have limitations in modeling considerations and are not as versatile as numerical models. The most commonly used analytical method is the line source method, which assumes an infinitely long linear heat source in modeling the heat conduction. This theory starts becoming inapplicable as the diameter of the heat source gets larger as in the case of a typical heat exchanger pile. Another limitation of the line source method is that it ignores end effects, which may start becoming more pronounced as the heat source becomes relatively short. However, analytical models are preferred in many practical applications because of their superior computation time efficiencies and better flexibility for parametric design. The inaccuracy in the results of the analytical models corresponds to the underlying modeling assumptions made when deriving analytical solutions for GHEs. Analytical models usually involve simplifying assumptions such as constant temperature or constant heat flux along the depth to make the analog solution possible. Consideration of constant fluid temperature along the depth or constant heat rejection can potentially result in over- or under-prediction of the ground temperature increments during GHE operations (Ghasemi-Fare and Basu 2013a and 2013b). Nevertheless, it must be noted that uncertainties in the field conditions and the associated quality of the input parameters may have a more significant effect on GHE performance estimation than the limitations originating from modeling approximations.

Moreover, thermal operation of a heat exchanger pile induces temperature changes in the piles. Such temperature changes create thermal stresses and displacements along the pile that, if ignored in the design process, can result on overstressing the piles and/or unacceptable building deformations (Laloui et al., 2003; Abdelaziz, 2013). Numerous numerical and analytical approaches were developed to approximate the thermally induced stresses and displacements in heat exchanger piles as presented by Abdelaziz (2013) and Knellwolf et al. (2011). All the techniques for analyzing the thermal effects on the mechanical behavior of the piles rely on accurate estimates of the temperature changes in the pile.

Sustainable operation of heat exchanger piles or any other thermo-active foundation element relies on maintaining the constant temperature of the ground. Ground temperature changes can directly affect the heat exchange capacity and efficiency of operations. It is critical to evaluate how the heat exchanger pile operation over the life cycle of the structure will change the temperature gradients around the pile. This is particularly important at regions where the respective heating and cooling energy demands are not balanced.

1.2 Research Objectives

The main objectives of this dissertation are listed below:

- Estimating the thermal properties of the ground accurately and efficiently using numerical modeling techniques. This is required for a robust design and assessing the thermal performance and efficiency of the GSHP system.
- Assessing the thermo-mechanical behavior of heat exchanger piles. The pile subjected to heating and cooling cycles will eventually expand and/or contract. It is thought that the volume changes and the constraints of the pile, i.e., the degree of fixity at the pile head and the pile toe will have an impact on the thermo-mechanical behavior. The amount of slip displacements at the pile-soil interface and the magnitudes of changes in pile axial stress, shear resistance and radial pressures on the pile soil-interface are the main points of concern.
- Predicting the long-term performance of heat exchanger piles and pile groups. The efficiency and sustainability of the system will be investigated for long-term seasonal operation using actual energy demands for locations with extreme climatic conditions, i.e., hot, moderate and cold. Single pile and pile

groups with numerous grid sizes will be investigated. Long-term thermo-mechanical behavior of single piles in each location will be assessed.

In order to meet the above mentioned research objectives, in the scope of this study, robust and competent finite element analysis tools will be developed for modeling geothermal heat exchangers. The numerical tools with different levels of complexity will be utilized to analyze various problems related to the operation of GHEs. These include modeling of field thermal conductivity testing, full scale thermo-mechanical testing of heat exchanger piles and long-term seasonal operation of heat exchanger piles and pile groups. The models are planned to be built extensible in nature so that modifications for considering various aspects such as the geothermal gradient or ambient temperature changes could be easily incorporated. Validation and calibration of the proposed numerical models are to be performed using available analytical methods and field testing data.

The dissertation is organized such that, Chapter 2 outlines the literature review on the theory of heat transfer and summarizes the academic background needed to study the thermal processes occurring in the geothermal heat exchangers. Chapter 3 sets out the concept of ground-source heating/cooling, types of commonly used geothermal heat exchangers, detailed discussion on field thermal conductivity testing and currently available analytical and numerical analysis methods. Chapter 4 presents three numerical modeling techniques for simulating the thermal processes in GHEs by utilizing finite elements. The validation of the models is performed via an analytical method, and a detailed comparison of the analysis results is presented. This chapter also includes several methods to improve the modeling and analysis of the field TCT. In Chapter 5, case studies of field TCTs performed in two different sites are discussed. Chapter 6 presents the concept of thermo-mechanical behavior of heat exchanger piles subjected to temperature changes. Thermo-mechanical numerical models are developed and utilized to assess the change in the stresses in the pile, contact pressures and the behavior of the pile-soil interface by performing parametric analyses with respect to soil strength and pile fixity. Chapter 7 extends the previously developed thermo-mechanical numerical model to study the long-term performance of the heat exchanger piles. This study includes analyses in three different cities, which have hot, moderate and cold climates, respectively. Finally, Chapter 8 draws the conclusions and discusses the recommendations for future research.

2. THEORY OF HEAT TRANSFER

Heat is thermal energy in transit due to a spatial temperature difference. Heat transfer is a thermodynamic process dealing with the rate at which thermal energy is transferred between a system and its surrounding with the mechanisms responsible for the transfer. Whenever a temperature difference exists in a medium or between media, heat transfer must occur (Bergman et al., 2011).

Heat transfer involves measuring and computing of two main entities: temperature and heat flow. The temperature represents the amount of thermal energy in a system, whereas the heat flow represents the movement of thermal energy as a result of the temperature gradient. The thermal energy is related to the kinetic energy of the material molecules, as at a higher temperature, the material exhibits more kinetic energy (Al-Khoury, 2011).

2.1 Modes of Heat Transfer

Different types of heat transfer processes, also known as modes, are classified into three main categories: conduction, convection and radiation as shown in Figure 2.1. When a temperature gradient exists in a stationary medium, which may be a solid or a fluid, heat transfer occurs by conduction across the medium. In contrast, convection occurs between a surface and a moving fluid when they are at different temperatures. For radiation, all surfaces of finite temperature emit energy in the form of electromagnetic waves. Hence, in the absence of an intervening medium, there is net heat transfer by radiation between two surfaces at different temperatures.

Heat transfer in borehole heat exchanger systems involves coupling between conduction and convection. Radiation heat transfer has no or negligible effect, hence it will not be discussed.

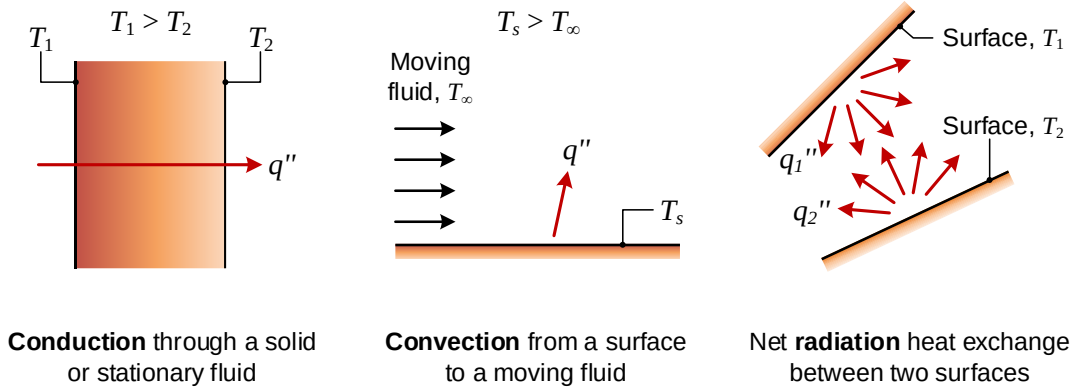


Figure 2.1 : Conduction, convection and radiation heat transfer modes (after Bergman et al., 2011).

2.1.1 Heat conduction

Heat conduction, also referred to as heat diffusion, is one of the most important modes of thermal energy flow. On a microscopic scale, and as required by the second law of thermodynamics, when two objects at different temperatures are placed in thermal contact, thermal energy always flows from the object with high temperature to the one with lower temperature, never in the opposite direction. Heat flow between the two regions continues until they reach the same temperature, a state known as thermal equilibrium. On a macroscopic scale, the heat transfer rate per unit area normal to the direction of heat flow, known as heat flux, is described by Fourier's law of heat conduction in three-dimensions as follows:

$$q'' = -\mathbf{k}\nabla T$$

$$q'' = -\mathbf{k} \left(\frac{\partial T}{\partial x} \hat{i} + \frac{\partial T}{\partial y} \hat{j} + \frac{\partial T}{\partial z} \hat{k} \right) \quad (2.1)$$

where,

q'' : heat flux (W/m^2),

$\mathbf{k}$: thermal conductivity tensor ($\text{W}/\text{m}\cdot\text{K}$),

∇T : temperature gradient (K/m).

Using the first law of thermodynamics (conservation of energy); and by adding the terms of internal energy storage and internal heat generation, Equation (2.1) can be extended to obtain the general heat diffusion equation, which is often referred as the heat equation:

$$\rho c_p \frac{\partial T}{\partial t} = \nabla \cdot (\mathbf{k} \nabla T) + Q \quad (2.2)$$

where,

ρ : mass density (kg/m^3),

c_p : specific heat capacity ($J/kg \cdot K$),

$\partial T / \partial t$: temperature change (K),

Q : internal heat generation (W).

Assuming an isotropic and constant thermal conductivity, and in the absence of internal heat generation, the heat equation can be written as:

$$\frac{1}{\alpha} \frac{\partial T}{\partial t} = \nabla^2 T \quad (2.3)$$

where,

α : thermal diffusivity (m^2/s),

∇^2 : Laplacian operator, which is described as the divergence of a gradient.

In the three-dimensional Cartesian coordinate system, the Laplacian operator is given as:

$$\nabla^2 = \nabla \cdot \nabla = \frac{\partial^2}{\partial x^2} + \frac{\partial^2}{\partial y^2} + \frac{\partial^2}{\partial z^2} \quad (2.4)$$

In the cylindrical coordinate system, it is written as:

$$\nabla^2 = \frac{\partial^2}{\partial r^2} + \frac{1}{r} \frac{\partial}{\partial r} + \frac{1}{r^2} \frac{\partial^2}{\partial \phi^2} + \frac{\partial^2}{\partial z^2} \quad (2.5)$$

2.1.2 Heat convection

Heat convection is an important mode of heat transfer, basically associated with fluid flow. Convection is an energy transfer across a system boundary due to a temperature difference by the combined mechanisms of intermolecular interactions and bulk transport. In addition to energy transfer due to random molecular motion (diffusion),

energy is also transferred by the bulk, or macroscopic, motion of the fluid (advection). Hence, heat convection occurs due to simultaneous heat diffusion and advection.

At the macroscopic level, the convective heat flow rate per unit area normal to the direction of heat flow (convective heat flux), for a one-dimensional case, is described as:

$$q_x'' = \underbrace{-k_x \frac{dT}{dx}}_{\text{diffusion}} + \underbrace{\rho c_p u_x T}_{\text{advection}} \quad (2.6)$$

where, u_x is the fluid velocity along the x-axis. The first term on the right hand side of Equation (2.6) denotes the diffusion heat transfer, while the second term is associated with advection.

Convection heat transfer occurs between a fluid in motion and a bounding surface when the two are at different temperatures. As shown in Figure 2.2, as a consequence of fluid-surface interaction, a boundary layer is developed in the fluid through which the velocity varies from zero at the surface to a finite value u_∞ associated with the flow (Bergman et al., 2011). Moreover, if the surface and flow temperatures differ, there will be a region of the fluid through which the temperature varies from T_s at $y = 0$ to T_∞ in the outer flow. This region, called the thermal boundary layer, may be smaller, larger, or the same size as that through which the velocity varies. In any case, if $T_s > T_\infty$, convection heat transfer will occur from the surface to the outer flow.

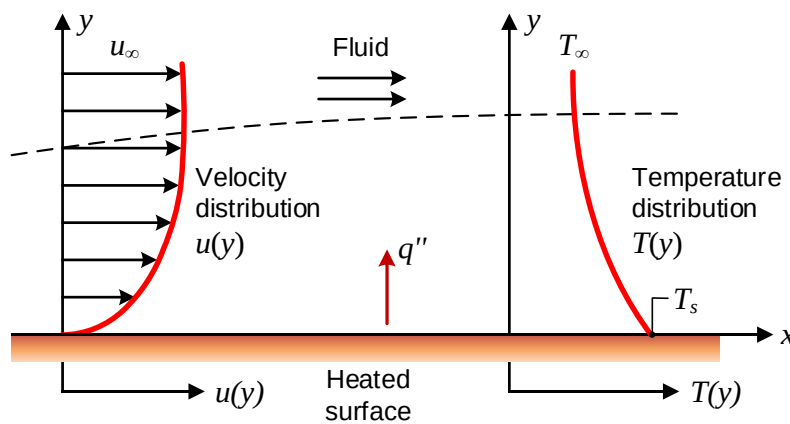


Figure 2.2 : Development of boundary layer in convection heat transfer (after Bergman et al., 2011).

Convective heat transfer processes involving heat transfer from a boundary surface of temperature T_s exposed to a fluid flow of temperature T_∞ is described by the Newton's law of cooling, as:

$$q'' = h(T_s - T_\infty) \quad (2.7)$$

where, h ($\text{W}/\text{m}^2\cdot\text{K}$) is the coefficient of convection heat transfer.

This coefficient depends on conditions in the boundary layer, which are influenced by surface geometry, the nature of the fluid motion, and an assortment of fluid thermodynamic and transport properties. Convection heat transfer may be classified according to the nature of the flow. If the fluid motion is induced by an external force, such as a pump, a fan or atmospheric winds, the heat flow process is called forced convection. If, however, the fluid motion is induced by buoyancy forces, which are due to density variations caused by temperature, the process is called free or natural convection. Typical values of convection heat transfer coefficient are given in Table 2.1 (Bergman et al., 2011).

Table 2.1 : Typical values of the coefficient of convection heat transfer (Bergman et al., 2011).

Process	h ($\text{W}/\text{m}^2\cdot\text{K}$)
Free convection	
Gases	2 – 25
Liquids	50 – 1,000
Forced convection	
Gases	25 – 250
Liquids	100 – 20,000
Convection with phase change	
Boiling or condensation	2,500 – 100,000

For three-dimensional problems, and in the presence of fluid flow, the temperature must be determined by solving for the corresponding three-dimensional heat equation. The three-dimensional heat conduction-convection equation can be obtained as:

$$\rho c_p \frac{\partial T}{\partial t} = \nabla \cdot (\mathbf{k} \nabla T) + \rho c_p \mathbf{u} \cdot \nabla T + Q \quad (2.8)$$

where, $\mathbf{u}$ is the flow velocity field.

2.2 Thermal Properties of Materials

There are several thermal parameters that have direct influence on the rate of heat transfer and the pattern of the temperature distribution in solids and fluids. In this section, an overview of thermal material properties and fluid flow parameters that are of direct interest in borehole heat exchangers are given. These properties are listed in Table 2.2 along with their units. The significance of these parameters, including the thermal fluid flow numbers that has great importance in understanding the heat transfer mechanisms in borehole heat exchangers, will be introduced in the following subsections.

Table 2.2 : Thermal properties of materials.

Property	Notation	Units
Thermal conductivity	k	$W/m \cdot K$
Mass density	ρ	kg/m^3
Specific heat capacity	c_p	$J/kg \cdot K$
Volumetric heat capacity	$C_p = \rho c_p$	$J/m^3 \cdot K$
Thermal diffusivity	$\alpha = k / \rho c_p$	m^2/s
Porosity	n	—
Dynamic viscosity	μ	$Pa \cdot s$
Kinematic viscosity	$\nu = \mu / \rho$	m^2/s
Reynolds number	Re	—
Prandtl number	$Pr = \mu / \alpha$	—
Peclet number	$Pe = Re \cdot Pr$	—
Nusselt number	Nu	—

2.2.1 Thermal conductivity

Thermal conductivity is fundamental thermal property of a material which describes its ability to conduct heat. It appears primarily in Fourier's law for heat conduction and is defined as the quantity of heat transmitted through a unit thickness in a direction normal to a surface of a unit area, due to a unit temperature gradient under a steady-state condition. As the physical basis for conductivity is at the molecular and atomic levels, solids would generally have the highest conductivities, followed by liquids, and finally gases. Ranges of thermal conductivities for various classifications of matters are given in Figure 2.3 (Wendl, 2012).

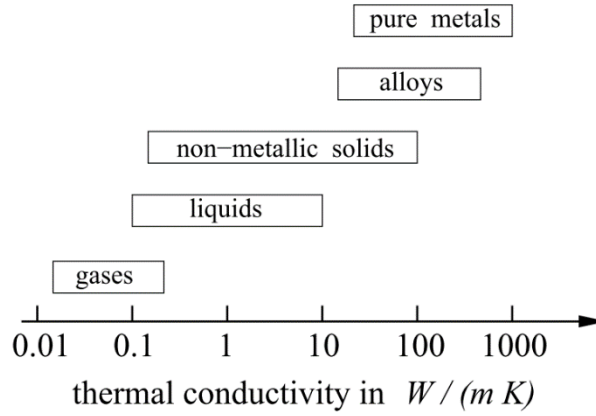


Figure 2.3 : Approximate ranges of thermal conductivity for various materials (Wendl, 2012).

2.2.2 Mass density

Mass density (or density) of a material is defined as its mass per unit volume. For a homogeneous material, it is given as:

$$\rho = \lim_{\Delta V \rightarrow 0} \frac{\Delta m}{\Delta V} \quad (2.9)$$

In general, density varies with pressure and temperature. Although this variation can be significant in gases, in the context of borehole heat exchangers, densities of involved materials can be regarded as constant.

The density of a multiphase material can be estimated using local volume averaging. For a two-phase porous material, the average density can be calculated by:

$$\rho_{\text{eff}} = n\rho_f + (1-n)\rho_s \quad (2.10)$$

where, n is the porosity and subscripts f and s denote the fluid and solid phases, respectively.

2.2.3 Specific heat capacity

Specific heat capacity, also termed specific heat or specific thermal capacity, characterizes the amount of heat required to change a unit mass of a substance's temperature by one Kelvin. It is defined as the ratio of the amount of transferred heat energy per unit mass to the resulting temperature change as:

$$c = \frac{Q}{m\Delta T} \quad (2.11)$$

In thermodynamics, there are two definitions of the specific heat. c_p is the specific heat at constant pressure and c_v is the specific heat at constant volume. For incompressible materials, such as solids and liquids, both properties are essentially equal.

The specific heat capacity varies with temperature. However, it can be treated as constant, if the temperature range is relatively small, as in the case of borehole heat exchanger systems. For example, at a constant atmospheric pressure, the specific heat of water varies by only 1% for a temperature ranging between 0°C and 100°C (Al-Khoury, 2011).

Volumetric heat capacity is obtained by multiplying the specific heat by the mass density. In case of a multiphase material, volumetric heat capacity can be calculated using local volume averaging. For a two-phase porous material, it is given by (Rees et al., 2000; Al-Khoury, 2011):

$$(\rho c_p)_{\text{eff}} = n(\rho c_p)_f + (1-n)(\rho c_p)_s \quad (2.12)$$

2.2.4 Thermal diffusivity

Thermal diffusivity is the ratio of the thermal conductivity to the volumetric heat capacity of the material and described as:

$$\alpha = \frac{k}{\rho c_p} \quad (2.13)$$

As it can be deduced from the Equation (2.13), thermal diffusivity is a measure of thermal inertia. Heat moves faster in materials with high thermal diffusivity, and they rapidly adjust their temperature to that of their surroundings because heat is conducted quickly relative to their volumetric heat capacity.

2.2.5 Viscosity

Viscosity is a measure of the resistance of a fluid which is being deformed by either shear stress or tensile stress. It can be described as the internal friction of the fluid, where less viscosity means less friction and the greater ease of movement. With the exception of superfluids, all real fluids have some resistance to shear stress and therefore are viscous. In general, a liquid with the viscosity less than water is known as a mobile liquid, while a substance with a viscosity substantially greater than water is simply called a viscous liquid. In a Newtonian fluid (such as water), the shear stress

between layers is proportional to the velocity gradient in the direction perpendicular to the layers, such that:

$$\tau = \mu \frac{du}{dy} \quad (2.14)$$

where, τ is the shear stress, du/dy is the velocity gradient perpendicular to the direction of shear and μ is the dynamic viscosity ($Pa \cdot s$).

Kinematic viscosity is obtained by dividing dynamic viscosity by the mass density:

$$\nu = \frac{\mu}{\rho} \quad (2.15)$$

Viscosity of a Newtonian fluid is a function of temperature and pressure, and for liquids, it depends mainly on temperature. For example, dynamic viscosity of water changes from $0.28 \text{ mPa} \cdot \text{s}$ to $1.79 \text{ mPa} \cdot \text{s}$ as the temperature ranges from 100°C to 0°C . It can be reasonably considered constant within the temperature range of geothermal heat exchangers (Al-Khoury, 2011).

2.3 Characteristic Numbers for Fluids

Characteristic numbers are dimensionless numbers used in fluid mechanics to describe the character of the flow, such that, the flow regime and the heat transfer mode which takes place. Thus, they have great importance in the definition of heat transfer mechanisms in a borehole heat exchanger system.

2.3.1 Reynolds number

Reynolds number, Re is defined as the ratio of inertial forces to viscous forces, such that:

$$Re = \frac{\rho u d_h}{\mu} = \frac{u d_h}{\nu} \quad (2.16)$$

where, u is the mean flow velocity (m/s) and d_h is the hydraulic diameter (m).

Reynolds number characterizes the regime of the flow; whether it is laminar, transient or turbulent. Typical classification of the flow regime is given in Equation (2.17).

$$\begin{aligned}
\text{Re} < 2300 & \text{ laminar} \\
2300 < \text{Re} < 4000 & \text{ transient} \\
\text{Re} > 4000 & \text{ turbulent}
\end{aligned}
\tag{2.17}$$

2.3.2 Prandtl number

Prandtl number, Pr , is the ratio of momentum diffusivity (kinematic viscosity) to thermal diffusivity. It is defined as:

$$Pr = \frac{\nu}{\alpha} = \frac{c_p \mu}{k} \tag{2.18}$$

Prandtl number is a measure of diffusion with respect to the fluid velocity and it controls the relative thickness of the momentum and thermal boundary layers. When Pr is small, heat diffuses very quickly compared to the flow velocity (momentum). Prandtl number of water at 0°C is 12.99 and at 20°C is 6.96 (Rohsenow et al., 1998).

2.3.3 Péclet number

Péclet number, Pe , is defined as the ratio of the rate of advection to the rate of diffusion. In the context of heat transfer, the Péclet number is equivalent to the product of the Reynolds number and the Prandtl number, that is

$$Pe = Re \cdot Pr \tag{2.19}$$

The Péclet number is used to characterize the mode of the heat transfer. At low Pe , the heat transfer is dominated by conduction, whereas at high Pe , it is dominated by convection.

2.3.4 Nusselt number

Nusselt number, Nu , can be described as the ratio of convective to conductive heat transfer across a boundary. It is defined as:

$$Nu = \frac{h d_h}{k} \tag{2.20}$$

A Nusselt number close to unity, namely convection and conduction of similar magnitude, is a characteristic of laminar flow. A larger Nusselt number typically in the range between 100 and 1000 corresponds to more active convection with turbulent flow.

For internal forced convection with turbulent flow conditions, which is the case for the borehole heat exchangers, Nusselt number is expressed as a function of Reynolds number and Prandtl number. There are many correlations for estimating the Nusselt number and the most common are given by Dittus-Boelter (1930) and Gnielinski (1976). Dittus-Boelter correlation is given as:

$$\begin{aligned} \text{heating of the fluid: } Nu &= 0.0241 Re^{0.8} Pr^{0.4} & 0.7 < Pr < 160 \\ \text{cooling of the fluid: } Nu &= 0.0264 Re^{0.8} Pr^{0.3} & Re \geq 10000 \end{aligned} \quad (2.21)$$

Gnielinski's (1976) correlation is defined as:

$$Nu = \frac{(f_D/8)(Re-1000)Pr}{1+12.7(f_D/8)^{1/2}(Pr^{2/3}-1)}, \quad \begin{aligned} &0.5 < Pr < 2000 \\ &3000 < Re < 6 \times 10^6 \end{aligned} \quad (2.22)$$

where, f_D is the Darcy friction factor, which can be estimated using the Churchill (1997) equation as follows:

$$f_D = 8 \left[\left(\frac{8}{Re} \right)^{12} + (C_A + C_B)^{-1.5} \right]^{1/12} \quad (2.23)$$

where,

$$C_A = \left[-2.457 \ln \left(\left(\frac{7}{Re} \right)^{0.9} + 0.27(e/D) \right) \right]^{16} \quad \text{and} \quad C_B = \left(\frac{37530}{Re} \right)^{16} \quad (2.24)$$

Equation (2.23) is valid for all flow conditions. Moreover, e in Equation (2.24) denotes the absolute surface roughness of the pipe; and for plastic pipes, it is given as 0.0015 mm.

2.4 Temperature Dependent Material Properties

Most of the time, the material properties are assumed to be constant during the operation of geothermal heat exchangers. On the other hand, these properties are usually temperature dependent. As an example, the temperature dependent material properties of water are shown in Figure 2.4.

This figure presents the temperature dependent four parameters of water which are: (1) dynamic viscosity; (2) thermal conductivity; (3) mass density; and (4) specific heat capacity. In numerical modeling approaches, other flow and heat transfer related

parameters can be evaluated during runtime, using these temperature dependent functions and the temperature of the water at an instance.

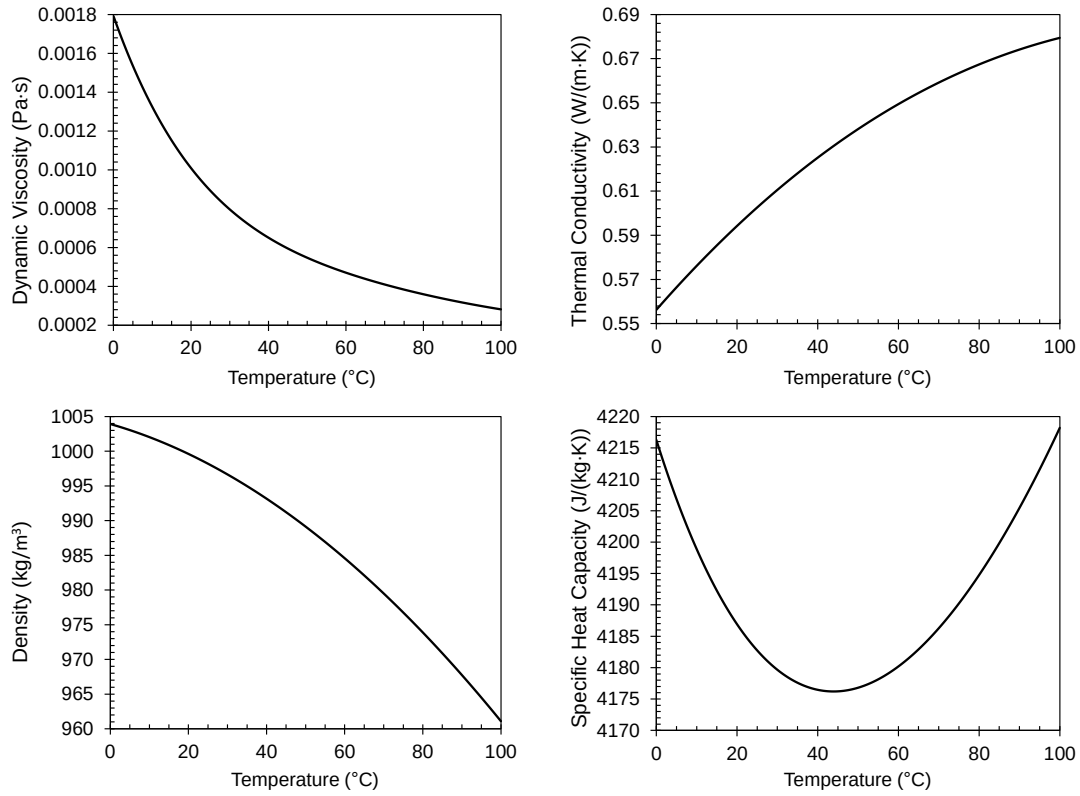


Figure 2.4 : Material properties of water with respect to temperature.

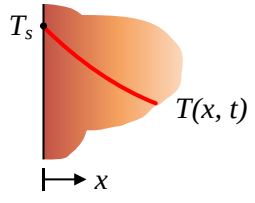
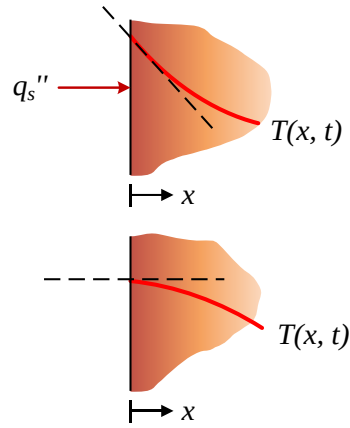
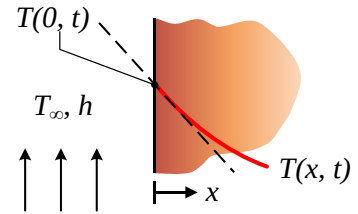
2.5 Boundary and Initial Conditions

The heat equation governs the physics of how heat is transferred in a domain. However, to determine the temperature distribution in a medium, it is necessary to solve the appropriate form of the heat equation. The solution of the heat equation depends on the physical conditions existing at the boundaries of the domain, which are called the boundary conditions. Moreover, if the problem is unsteady (transient), the conditions of the medium at some initial time must be known. These are called the initial conditions. Since the heat equation is second order in the spatial coordinates, two boundary conditions must be expressed for each coordinate needed to describe the system. Because the equation is first order in time, however, only one initial condition must be specified.

Three kinds of boundary conditions commonly encountered in heat transfer are summarized in Table 2.3 (Bergman et al., 2011). For the case presented here, the conditions are specified at the surface $x = 0$ for a one-dimensional system. Heat

transfer is in the positive x -direction with the temperature distribution, which may be time dependent, designated as $T(x, t)$. These boundary condition types will be explained in the following sub-sections.

Table 2.3 : Boundary conditions for the heat diffusion equation at the surface, where $x = 0$ (after Bergman et al., 2011).

Boundary Condition	Representation
Prescribed temperature $T(0, t) = T_s$	
- Prescribed heat flux - Finite heat flux $-k \frac{\partial T}{\partial x} \Big _{x=0} = q_s''$ - Adiabatic or insulated surface $\frac{\partial T}{\partial x} \Big _{x=0} = 0$	
Convection $-k \frac{\partial T}{\partial x} \Big _{x=0} = h [T_\infty - T(0, t)]$	

2.5.1 Initial conditions

The initial condition in a heat transfer problem is defined by prescribing the temperature T on the domain of interest at an instant of time, which is usually selected as $t = 0$.

$$T(x, y, z, t) \Big|_{t=0} = T(x, y, z, 0) = T_0 \quad (2.25)$$

2.5.2 Prescribed temperature boundary condition (first kind: Dirichlet)

Similar to the initial condition, this boundary condition sets a specific value of temperature at a coordinate location, which is a boundary of the domain. Prescribed temperature boundary condition is also known as the boundary condition of the first

kind, or Dirichlet boundary condition. As an example, to set a surface temperature of T_s at boundaries $x = 0$ and $x = L$, the boundary conditions are defined as:

$$T(x, y, z, t)|_{x=0} = T_s \quad \text{and} \quad T(x, y, z, t)|_{x=L} = T_s \quad (2.26)$$

2.5.3 Prescribed heat flux boundary condition (second kind: Neumann)

Boundary condition of the second kind, also known as Neumann condition, corresponds to the existence of a fixed or constant heat flux at the surface boundary. This heat flux is related to the temperature gradient at the surface by Fourier's law, given in Equation (2.1). Prescribed heat flux boundary condition at a boundary of $x=L$ can be written as:

$$q_s'' = -k \frac{\partial T}{\partial x} \bigg|_{x=L} \quad (2.27)$$

The special case of this boundary condition is the no heat flux boundary condition, also known as adiabatic or perfectly insulated surface and given as:

$$\frac{\partial T}{\partial x} \bigg|_{x=L} = 0 \quad (2.28)$$

2.5.4 Convective boundary condition (third kind: Robin)

The boundary condition of the third kind, also called Robin boundary condition, corresponds to an energy balance between conduction and convection at the boundary. The latter is expressed by Newton's law of cooling and was given in Equation (2.7), by. This equation can be rewritten for a boundary at $x=L$ as follows:

$$q_s'' = h(T|_{x=L} - T_\infty) \quad (2.29)$$

The form of Robin boundary condition is a combination of the first two kinds of boundary conditions in that it contains both temperature itself and its derivative. Equating equations (2.27) and (2.29), the boundary condition is obtained as:

$$-k \frac{\partial T}{\partial x} \bigg|_{x=L} = h(T|_{x=L} - T_\infty) \quad (2.30)$$

2.6 Heat Transfer Mechanisms in a Geothermal Heat Exchanger

A typical geothermal heat exchanger consists of one or more plastic U-tubes, surrounded by a bentonite-cement grout. The U-tubes carry a circulating fluid, usually water or water with 20%–25% anti-freezing coolant. Since these materials have different thermal parameters, the mechanism of heat transfer in each component differs significantly from the others, and as a result, affects the mechanism of heat transfer in the GHE system.

Another important factor that affects the mechanism of heat transfer in the GHE is the geometry of the involved components and their thermal interactions. As shown in Figure 2.5, each component is in direct contact with one or more components, and in an indirect contact with some others. The importance of the thermal interactions in the whole process depends on the size of the borehole and the distances between the pipes, known as the shank space.

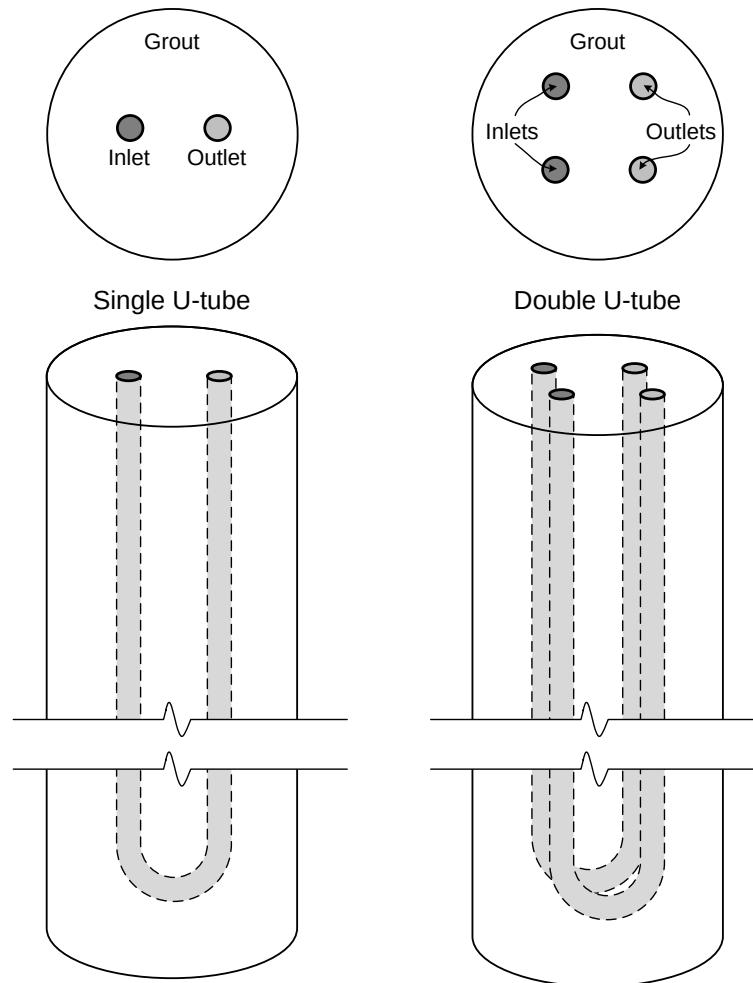


Figure 2.5 : Geothermal heat exchangers with single and double U-tubes.

Due to its complex nature, a GHE system involves different modes of heat transfer and requires coupling of fluid flow with heat transfer as well. Furthermore, since convection dominates the heat exchange between the carrier fluid and the pipe, and conduction governs in other system components, these heat transfer modes must be coupled throughout the whole system. The heat transfer processes taking place in a borehole heat exchanger can be investigated in two parts, namely inside the borehole and in the surrounding soil as follows:

Heat transfer inside the borehole: coupling of the fluid flow and heat transfer

- Convection between the carrier fluid and the pipe inner wall
- Conduction between the pipe inner wall and the pipe outer wall
- Conduction heat transfer between the pipe outer wall and the borehole wall

Heat transfer between the GHE and the surrounding ground

- Conduction between GHE wall and the surrounding ground
- Convection in the ground, if there is ground water flow

The heat transfer mechanisms inside the borehole and in the surrounding ground are schematically shown in Figure 2.6 (Abdelaziz et al., 2011). The direction of the arrows points to the direction of heat flow as the dark arrows indicate heat injection into the ground (cooling operation) and light arrows denote heat extraction from the ground (heating operation). Different factors related to these heat transfer mechanisms are listed in Table 2.4 (Abdelaziz et al., 2011).

Table 2.4 : Factors affecting the thermal design of geothermal heat exchangers classified with respect to the corresponding heat transfer mode (Abdelaziz et al., 2011).

Heat Transfer Mode	Controlling Factor
Conduction	• Thermal conductivity of the materials; carrier fluid, pipe walls, pile and in-situ soil
Convection	• Fluid flow conditions (flow rate of the carrier fluid, laminar vs. turbulent flow) • Inner diameter of the circulation pipe • Properties of the carrier fluid (dynamic viscosity, mass density, specific heat capacity etc.) • Ground water flow

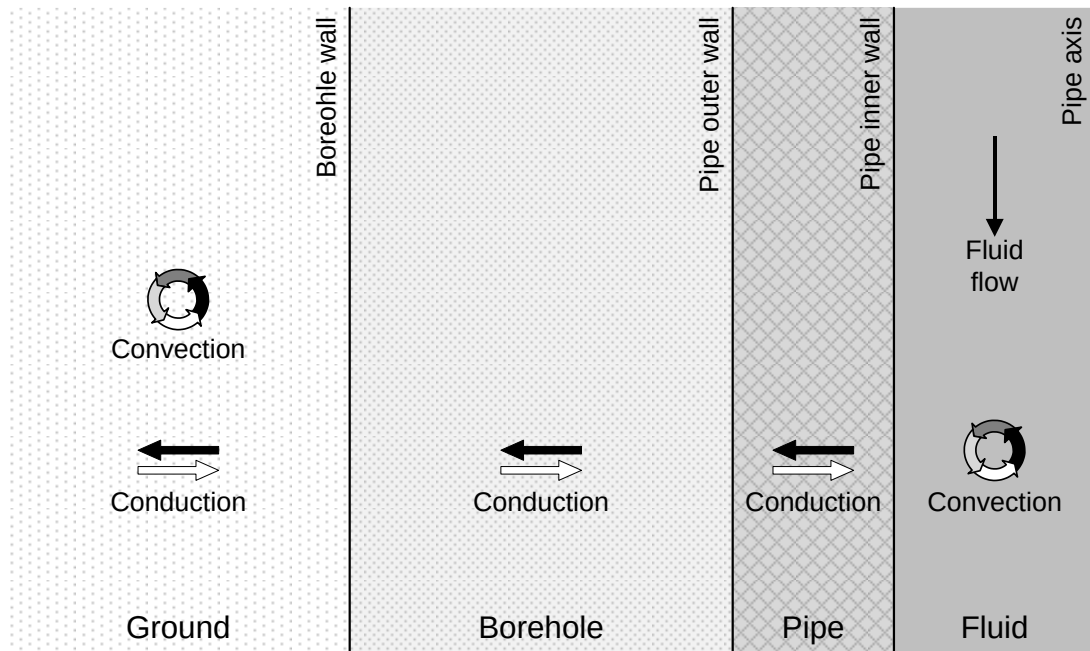


Figure 2.6 : Schematic view of heat transfer mechanisms in a borehole heat exchanger (not to scale) (after Abdelaziz et al., 2011).

For reference, thermo-physical properties of various materials that are involved in heat transfer mechanisms in a BHE are presented in Table 2.5 (Al-Khoury, 2011).

Table 2.5 : Thermo-physical properties of typical materials (Al-Khoury, 2011).

Material	k (W/m·K)	ρ (kg/m ³)	c_p (J/kg·K)	n	ν (m ² /s)
Ground					
Limestone	1.20–2.15	2300–2500	800–900	0.00–0.20	–
Sandstone	1.80–2.90	2160–2300	700–800	0.05–0.30	–
Sand	0.15–4.00	1280–2150	800–1480	0.20–0.60	–
Clay	0.15–2.50	1070–1600	920–2200	0.33–0.60	–
Soil	0.40–0.60	1600–2050	1800–1900	0.30–0.50	–
Borehole and fluid					
Polyethylene	0.33	960	2100	–	–
Grout	0.80–1.50	1100–1400	2000–2200	–	–
Water	0.56	1000	4180	–	1.006e-6
Water + 25% ethyl. glycol	0.50	1050	3795	–	4.950e-6

3. GROUND-SOURCE HEATING/COOLING AND FIELD THERMAL CONDUCTIVITY TESTING

Residential installation of ground source heat pumps (GSHPs) began in the late 1970s, and commercial installation has become increasingly popular in the last 15 years (Spitler, 2005). This low carbon emissions technology is mostly used to heat and cool buildings and their water supplies. About 900,000 units are estimated to be in operation throughout the world, for a total installed capacity of 10,100 MW and an annual energy use of 59,000 TJ (Curtis et al., 2005). A GSHP relies on the earth's renewable resources to provide or absorb thermal energy at low operating cost, making this system one of the most energy-efficient alternatives. The system is expensive to install because of the cost associated with ground heat exchangers. The installation cost can, however, be lowered in mining environments due to the abundance and accessibility of resources to be used with heat pumps.

3.1 Ground-Source Heat Pumps

Ground-source heat pump systems are subdivided into three types (ASHRAE, 2009): ground water heat pumps (GWHPs), surface water heat pumps (SWHPs) and ground-coupled heat pumps (GCHPs). All systems consist of heat pump units connected to different types of ground heat exchangers and linked to the earth's thermal resources. In winter, the GSHP operates under a heating mode and transfers thermal energy from the earth to the building. In summer, the system reverses to a cooling mode and transfers thermal energy from the building to the earth, which acts as a heat sink. Energy loss or gain is transferred to the earth by the ground heat exchangers in an open or closed-loop system. The former system uses water pumped directly from an aquifer or from a surface water body. The latter system circulates water, or a mixture of water and antifreeze, in a network of pipes commonly installed in soils, sedimentary deposits, host rock or surface water body.

The three different types of GSHPs, with their respective ground heat exchangers, are shown in Figure 3.1 (Florides and Kalogirou, 2007). GWHPs are open-loop systems

where groundwater can be pumped from water wells and discharged to a surface water body (Figure 3.1a), or returned to other wells for injection in the aquifer (Figure 3.1b). Heat pump heat exchangers are difficult to remove and clean. To prevent corrosion and fouling, it is a common practice to circulate water from the open-loop into an intermediate plate heat exchanger (shown in yellow in Figure 3.1a, b and c) to isolate the building loop (ASHRAE, 2009). Acidic and sulfuric water can corrode and damage heat pump heat exchangers, hard water can precipitate calcium carbonate scales that reduce the performance of heat pumps, and the presence of Fe^{2+} in sufficient concentration can cause fouling when water in the system is exposed to air (Rafferty, 2003). Intermediate plate heat exchangers are, on the other hand, made of corrosion-resistant materials (commonly stainless steel) to cope with various types of water and are used to transfer energy from the groundwater to another fluid going through the heat pump. Maintenance is performed easily on the intermediate plate heat exchangers because they can be dismantled.

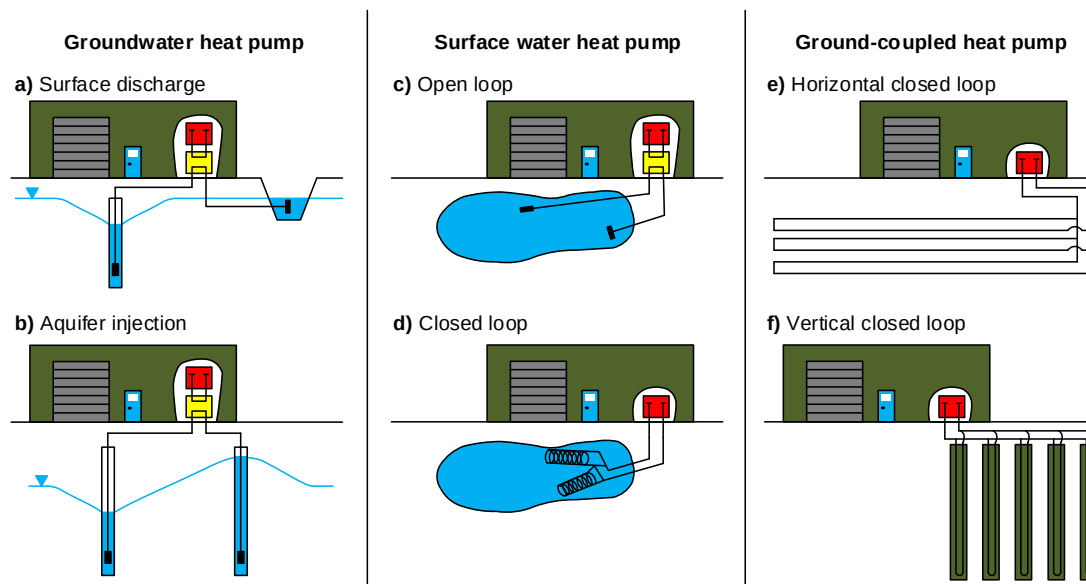


Figure 3.1 : Schematic drawing showing the types of GSHP system configurations: GWHP a) discharging water at surface or b) in the sub-surface, SWHP using c) the surface water directly or d) close-loop heat exchangers and GCHP using e) horizontal or f) vertical ground heat exchangers. The heat pump units and the intermediate plate heat exchangers located in the buildings are schematized with red and yellow rectangles, respectively (Florides and Kalogirou, 2007).

SWHPs can be designed as open or closed-loop systems. Water is directly pumped and returned to the surface water body in the open-loop SWHP (Figure 3.1c). Slinky, high-density polyethylene (HDPE) pipes, metallic plates or other types of exchangers are installed in the surface water body and make the closed-loop heat exchangers (Figure

3.1d). SWHP with a closed loop is a viable alternative when the pond water quality is low because polyethylene is a polymer that resists various chemicals (Budinski and Budinski, 2005). For example, it is recommended to use HDPE with up to a 75% volume of sulphuric acid solution at 20°C (Mellan, 1976).

Ground-coupled heat pumps use closed-loop ground heat exchangers installed horizontally or vertically. Trenches are dug to install the horizontal heat exchangers (Figure 3.1e), which are made of HDPE or PEX piping. A borehole is drilled to insert HDPE piping and the borehole is then filled with materials, such as grout, to form vertical heat exchangers (Figure 3.1f). Large commercial systems using closed-loop configurations are designed with parallel piping connections.

The heat pump unit, mostly water-to-air or water-to-water, transfers heat either from the building air or a hydronic distribution system to the ground heat exchanger, or in the opposite direction. The operation of a water-to-air unit is schematized in Figure 3.2 and explained below. During the heating cycle (Figure 3.2a), water coming from the ground heat exchanger circulates in a refrigerant-to-water heat exchanger, where the water heat is transferred to a low-pressure refrigerant that is in a vapor-liquid mixture state. The refrigerant evaporates in the refrigerant-to-water heat exchanger and continues to a compressor. Work is performed by the compressor to increase the pressure of the refrigerant, resulting in a temperature increase. The compressor work requires an electric energy input. The high-pressure refrigerant vapor moves towards a refrigerant-to-air heat exchanger, where the refrigerant heat is transferred to air. The warm air can then be distributed in the building. The refrigerant condenses in the refrigerant-to-air heat exchanger and goes to an expansion valve, where the refrigerant temperature is reduced through pressure decrease. The low-pressure refrigerant is returned to the refrigerant-to-water heat exchanger and the cycle continues.

The system is reversed for the cooling cycle (Figure 3.2b). The high-pressure refrigerant vapor leaving the compressor transfers its heat to the water coming from the ground heat exchanger. The refrigerant is cooled and condensed in the refrigerant-to-water heat exchanger and moves to the expansion valve. The pressure is released and the refrigerant becomes a mixture of liquid and vapor. The warm building air transfers its heat to the refrigerant-to-air heat exchanger, where the refrigerant is evaporated. Building water supplies can optionally be heated by the heat pump unit.

Descriptions of GSHP operation and characteristics are found in Wright and Colvin (1993), Kavanaugh and Rafferty (1997), and Omer (2008).

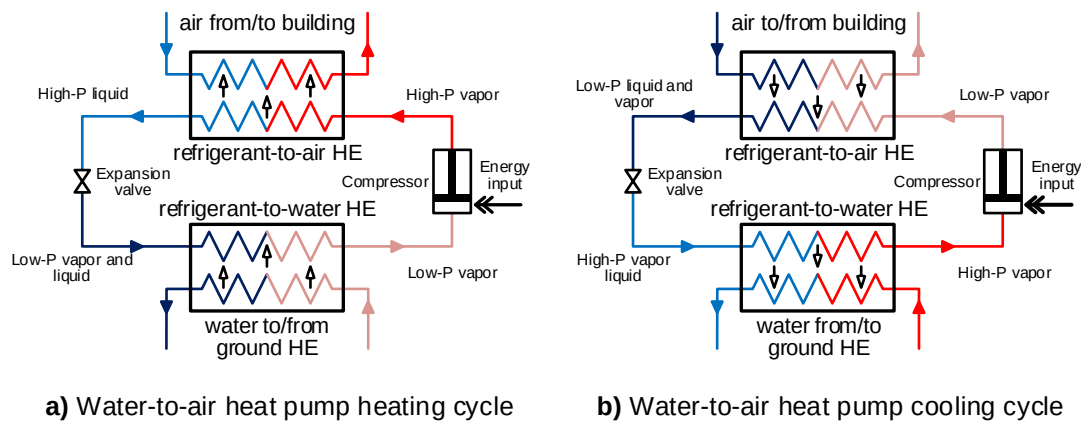


Figure 3.2 : Schematic drawing showing the operation of a water-to-air heat pump unit in a) heating mode and b) cooling mode (after Wright & Colvin, 1993).
HE: heat exchanger; P: pressure.

3.2 Borehole Heat Exchangers

Energy wells are boreholes through which heat is exchanged, to or from the ground. The term refers to systems where the underground heat source or sink is groundwater from an aquifer (aquifer thermal energy storage, ATES), or ground in the form of hard rock or more or less consolidated sedimentary layers (borehole thermal energy storage, BTES) (Gehlin, 2002).

Borehole heat exchangers are vertical loops, which are generally considered when the available land area (foot print) is limited. Wells are bored to depths that typically range from 200 to 300 feet deep. The closed-loop pipes are inserted into the vertical boreholes (Figure 3.3). Typical requirements for piping range from 400 to 600 linear feet per system ton, depending on the soil and temperature conditions.

Vertical loops typically require one to two boreholes per ton of system load, depending on the soil thermal conductivity and borehole thermal resistance. To avoid long-term degradation of the thermal resource, boreholes should be spaced 15 to 20 feet apart, depending on the climate and soil conditions.

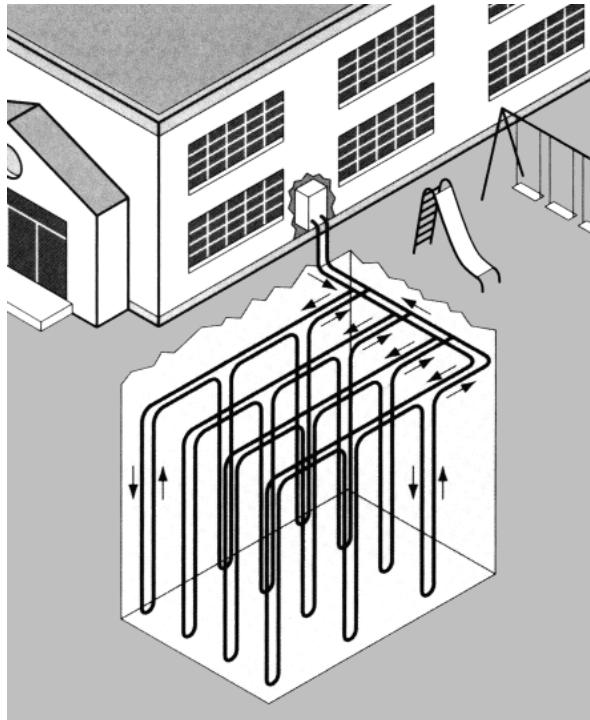


Figure 3.3 : Vertical ground loop configuration.

The most common configuration for the vertical loop-piping element in the drilled bore is a U-tube, where a 180° bend fitting has been factory fused to join two lengths of HDPE pipe, and inserted into the borehole (Figure 3.4).



Figure 3.4 : Insertion of vertical U-tube piping prior to backfilling or grouting.

Where the local water table is known to be reliably near the surface, the borehole can be backfilled with pea gravel, which allows groundwater circulation around the U-tube elements. Where the soil is dry or where there are large seasonal fluctuations in the

groundwater level, or where local regulations require permanent sealing of the borehole, a thermally enhanced grout should be used to backfill around the U-tube. Thermal performance also can be enhanced by the use of spacer clips at 5-foot intervals along the length of the piping element, which force the legs of the U-tube against the borehole wall.

Borehole thermal energy storage systems consist of several borehole heat exchangers (BHE). Applications where thermal energy is injected or extracted through the borehole with the use of heat pumps are commonly referred to as ground-coupled heat pump systems (GCHP), or ground-source heat pump (GSHP) systems. There are also feasible applications for ground heat exchangers, where heat pumps are not used, e.g. dissipative systems for direct cooling, or high temperature thermal storage for low-temperature applications (Figure 3.5). BHEs are boreholes of a diameter normally in the range 0.09-0.15 m, drilled in the ground to a typical depth of 30-200 m. A heat carrier fluid is circulated through the borehole, usually in a closed circuit, exchanging heat or cold from the ground to the user unit (Gehlin, 2002).

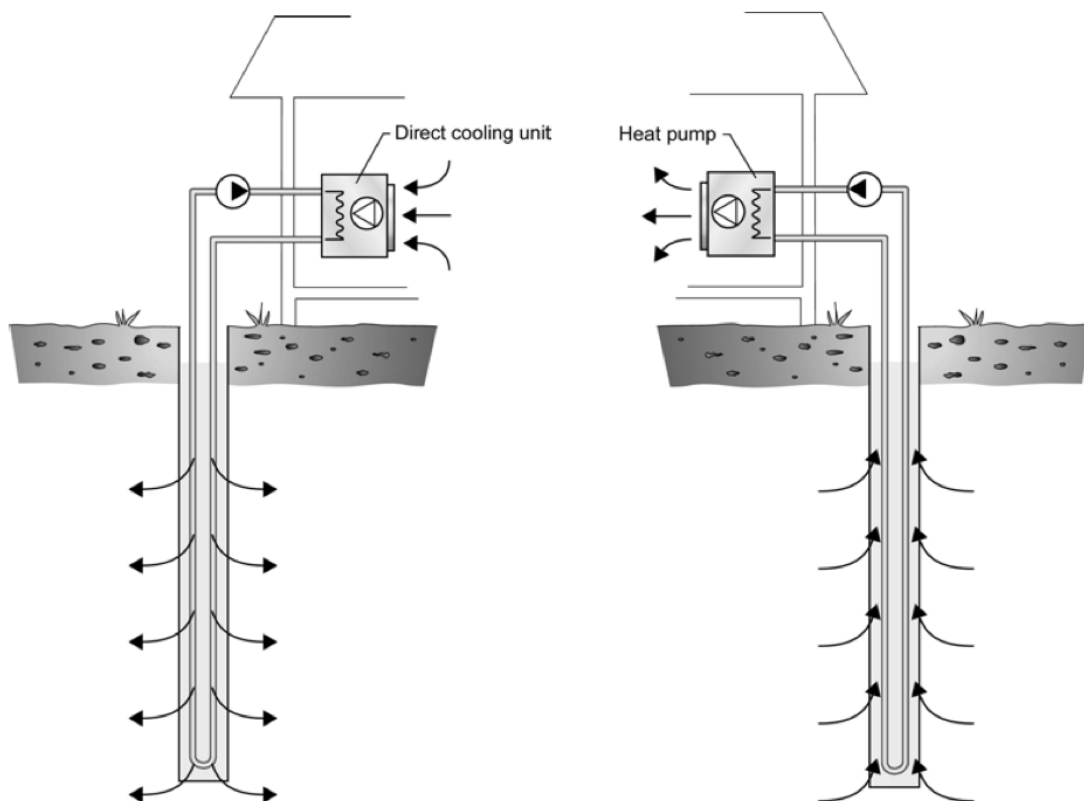


Figure 3.5 : Borehole heat exchanger in a dissipative application for direct cooling of electronic equipment and in a heat pump application for domestic space heating (Gehlin, 2002).

3.3 Thermal Conductivity Test

Proper design of borehole heat exchangers for space heating utilizing ground source heat pump systems requires a good estimate of the ground thermal properties in order to avoid significantly over-sizing or under-sizing the ground heat exchanger. The most important parameter is the thermal conductivity of the ground. Since the mid-80s, a method has been developed and refined to measure the underground thermal properties on site, and mobile equipment for these measurements has been built in several countries.

The idea of a field thermal conductivity test (TCT, also referred as thermal response test – TRT) is to determine the thermal properties of the subsurface and the borehole in situ. These parameters are required for the design and dimensioning of ground-coupled heat pump systems. Estimated properties are the thermal conductivity of the subsurface and the thermal resistance of the borehole. These properties are needed to calculate the required length of ground heat exchangers in a given application, which is the main cost associated with a ground-coupled heat pump system. An accurate estimation of thermal properties from TCTs therefore has a direct impact on the efficiency and costs of heat pump systems (Marcotte and Pasquier, 2008).

In-situ thermal conductivity testing was developed and initially introduced by Mogensen (1983), whose installation was designed as an immobile system. He suggested a system with a chilled heat carrier fluid being circulated through a GHE system at constant heat extraction (or cooling) rate, while the outlet fluid temperature from the GHE was continuously recorded. Later, in 1995, the first mobile thermal conductivity testing units were independently constructed in Sweden (Eklöf and Gehlin, 1996) and in USA (Austin, 1998). The set-up for a typical in-situ thermal conductivity test is shown in Figure 3.6.

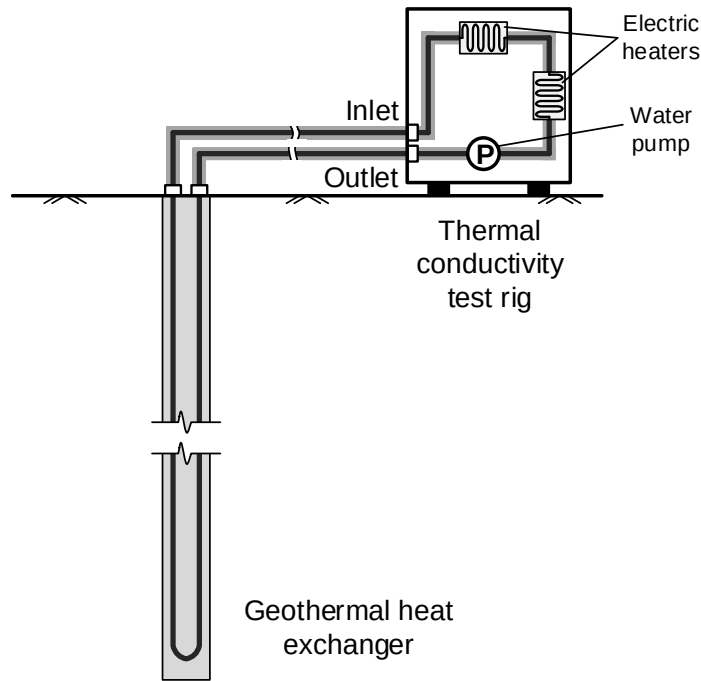


Figure 3.6 : Schematic view of the borehole thermal conductivity test.

Similar testing systems were later developed by other researchers (Mattsson et al., 2008; Georgiev et al., 2009). Several commercial units have been designed which fit into small (airliner-transportable) shipping containers. In the Netherlands, a large (housed in a shipping container) thermal conductivity test measurement unit housed in a cargo shipping container was constructed (Witte et al., 2002). It is operated with a reversible heat pump, and thus can be run in either heating or cooling mode. The heat pump generates a supply of warm or cold fluid, which is used to maintain a certain temperature difference between fluid entering and leaving the borehole. Various testing systems developed up till now are depicted in APPENDIX A.

In a thermal response test, a constant heat injection or extraction is imposed on a test borehole or pile with known depth and radius. By measuring the inlet and outlet temperatures of the heat carrier fluid during a period of time, the mean fluid temperature, T_f is determined for each time. The resulting temperature response can be used to determine the ground thermal conductivity, and to test the performance of GHEs.

The guidelines of thermal conductivity test for geothermal applications were developed by ASHRAE (Kavanaugh et al., 2001). The test is performed by circulating a carrier fluid within the geothermal loop integrated in the borehole, while heat is injected into the system at a constant rate. General recommendation is to apply 50 to

85 Watts of heating energy per meter length of the geothermal heat exchanger. This is carried out by arranging a set of heaters for the desired heat injection rate and a flow rate is selected in conjunction with the length of the geothermal loop. Ground temperature increases as heating energy is injected into the ground through the geothermal loop. Temperature of the circulating fluid is monitored over time along with the rate of energy injection. Progression of the inlet and outlet fluid temperatures is then used to evaluate thermal conductivity of the system using several analytical or numerical analysis methods.

There are a variety of analytical and numerical methods that can be used for analyzing thermal conductivity test data. These methods include infinite and finite line source (ILS/FLS) methods, cylindrical heat source (CHS) method, ORNL/Shonder method and the OSU numerical method (Kavanaugh, 2010). They typically result in similar thermal conductivity values for a given set of test results. The thermal conductivity obtained is a value for the total heat transport in the underground, as the thermal characteristics of the different soil/rock layers in the formation are collectively reflected in the test results and in the evaluated thermal conductivity. Other effects such as the convective heat transport (in permeable layers with groundwater) and further disturbances are automatically included, so it may be more correct to speak of an “effective” thermal conductivity. The configuration details of the GHE; such that, heat exchanger diameter, pipe diameter, shank spacing, number of loops and thermal properties of the backfill and the pipe materials do not have any effect on the estimated thermal conductivity. Different configurations result in different values of thermal resistance of the borehole or heat exchanger pile, which is an important factor in the evaluation of the GSHP system’s overall performance.

There are many possible sources of error when performing a TCT. These can be grouped in two categories:

- Underground influence (high regional groundwater flow, confined or artesian groundwater in combination with not or poorly grouted BHE, karst, etc.)
- Technical influence (fluctuations in heat input, sensor errors/failures, system leakage, poor thermal insulation in combination with solar irradiation or ambient temperature changes)

External influences caused by groundwater flow, power fluctuations and ambient temperature changes can be accounted for by using several techniques during the evaluation of the test data. These methods will be discussed in the following sections. However, there are practical cases where a meaningful test result cannot be obtained at all, due to several external influences.

The theoretical bases of the thermal conductivity test are presented by Hellström (1991), Gehlin (2002) and Kavanaugh and Rafferty (1997). Prior to explaining the test procedures and further related information, test equipment is introduced in the following section.

3.3.1 Test equipment

A TRT unit typically consists of a pump, purge valves, electric heating elements, temperature sensors, a flow meter, a data logger and a power supply. Some units also contain a heat pump for testing in cooling mode, but they are not widely used. The testing units come in various sizes, ranging from that of a suitcase to a trailer. Various testing systems developed up till now are depicted in APPENDIX A.

3.3.1.1 Power supply

A TRT unit needs power supply for operation. While larger units have their own generators, other units require to be hooked up to a generator or a power grid. It is important to have a steady power supply in order to prevent power fluctuations during testing.

3.3.1.2 Fluid supply tank

A fluid supply tank is used to fill the U-tube(s) in the borehole with the carrier fluid prior to testing. The tank itself can also be used to air purge the loops. A fluid supply tank typically has two ports at least, of which one is for inlet and the other for outlet. Inlet line is located on the bottom of the tank, whereas the outlet is located near the top of the tank. The tank may also have a separate drain line on the bottom to completely drain the system.

3.3.1.3 Air purging system

For a proper thermal response test, the ground loop and the piping of the testing equipment must be completely de-aerated. Various types of problems can appear in

poorly air purged systems, e.g. reduced pump capacity, reduced heat transfer, regulation problems, erosion and corrosion and leakage at seals (Melinder, 2009).

Air purging of the system is achieved by purge valves or with the help of the supply tank. The latter is the most common method in a thermal response test, and will be explained in Section 3.3.2.

3.3.1.4 Expansion tank

Expansion tank absorbs the expanding carrier fluid as its temperature increases due to heating, and limit the pressure within the system, thus preventing the pressure in the piping to exceed critical limits. The expansion tank uses compressed air to maintain system pressures by accepting and expelling the changing volume of water as it heats and cools.

3.3.1.5 Circulation pump

Circulation pump circulates the carrier fluid in a closed loop through the pipes in the testing unit and the U-tubes in the borehole. A separate pump can be used with the fluid supply tank in order to fill the ground loop and de-aerate the system. The circulation pump should be capable of generating the desired mass flow rates for a given borehole setup.

3.3.1.6 Heating/cooling unit

In a typical thermal response test application, heating mode is utilized. Accordingly, the test unit includes several electric heater elements. These heater elements can be of different rates (e.g. 1 kW, 1.5 kW, 2 kW, 4 kW, etc.), thus giving the flexibility of different ranges of power input to be applied on the carrier fluid. The total rate of heating elements in the test unit should be sufficient to provide the desired power for a given borehole setup. Another important thing to note is that, the heaters should be steady in order to generate a constant heat rate throughout the test. Instead of electric heaters, a TRT unit can also have a reversible heat pump, which allows the test to be run in cooling mode as well as heating mode.

3.3.1.7 Sensors and data acquisition apparatus

A TRT unit is equipped with temperature sensors, flow meters and watt transducers. For a minimum test setup, the inlet and outlet temperatures entering and exiting the ground loop, heat input of the heater elements and mass flow rate of the carrier fluid

are measured and logged with the corresponding date/time data at specified time intervals during the test.

For a more sophisticated analysis and for a better understanding of the soil profile being tested, additional data should be recorded. These may include fluid pressure in the loop, temperature recordings of the fluid along the ground loop, temperature recordings in the borehole and/or nearby soil by pre-installed instrumentation, exterior air temperature and ambient temperature in the test unit.

Each measuring element in the test unit should have backup unit to minimize errors due to a malfunction. In addition to data recording of each measuring element at the specified time intervals, the measurement system in the test unit should also allow the observation of the data real time.

3.3.1.8 Insulation

All the piping outside the borehole loop, including the pipes inside the test unit must be well insulated. The exterior exposed piping from the borehole to the test unit should be as short as possible, and for its insulation, highly resistant foams and round ducts with aluminum coating should be used to prevent heat loss/gain due to ambient temperature changes and irradiation from the sun.

3.3.2 Test procedures

In this section, testing procedures of a typical thermal response test, including the pre-test preparations (i.e. installation of a test borehole, determination of the undisturbed ground temperature) are explained.

3.3.2.1 Construction of the test borehole

The first step is to drill the test borehole until the desired depth. The construction method best suited to the type of geology can vary with different driller contractors and with the type of equipment used. After drilling is complete, piping of the borehole heat exchanger is installed by lowering down the hole. For ease of installation and to benefit from the weight of water, the pipes are prefilled with water. This helps keeping the pipes stationary, especially during pouring of the filling material. The use of spacers between the legs of U-bends keeps the legs apart from each other and close to the borehole wall. In order to complete the closed loop borehole, placing a filling

material in the space between the heat exchanger pipes and the borehole wall is required. Grout, cement or sand may be used to as the borehole material.

Grout is a high solids fluid mixture of cement or bentonite of a consistency that can be forced through a pipe and placed as required. Various additives, such as sand or hydrated lime, may be included in the mixture to meet certain requirements. For example, sand is added when a considerable volume of grout is needed. The reason fill is used is to achieve greater heat transfer than grout can provide. Fill is the use of cuttings or other materials that can be placed in the borehole under site-specific conditions, such as void zones or dry boreholes, or single, non-flowing aquifer (McCray, 1997).

The reasons for grouting are: (1) protection of the aquifer, or aquifers, including limiting the potential for water movement between aquifers, for purposes of maintaining quality or preserving the hydraulic response of the producing zone(s), (2) provide thermal contact between the loop piping and the formation (borehole wall), and (3) accomplishing these two in an efficient and economical manner.

High solids bentonite grouts, the most common grout material, are extremely beneficial as a closed loop borehole sealant, except where drying out and washing away cannot be prevented. Bentonite drilling fluids are not an acceptable grout. As engineered products, bentonite grouts must be carefully mixed and installed according to the manufacturer's specifications (McCray, 1997).

Parameters such as; diameter of the borehole, borehole filling material, pipe size, pipe material, pipe spacing, number of U-bends and their location in the borehole affect the borehole thermal resistance, thus the performance of the heat exchanger. Thermally enhanced grouts and pipes can be used to increase the performance of the heat exchanger.

Although it is recommended that the pipes should be filled with water prior to installation of the U-bend, the freezing of water inside the pipes must be prevented if there is such risk. It is also important to keep the open ends of the pipes sealed during and after the construction of the borehole to prevent entrance of debris in the piping.

3.3.2.2 Air-purging of the ground loop

Air purging of the ground loop is achieved by the use of the fluid supply tank. A closed loop is formed between the ground loop, piping of the TRT unit and the tank using

quick coupling connections. If the ground loop is not pre-filled with the fluid, filling process is also performed in this step.

Carrier fluid exits the tank via fluid supply line, circulates through the borehole loop and the piping of the testing unit, and enters the tank via the return line. The fluid supply line is located on the bottom of the tank, ensuring the pump to draw fluid without air bubbles. The return line is near the top of the tank, which allows any air in the fluid purged from the borehole or the piping system inside the unit to bubble out the top portion of the tank. Returning carrier fluid to the top of the tank minimizes the air bubbles in the fluid being drawn out of the bottom of the tank (Austin III, 1998). The air-purging process is continued until the fluid entering the tank is air bubble free.

3.3.2.3 Determination of the undisturbed ground temperature

When measuring the undisturbed ground temperature, the borehole should be in thermal equilibrium with the surrounding ground. Since the ground temperature is disturbed because of drilling and grouting/filling processes, after the construction of the borehole, there is a waiting period before the test can be initiated. According to ASHRAE (2009), this delay is 3 to 5 days, depending on the thermal conductivity of the subsurface. A waiting period of five days is recommended for low-conductivity soils ($k < 1.0 \text{ Btu/hr-ft-}^\circ\text{F}$ ($1.7 \text{ W/m-}^\circ\text{C}$)) and a delay of three days is recommended for higher conductivity formations ($k > 1.0 \text{ Btu/hr-ft-}^\circ\text{F}$ ($1.7 \text{ W/m-}^\circ\text{C}$)) (Kavanaugh et al., 2001).

At the end of the waiting period, undisturbed ground temperature is determined before the system is thermally loaded. This can be achieved by various methods. One way to measure the initial ground temperature is performed by direct insertion of a probe inside the liquid filled ground loop and recording the temperatures at various locations along the length of the ground loop representing the average.

The most common method to determine the undisturbed ground temperature is running the circulation pump to allow the carrier fluid to flow through the ground loop and the test unit without switching the heaters on. During the circulation, inlet and outlet temperatures of the ground loop are recorded and as the temperatures stabilize, the undisturbed temperature of the ground will be acquired. The piping of the test unit and the ground loop should be air-purged before temperature recording.

Even though heater elements are switched off during this period, there will always be some heating of the water caused by the pump work. The rate of this heating is related to the type of pump and the flow rate of the carrier fluid. Gehlin and Nordell (2003) showed that, the contribution of a circulation pump to the initial ground temperature could be as high as 2°C in 1 hour of circulation period. This test was performed on a borehole of 152 mm diameter and 60 m length, with a circulation period of 77 minutes (Figure 3.7).

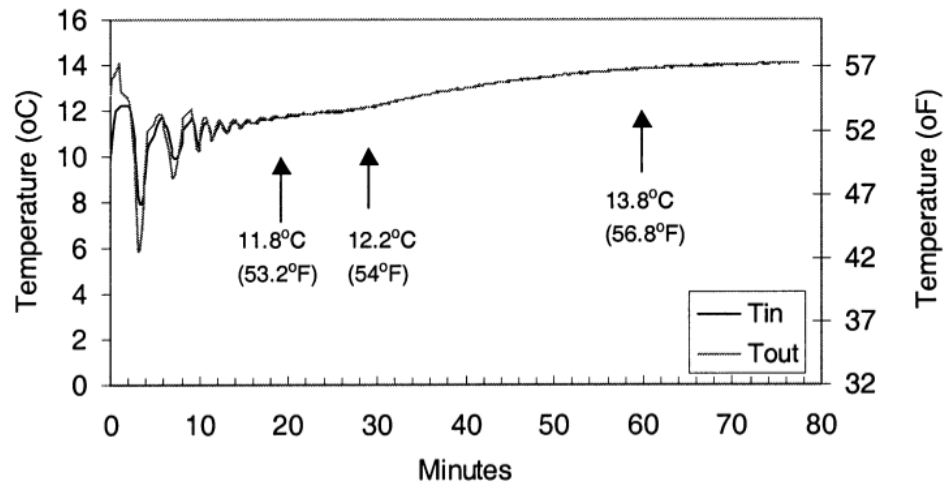


Figure 3.7 : Ten second interval temperature loggings in the borehole during measurement of undisturbed ground temperature period (Gehlin and Nordell, 2003)

3.3.2.4 Thermal loading period

After the whole piping system is completely air-purged and the undisturbed ground temperature is measured, the borehole is ready for thermal loading. In this step, the connections to and from the supply tank are disconnected or closed by corresponding valves. The circulation pump is running and the electric heater elements are switched on to generate the specified power input.

The carrier fluid is circulated in the closed system, and on its way, it passes through the heaters, which injects constant heat into the system. As the fluid flows through the temperature sensors at the inlet and outlet pipes, the temperatures are recorded by the data logger. Date, time, input power, inlet and outlet temperatures and other parameters specified in the previous sections are logged at selected time interval during the thermal loading period.

Although there is no guideline for the duration of thermal loading, various durations have been recommended, ranging from 36–48 hours (ASHRAE, 2009), 50 hours

(Austin III et al., 2000) to 60 hours (Eklöf and Gehlin, 1998). Witte et al. (2002) performed tests over 250 hours for research purposes, while their commercial tests are 50 hours in length. Longer tests are more expensive than shorter ones and practitioners therefore tend to conduct shorter tests. The disadvantage of shorter tests is that they can underestimate the subsurface thermal conductivity and lead to a conservative design estimate, which can overestimate the length of the ground heat exchanger needed for a ground-coupled heat pump system, and therefore increase costs.

3.3.2.5 Thermal recovery period

Thermal recovery period is the duration between the end of thermal loading and the time at which borehole/ground reaches thermal equilibrium (i.e. undisturbed ground temperature) again.

The test duration can further be increased to measure the water temperature recovery after heat injection is stopped. In that case, water circulation is maintained in the heat exchanger once heat injection stops, and temperature is measured until it approaches the initial background temperature. The flow rate is monitored during recovery to compute the heat injection rates and to verify if external heat transfer affects temperature, such as during a circulation test used to measure the undisturbed subsurface temperature (Raymond et al., 2011).

3.3.3 Operational problems and considerations

Operational experiences of the test units have shown some sources of error that can affect the results. These include heat leakage to or from the air, fluctuations in electrical power, and inaccurate measurements of the undisturbed ground temperature.

3.3.3.1 Heat losses or gains

Uncontrolled heat losses or gains to or from the environment due to insufficient thermal insulation cause problems (Austin, 1998; Witte et al., 2002) in the analysis of the experimental data. Even though the heat transfer to or from the environment may be relatively small compared to the heat transfer to or from the earth, it can have a significant adverse influence when the results are analyzed with the line source method. This problem may be overcome by adequate insulation of the experimental apparatus and piping. In systems where the injected/extracted heat is determined by measuring the inlet and outlet fluid temperatures and flow rate, moving the

temperature sensors into the piping in the ground (Witte et al., 2002) may also help. It is helpful to measure ambient air temperatures during the test so that the effects of changing ambient air temperature may be investigated. It may be possible to correct for these effects with some analysis procedures if a good estimate of the heat loss or gain can be made.

The issue of ambient temperature effect on the thermal conductivity test results has been addressed by Bandos et al. (2011) and a methodology for calculating the actual temperatures has been proposed.

3.3.3.2 Power stability

A common problem is fluctuations in the electrical power supply (Austin, 1998). This can cause problems with line source analysis, which usually assumes a constant heat injection rate. A recommended solution reported in the USA (Ewbanks, 1999) is to use a significantly oversized generator (e.g. a 50 kW generator for a 5 kW load), which should maintain a relatively constant power. Another solution is to control the temperature difference directly, while maintaining a constant flow rate or to control the temperature difference while measuring the flow rate, so as to maintain a constant heat injection or extraction rate. This approach has been utilized by Groenholland (Witte et al., 2002). A third solution is to use an analysis procedure that can account for fluctuating power.

3.3.3.3 Ground temperature

All analysis procedures depend on the ground being thermally undisturbed. The ground is necessarily disturbed by the drilling process, which may result in the ground surrounding the borehole being warmer (due to energy input or exothermic heating with cementitious grouts) or wetter (due to circulation of drilling fluid) or dryer (due to circulation of air) than it would otherwise be. The time required for the ground to return to an approximately undisturbed state has not received enough systematic study. Kavanaugh (2000) recommends that a thermal response test be delayed at least 24 hours after drilling, and at least 72 hours if cementitious grouts are used. Earlier work by Lilja (1981), Bullard (1947), Lachenbruch and Brewer (1959) might also be helpful in determining temperature disturbances caused by drilling.

3.3.3.4 Influence of variations in thermal conductivity with depth

For the analysis of a thermal response test it is normally assumed that the ground thermal conductivity along the borehole is homogeneous. However, there is normally a different top-soil layer with a considerably lower thermal conductivity than the deeper rock or sediments. According to Eskilson (1987), a numerical simulation of a deep borehole in granite ($k = 3.5 \text{ W/m}$, $K = 2 \text{ Btu/hr-ft-F}$) with a 5 m thick top-soil layer ($k = 1.5 \text{ W/m}$, $K = 0.9 \text{ Btu/hr-ft-F}$) shows that the thermal performance changes less than 2% for a 100 m (328 ft) deep borehole. His conclusion is therefore that the effect of a top-soil layer of less than 10 m (33 ft) can be neglected.

This may be further complicated by a difference in conductivity above and below the static groundwater level. The thermal response test naturally gives an aggregate value of all the layers. Some insight into the variation of conductivity with depth may be obtained by measuring the temperatures along the borehole after the test. (Witte 2001) In the case of a heat rejection test, areas of the ground with higher conductivities will have lower temperatures, and areas with lower conductivities will have higher temperatures.

3.3.3.5 Groundwater flow

The influence of groundwater flow on the performance of borehole heat exchangers has been a topic of discussion. Field observations have suggested that there is a groundwater aspect on the borehole performance (Gehlin 1998, Helgesen 2001). Some theoretical studies have been published on the subject. Eskilson (1987), Claesson & Hellström (2000) and Chiasson et al (2000) presented models for the influence of regional groundwater flow based on the assumption that the natural groundwater movements are reasonably homogeneously spread over the ground volume. This applies well on a homogeneous and porous ground material. Eskilson and Claesson & Hellström use the line source theory for modeling the groundwater effect on a single vertical borehole. They conclude that under normal conditions, the influence of regional groundwater flow is negligible.

Chiasson et al. (2000) use a two-dimensional finite element groundwater flow and mass/heat transport model and come to the conclusion that it is only in geologic materials with high hydraulic conductivities (sand, gravels) and in rocks with secondary porosities (fractures and solution channels in e.g. karst limestone), that

groundwater flow is expected to have a significant effect on the borehole performance. Simulations of the effect of groundwater flow on thermal response tests give artificially high conductivity values.

The influence of single or multiple fractures and fracture zones has not been thoroughly studied, and may give some explanation to field observations where groundwater flow has occurred.

3.3.4 ASHRAE recommendations

The ASHRAE recommendations for performing a thermal response test are listed below (Kavanaugh et al., 2001).

- Test durations of 36 to 48 hours are recommended. In some cases shorter periods were adequate. However, the recommendation is due to the high frequency of data sets that required longer tests for convergence. Information collected during the project indicates that longer tests do not significantly impact total cost.
- Acceptable power quality can be obtained when the standard deviation $< 1.5\%$ of average power and the maximum variation (spikes) $< 10.0\%$ of average power. When the deviations are larger, acceptable results can be obtained if the maximum deviation of the average loop temperature is $< 0.5^{\circ}\text{F}$ (0.3°C). The heat rate should be 15 to 25 W/ft (50 to 80 W/m) of bore depth, in order to reproduce the impact of typical loads on actual loops.
- A minimum delay of five days between loop completion and test start-up is recommended in formations that are expected to have low conductivity [< 1.0 Btu/h-ft- $^{\circ}\text{F}$ (1.7 W/m- $^{\circ}\text{C}$)] and three-days for other formations. A twelve-day delay is recommended before retesting a borehole if a complete 48-hour test has been conducted. An alternative recommendation is to delay retesting until the loop has returned to within 0.5°F (0.3°C) of the original undisturbed temperature.
- The initial ground temperature measurement should be made at the end of the waiting period by direct insertion of a probe inside a liquid filled ground heat exchanger at three locations representing the average or by the measurement of temperature as the liquid exits the loop during the period immediately following start-up.

- At a minimum, the measurements required are initial ground, loop inlet, and outlet liquid temperatures [$+0.5^{\circ}\text{F}$ (0.3°C) accuracy], input power to heating elements and pump (2.0% accuracy of reading), and ground heat exchanger length (+1%). Suggested information for higher quality tests include liquid flow rate (5% accuracy), a drilling log, diameter of bore, depth of bore, tube dimensions and type, grout (or fill) specifications and amounts, a record of drilling fluid additives used, and a description of surface casing.
- Water flow rates that result in 6 to 12°F (3 to 7°C) differential are recommended, as this is the temperature differential for an actual heat pump system. Connections from the test apparatus and the loop should be short and well insulated.

3.4 Parameters of the Subsurface and the Borehole

3.4.1 Thermal conductivity of the ground

When performing a thermal response test, the main goal is to determine the thermal conductivity of the ground. The value of thermal conductivity given by the test will be a mean value for the subsurface around the borehole, and represents the effective thermal conductivity that includes the influence of fractures, groundwater flow, bedrock, etc.

3.4.2 Heat capacity of the ground

When evaluating a thermal response test, it is necessary to assume a heat capacity, to be able to calculate the thermal conductivity and the thermal resistance.

3.4.3 Undisturbed ground temperature

Thermal response test makes it possible to determine the undisturbed ground temperature. This value will show the actual mean temperature in the borehole when the test is performed and will include the effect of the geothermal gradient and the current conditions on the ground surface. The question is whether the obtained value for this temperature will be the annual mean value needed for dimensioning of a borehole heat exchanger or not.

For a deep well, it will not make a difference whether the measurement is made during winter or summer. In shallower wells, the season might make a difference in regions with high annual differences in the air temperature (Eklöf and Gehlin, 1996).

3.4.4 Borehole thermal resistance

The value of the thermal resistance evaluated by thermal response test only applies to that certain borehole installation, heat carrier fluid, state of flow and borehole filling material. However, by using the response test it is easy to change the different parameters and determine how each property affects the thermal resistance. Several methods to estimate the borehole thermal resistance are discussed in Section 3.6.

3.5 Analytical Models

There are several analytical methods for the analysis of thermal response test data. However, there is no analytical solution for the three-dimensional transient heat transfer problem of the complex geometry and boundary conditions found in borehole heat exchanger systems. In order to derive analytical models for BHEs, a number of simplifying assumptions need to be made. The most widely used method is the infinite line heat source method, due to its simplicity and reliability. Other commonly used analytical methods are the finite line heat source and the cylindrical heat source methods.

The simplest one is to consider the whole borehole as an infinitely long line source and ignore the thermal properties of the borehole, for example, the thermal mass of the fluid, pipes, and grout. By making these assumptions, the line source theory propounded by Lord Kelvin (Thomson, 1884) can be applied to develop the so called the infinite line source model. A similar model, which is called finite line source, was derived by Zeng et al. (2002) to consider the finite length of the heat exchanger, thus accounting for the end effects. The cylinder source solution (Carslaw and Jaeger, 1959) is also similar but makes the assumption that the two legs of the U-tube can be considered as a single pipe that is co-axial with the borehole. The thermal mass of the fluid, pipes and grout are also ignored using this approach.

Analytical models allow computationally efficient calculation of BHEs. Their applications include designing the BHEs and analyzing in-situ ground conductivity testing data.

3.5.1 Infinite line heat source

The infinite line heat source (ILS) model is a direct application of Lord Kelvin's (Thomson, 1884) heat source approach to ground heat exchangers. This model relies on the fact that any line can be treated as numerous points positioned next to each other. Thus, it starts with calculating the temperature change at a specific location in an infinite homogeneous medium based on the formula of point heat source, and then using integration, accounts for the effect of other point sources at that desired location (Figure 3.8).

The following general assumptions were made for the derivation of this model:

1. The ground is regarded as a homogeneous medium and its thermophysical properties do not change with temperature,
2. The medium has a uniform initial temperature,
3. The heating rate per unit length is constant from the starting instant, up to the end of thermal loading.

The infinite line heat source model is the analytical solution to the equation of conductive heat transfer in a homogeneous and isotropic medium, expressed in radial coordinates as (Carslaw, 1945):

$$\frac{\partial^2 T}{\partial r^2} + \frac{1}{r} \frac{\partial T}{\partial r} = \frac{\rho c_p}{k} \frac{\partial T}{\partial t} \quad (3.1)$$

where,

T : temperature of the medium,

r : radial distance from the heat source,

ρ : density of the medium,

c_p : specific heat capacity of the medium

k : thermal conductivity of the medium,

t : time.

To solve the equation, it is assumed that the system is initially at a constant temperature ($T(r, t=0) = T_0$) and that boundaries located at an infinite distance from the heat source ($r=\infty$) always remain at a constant temperature ($T(r=\infty, t) = T_0$). The surrounding medium is considered homogenous and isotropic.

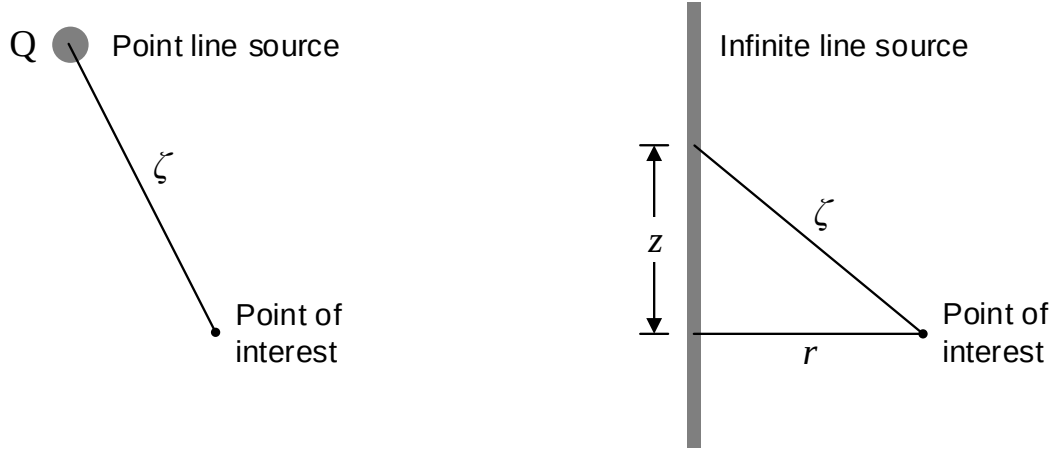


Figure 3.8 : Point heat source and infinite line heat source (ILS).

Ingersoll and Plass (1948) showed that the temperature change at time t and at a distance ζ from a point heat source emitting constant heat of q can be calculated as follows:

$$\Delta T(t) = \left(\frac{q\alpha}{k} \right) \left(\frac{1}{2\sqrt{\pi\alpha t}} \right)^3 \exp(-\zeta^2/4\alpha t) \quad (3.2)$$

The following integration is then used to estimate the temperature change due to an infinite line heat source. Again, the line-source is represented by adjacent points which for each one of them the distance to the location of interest are $\zeta = \sqrt{r^2 + z^2}$ as shown in Figure 3.8.

$$\Delta T = \int_{-\infty}^{\infty} \left(\frac{q\alpha}{k} \right) \left(\frac{1}{2\sqrt{\pi\alpha t}} \right)^3 e^{-(r^2+z^2)/4\alpha t} dz = \frac{q}{4\pi k t} e^{-r^2/4\alpha t} \quad (3.3)$$

As the heat source is emitting continuously from time 0 to t , the change of temperature at the point of interest can be estimated as:

$$\Delta T = \frac{q}{4\pi k} \int_0^t \frac{e^{-r^2/4\alpha t}}{t} dt \quad (3.4)$$

Assuming an infinite line heat source located at ($r=0$) and represented by a constant heat flux per unit length, and for the prescribed initial and boundary conditions, the solution to Equation (3.4) can be expressed as the following temperature increment ($\Delta T(r, t) = T(r, t) - T_0$) that depends on the distance r from the source and the time t :

$$\Delta T(r, t) = -\frac{q}{4\pi k} \text{Ei} \left(-\frac{r^2}{4\alpha t} \right) \quad (3.5)$$

where,

$E_i(u)$: exponential integral,

q : heat injection rate per unit length of borehole (heat flux).

The exponential integral in Equation (3.5) can be derived as the following Taylor series expansion (Abromovitz and Stegun, 1964).

$$E_i(u) = \gamma + \ln(u) + \sum_{i=1}^{\infty} \frac{u^i}{i \cdot i!} \quad (3.6)$$

where, γ is the Euler–Mascheroni constant ($\gamma=0.57721\dots$).

For large values of $\alpha t/r^2$, $E_i(u)$ can be approximated as:

$$E_i(u) = \gamma + \ln(u) \quad \text{for} \quad \frac{\alpha t}{r^2} \geq 5 \quad (3.7)$$

Using Equation (3.7), the temperature increment at the borehole wall at time t can be estimated as:

$$\Delta T(r_b, t) = \frac{q}{4\pi k} \left[\ln\left(\frac{4\alpha t}{r_b^2}\right) - \gamma \right] \quad (3.8)$$

The relationship between the measured mean fluid temperature (T_f) during a thermal response test and the temperature at the borehole wall is:

$$T_f(t) = T_0 + qR_b + \frac{q}{4\pi k} \left[\ln\left(\frac{4\alpha t}{r_b^2}\right) - \gamma \right] \quad (3.9)$$

where,

T_0 : initial (undisturbed) subsurface temperature,

R_b : borehole resistance.

Equation (3.9) can be represented in the form $T_f(t) = m \cdot \ln(t) + n$ to determine the thermal conductivity of the subsurface. In this way, using the slope of the mean fluid temperature development versus the natural log of time curve, thermal conductivity can be evaluated. The slope of the curve gives:

$$m = \frac{q}{4\pi k} \quad (3.10)$$

Thermal conductivity can be evaluated using this slope:

$$k = \frac{q}{4\pi m} \quad (3.11)$$

ILS model does not represent the limited length of general heat exchangers. The limited length causes the edge effect, which is a reduction in the temperature change at the edges of the line. This axial effect is found to be significant for time $t \approx H^2/180\alpha$ for a single line source, where H is the length of the heat exchanger. This limiting time is even less if there is an interaction between different line sources (Lamarche and Beauchamp, 2007).

Representing the heat exchanger as a line heat source does not provide accurate results in short time periods and for the case of large diameter vertical heat exchangers such as heat exchanger piles.

3.5.2 Finite line heat source

Finite line source model (FLS) is a special case of the line source model, which is a direct application of Lord Kelvin's point heat source approach to ground coupled heat exchangers (Thomson, 1884). It relies on the principle of superposition where a line consists of a combination of sequentially positioned points. Thus, it starts with calculating the temperature change at specific location in an infinite homogeneous medium based on the formula of point heat source, then integrating along a line to account for the effect of point sources positioned along the line. The model is called infinite line source (ILS) model when the integral of the point source formula is performed on a line with an infinite length. When a line source with a finite length is considered, the model is called finite line source model, which is used to account for the end effects of the GHE.

Zeng et al. (2002) developed an approach to estimate the temperature changes in a homogenous semi-infinite medium induced by a finite line source. The model accounts for the end effects of the GHE by including a virtual line sink with the same length as the GHE but with a negative heating rate $-q$ on symmetry to the ground surface boundary (Figure 3.9). This ensures that, the temperature at the ground surface is kept constant throughout the period considered.

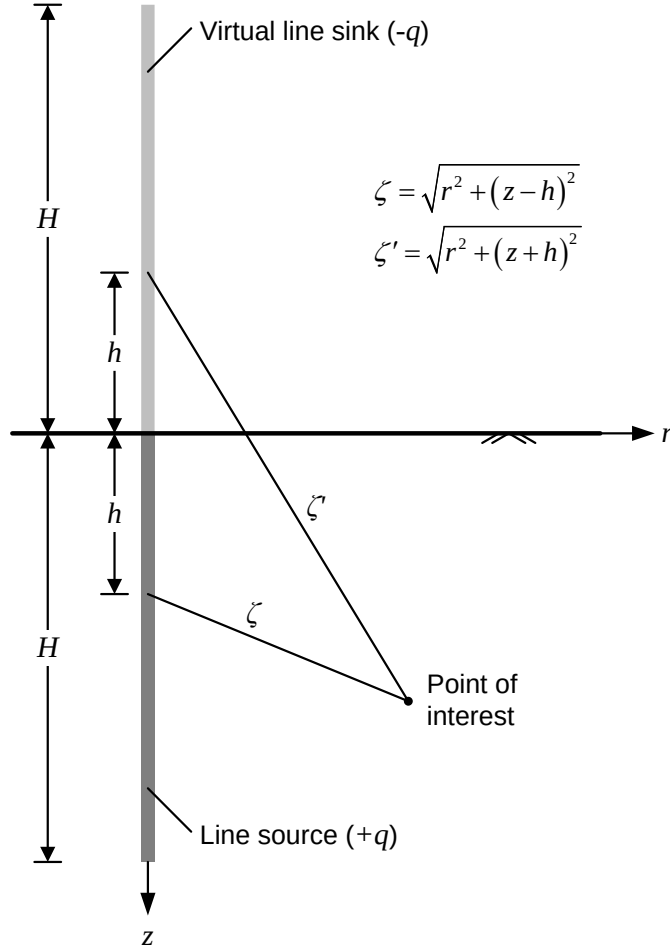


Figure 3.9 : The geometry of the finite line source model (after Zeng et al., 2002).

The FLS model can be derived by integrating Equation (3.2) over the length of the heat exchanger and including the identical yet opposite imaginary source. The final form of the FLS model equation is given as:

$$\Delta T(t) = \left(\frac{q}{4\pi k} \right) \int_0^H \left[\frac{\text{erfc} \left(\frac{\sqrt{r^2 + (z-h)^2}}{2\sqrt{\alpha t}} \right)}{\sqrt{r^2 + (z-h)^2}} - \frac{\text{erfc} \left(\frac{\sqrt{r^2 + (z+h)^2}}{2\sqrt{\alpha t}} \right)}{\sqrt{r^2 + (z+h)^2}} \right] dz \quad (3.12)$$

This relation represents the general FLS model. This model has the same limitations of the ILS model except that it considers the edge effects as well as the finite homogeneous medium.

3.5.3 Cylinder heat source

The cylinder source model approach models the borehole heat exchanger as an infinite cylinder with a constant heat flux through U-pipe exchangers. Considering the two legs of the U-pipe as one infinitely long pipe, and coaxial with the borehole with

infinite length, the cylinder source solution (Carslaw and Jaeger, 1959) can be used to calculate the temperature distribution of the ground surrounding the borehole at any time with the knowledge of the initial ground temperature. The following equation describes the cylinder source solution:

$$\Delta T = \frac{q}{k\pi^2} \int \frac{e^{-\beta^2 Fo} - 1}{\beta^2 (J_1^2(\beta) - Y_1^2(\beta))} [J_0(\rho\beta)Y_1(\beta) - J_1(\beta)Y_0(\rho\beta)] d\beta \quad (3.13)$$

where,

Fo : Fourier number, $Fo = \alpha t / r_b^2$,

J_0 and J_1 : Bessel functions of the first kind of order 0 and 1, respectively

Y_0 and Y_1 : Bessel functions of the second kind of order 0 and 1, respectively,

ρ : dimensionless distance, $\rho = r / r_b$,

r_b : borehole radius.

Ingersoll et al. (1954) simplified the above form as follows:

$$\Delta T = \frac{q}{k} G(z, p) \quad (3.14)$$

where $G(z, p)$ is the cylinder source function. The solution of the function $G(z, p)$ was given in a tabulated form by Ingersoll et al. (1954). Hellström (1991) provided a correlation function of Fo to represent $G(z, p)$. His correlation is given as follows:

$$\ln(G) = -2.05227 + 0.36081 \ln(Fo) - 0.02392 \ln^2(Fo) + 0.00068 \ln^3(Fo) \quad (3.15)$$

The general equation for estimating the mean fluid temperature can be written as:

$$T_f(t) = \frac{q}{k} G(z, p) + qR_b + T_0 \quad (3.16)$$

Carslaw and Jaeger (1959) developed a model based on the cylinder source theory, in which solutions are obtained varying boundary conditions for regions bounded by cylinder geometry. The borehole heat exchanger is represented as a cylinder filled with a backfill material and the fluid temperature for large values of the time or a small radius can be determined with the following approach cylinder theory:

$$T(t) = T_0 + \frac{Q}{4\pi k_{\text{ground}}} \left[2h + \ln\left(\frac{4\zeta}{c}\right) - \frac{4h - a_1}{2\alpha_1\zeta} + \frac{\alpha_1 - 2}{2\alpha_1\zeta} \ln\left(\frac{4\zeta}{c}\right) + \dots \right] \quad (3.17)$$

where,

$$h = 2\pi k_{\text{ground}} R_b, \quad \alpha_1 = \frac{2\pi r_b^2 c_{p \text{ ground}}}{c_{\text{cyl}}}, \quad \zeta = \frac{\alpha_{\text{ground}} t}{r_b^2} \quad \text{and} \quad C = e^\gamma$$

Deerman and Kavanaugh (1991), and Kavanaugh and Rafferty (1997) developed a methodology based on the cylinder source theory that uses an iterative procedure, in which, the effective thermal conductivity is calculated by means of the reverse process used to obtain the length of the ground heat exchanger. The thermal conductivity and diffusivity are introduced in a dimensionless cylinder-source function to obtain the thermal resistance, which is compared with its measured value of a short-term in-situ test.

3.6 Thermal Resistance Models

Several steady state geothermal heat exchanger models have been developed mainly for calculating the GHE thermal resistance. The fundamental assumption made in those models is that the heat transfer in the GHE is in steady state. The ratio between the heat flux of the GHE and the temperature difference between the fluid and the GHE wall is then a constant value. Thus, a thermal resistance can be used to define the relationship between the heat flux and the temperature difference of the fluid and the GHE wall as follows:

$$R = \frac{T_f - T_{\text{wall}}}{q'} \quad (3.18)$$

where R is the thermal resistance of the heat exchanger ($m \cdot K/W$), q' is the heat flux per unit length of GHE (W/m), T_f is the mean fluid temperature ($^{\circ}C$) calculated by averaging the inlet and outlet temperatures and T_{wall} is the average temperature of the heat exchanger wall ($^{\circ}C$).

The GHE thermal resistance includes the convective resistance of the fluid, the conductive resistance of the pipes, and the conductive resistance of the grout. For a classical arrangement with single U-tube GHE, the equivalent thermal resistance circuit is shown in Figure 3.10.

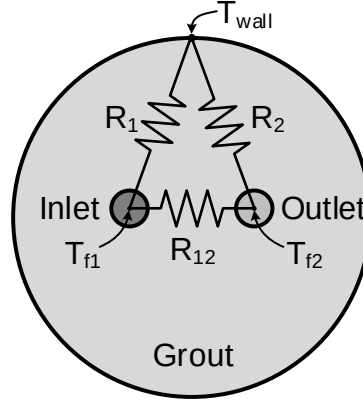


Figure 3.10 : Thermal resistance of a GHE integrated with single U-tube.

In the symmetric case, where each pipe is located at the same distance to the GHE wall, the borehole thermal resistance can be written as:

$$R_b = \frac{R_1}{2} = \frac{R_2}{2} \quad (3.19)$$

Although different methods are available to obtain the thermal resistance of the GHE, they all decompose it into three different contributions. The relationship of the borehole thermal resistance and the thermal resistance of fluid, pipe and grout can be defined as:

$$R = \frac{R_{pipe}}{2} + R_g = \frac{R_{conv} + R_{cond}}{2} + R_g \quad (3.20)$$

where R_{conv} is the convective resistance of the fluid within one pipe, R_{cond} is the conductive resistance of one pipe, R_{pipe} is the total resistance of one pipe (Figure 3.11) and R_g is the conductive resistance of the grout.

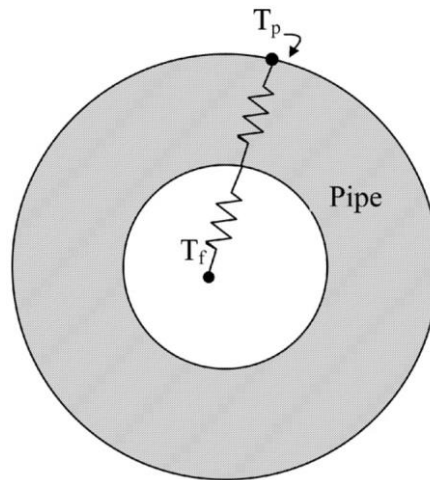


Figure 3.11 : Schematic view of pipe resistance.

The convective resistance of the fluid within one pipe can be calculated by (Bergman et al., 2011):

$$R_{conv} = \frac{1}{\pi d_{pi} h_i} \quad (3.21)$$

where d_{pi} is the inner diameter of the pipe (m), and h_i is the convective heat transfer coefficient of the carrier fluid ($W/m^2 \cdot K$).

The conductive resistance of one pipe can be calculated by (Bergman et al., 2011):

$$R_{cond} = \frac{\ln(d_{po}/d_{pi})}{2\pi k_p} \quad (3.22)$$

where d_{po} is the outer diameter of the pipe (m) and k_p is the thermal conductivity of the pipe ($W/m \cdot K$).

The conductive resistance of the grout cannot be calculated as straight forwardly as the thermal resistance of the fluid and the pipe due to its irregular geometry. A number of steady state borehole models have been developed to address this and are discussed below.

3.6.1 Equivalent diameter model

The equivalent diameter model developed by Gu and O'Neal (1998) illustrates a very simple way of calculating the borehole thermal resistance in one-dimensional steady-state conditions. Taking the assumption that the thermal influences of the two legs of a U-tube can be represented by the thermal influences of one pipe that is concentric with the borehole, Gu and O'Neal (1998) derived an expression to calculate the equivalent diameter of a concentric pipe (Figure 3.12).

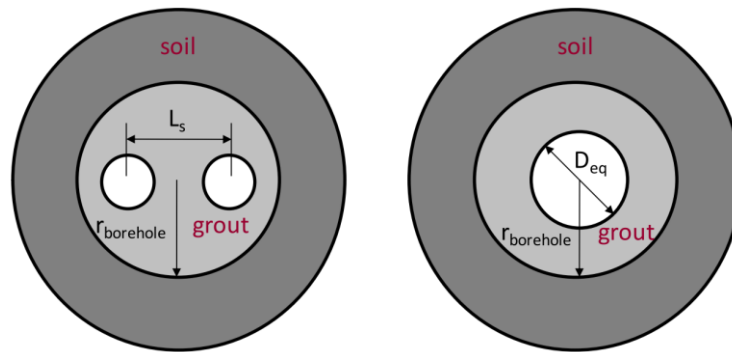


Figure 3.12 : Diagram of equivalent diameter of a BHE with single U-tube (Gu and O'Neal, 1998).

$$D_{eq} = \sqrt{2d_p L_s} \quad (3.23)$$

where D_{eq} is the equivalent diameter of the pipe (m), d_p is the diameter of the pipe, and L_s is the center to center distance between the two legs of the U-tube.

The resistance of the grout, in this case, can be calculated by employing the general equation for radial heat conduction through a cylinder (Bergman et al., 2011).

$$R_g = \frac{\ln(d_b/d_{eq})}{2\pi k_g} \quad (3.24)$$

where d_b is the diameter of the borehole (m) and k_g is the thermal conductivity of the grout (W/m·K).

3.6.2 Paul's model

Paul's model was developed to calculate the grout thermal resistance by using 'shape factor correlations', which were created using experimental data from a test apparatus and simulation results from a two dimensional finite element model (Paul, 1996).

Four different configurations classified according to the shank spacing S_1 and S_2 were taken into account in Paul's Model (Figure 3.13).

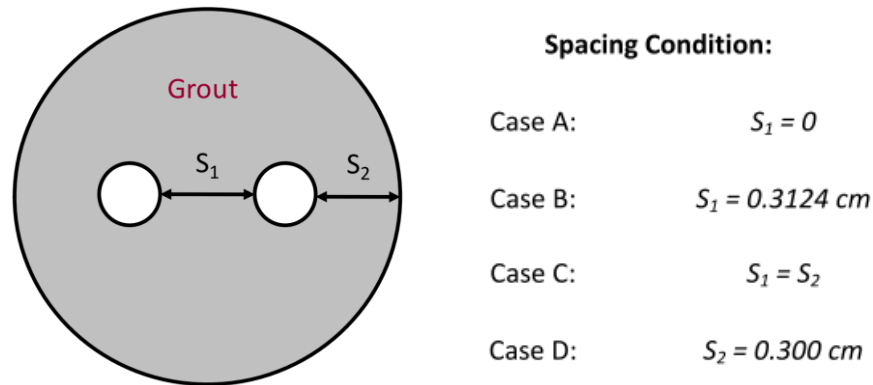


Figure 3.13 : Different shank spacing cases considered in Paul's model (Paul, 1996).

By adding the shape factor, the thermal resistance of the grout can be expressed as:

$$R_g = \frac{1}{S \cdot k_g} \quad (3.25)$$

where S is the dimensionless shape factor which is given as:

$$S = \beta_0 (d_b/d_{po})^{\beta_1} \quad (3.26)$$

where β_0 and β_1 are the dimensionless coefficients, obtained by best-fit estimates. For the cases shown in Figure 3.13, the values of the equation fit coefficients are given in Table 2.1. R indicates the accuracy of the equation fit with reference to the experimental data or numerical model. An R value of 1 indicates a perfect fit.

Table 3.1 : Equation fit coefficients given by Paul for four different cases (Paul, 1996).

Coefficient	Case A	Case B	Case C	Case D
β_0	14.4509	20.1004	17.4427	21.9059
β_1	-0.8176	-0.9447	-0.6052	-0.3796
R	0.9971	0.9926	0.9997	0.9699

3.6.3 Multipole method

The multipole method developed by Bennet et al. (1987) is probably the most rigorous method of calculating the borehole thermal resistance. It represents the pipes in the circular borehole using a series of line heat sources or sinks and can model borehole configurations with more than one U-tube. It can also calculate borehole resistance of U-tubes that are not symmetrically arranged in the borehole. It simulates each pipe by representing it with a line source or line sink, and for each line source or line sink, there is at least one line sink or line source at a mirror point (zero order multipole) (Figure 3.14). More line sinks or line sources are added to account for the non-concentric nature of the geometry (higher order multipoles). The solution can be found limited only by machine accuracy.

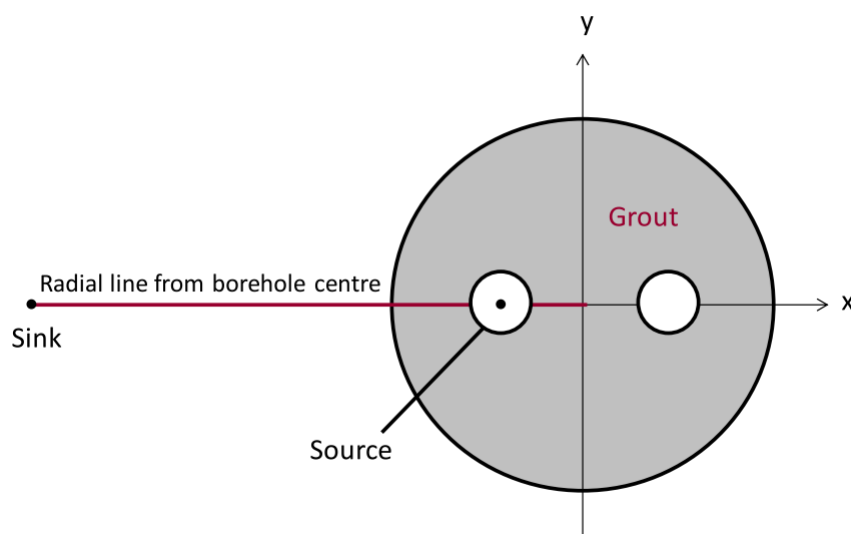


Figure 3.14 : A single line source with a single line sink.

By setting a specific temperature for each pipe, the multipole method calculates the heat flux of each pipe and the average temperature of the GHE wall so that the borehole resistance can be derived as follows:

$$R = \frac{T_f - T_{wall}}{\sum_{i=1}^n q'_i} \quad (3.27)$$

3.7 Response Factor Models

Although analytical models require less computing effort than numerical approaches and are suitable to be used in the analysis of in-situ thermal response test data, they are less suited to design and simulation tasks where one would like to take account of time varying heat transfer rates and the influence of surrounding boreholes over long timescales. Response factor models, on the other hand, have been developed to address these issues.

Response factor models use pre-defined response functions to describe the relationship of temperature changes and heat fluxes of BHE fields/arrays. Hybrid approaches combining analytical and numerical methods are usually adopted in order to derive response functions for pre-defined configurations of BHEs. Response factor models have been proved to be highly efficient and have been implemented in both design and simulation software.

3.7.1 Long time step g-function model

The long time step g-function model was developed by Eskilson (1987) to model BHEs for thermal storage applications. The g-function is essentially a normalized step-response function that describes the relationship between the average borehole temperature and a step in the extraction/rejection for a defined configuration of BHEs. It is calculated through a combination of analytical and numerical techniques. Firstly, a two-dimensional (radial-axial) finite difference model for a single BHE was constructed to determine its temperature response to a unit step heat pulse. After that, the superposition technique was applied to determine the temperature response to the unit step heat pulse for a pre-defined configuration of multiple BHEs using the temperature distribution of a single borehole obtained previously. These temperature

responses are then normalized to the so called g-function. The g-function is usually defined by discrete values that are interpolated in the simulation.

The mathematical expression of the g-function can be derived as follows: given the constant heat extraction rate q' (W/m) and constant far-field ground temperature T_s , the borehole temperature $T_b(t)$ can be written as:

$$T_b(t) = T_s - q' \cdot R_q \quad (3.28)$$

where R_q can be regarded as a time-dependent thermal resistance for a unit heat extraction step, and is written in the following way:

$$R_q = \frac{1}{2\pi k} \cdot g(t/t_s, r_b/H) \quad (3.29)$$

where g denotes the dimensionless step-response function, which depends on dimensionless time t/t_s and dimensionless distance r_b/H . t_s is the steady-state time-scale which is defined as:

$$t_s = \frac{H^2}{9\alpha} \quad (3.30)$$

where α is the thermal diffusivity of the ground (m^2/s).

Rearranging Equation (3.28) and Equation (3.29), the g-function can be expressed as:

$$g(t/t_s, r_b/H) = \frac{2\pi k \cdot (T_b(t) - T_s)}{q'} \quad (3.31)$$

Eskilson (1987) calculated over 200 g-functions for multiple BHEs arranged in different shapes, e.g. in a line, in a rectangle, and in a square, with different distance between two boreholes. The results for each configuration also vary according to the ratio of borehole distance and borehole length (B/H). Four examples are shown in Figure 3.15.

Any heat extraction/rejection varying according to time can be decomposed into a set of unit steps. Consequently, the response to each unit step can be superimposed to calculate the overall response to any heat extraction/rejection time series.

In this approach, the complex geometry of a borehole is simplified as a cylinder with a finite length and diameter. A single temperature is used to represent the borehole wall temperature. The thermal conductivities and capacities of all the different

materials inside a borehole, including fluid, pipes and grout, are neglected. For this particular reason, the solution of the equation is only valid for time to be greater than a few hours. As a result, this g-function is known as a long time step g-function.

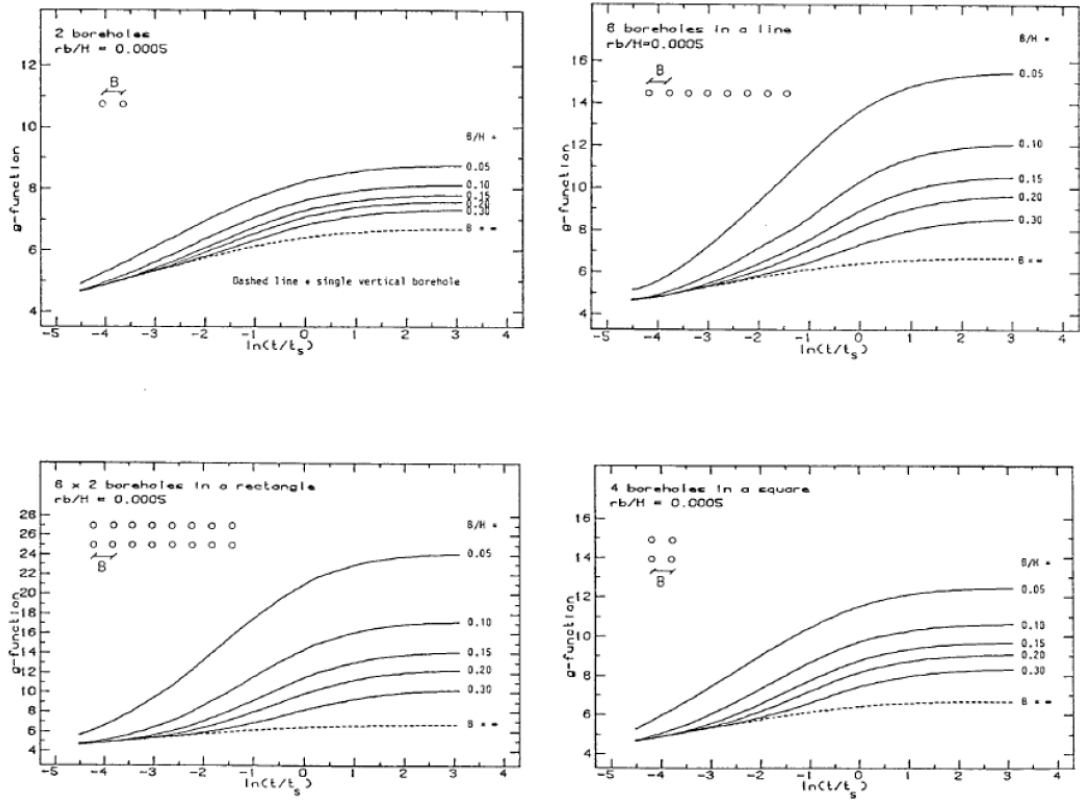


Figure 3.15 : Dimensionless g-function for different borehole configurations (Eskilson, 1987).

3.7.2 Short time step g-function model

The long time step g-function model developed by Eskilson (1987) provides an efficient solution to simulate borehole fields with defined configurations over long timescales, ranging from one month to several years. However, due to the simplification of borehole geometry and neglecting the thermal properties of all the components inside the borehole, this model is not suitable for simulation of BHEs on shorter timescales (e.g. hours), which is an essential requirement for system simulation tasks. In order to expand the application of the long time step g-function model, Yavuztürk (1999) developed a two-dimensional numerical model on a polar grid to compute the so-called ‘short time step g-function’.

Only half of the borehole was simulated due to its symmetrical geometry. The thermal properties of the materials inside the borehole were modeled, such as the resistance of the pipe and grout due to conduction and the convection resistance due to the flow of

the fluid inside the pipes. Figure 2.6 shows the discretization of the borehole geometry on a polar grid.

The pipe was not explicitly modeled but was approximated by cells in a pie-sector. In addition, the fluid inside the pipes was not explicitly modeled so that the heat transfer from the fluid was treated by a heat flux boundary condition at the pipe wall.

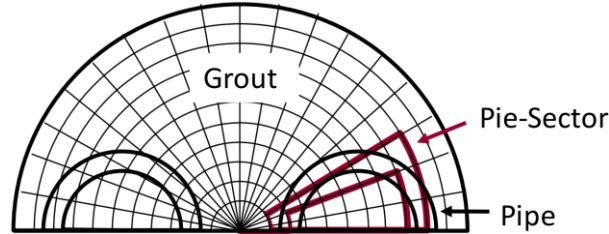


Figure 3.16 : Simplified representation of the borehole using the pie-sector approximation of the pipes (Yavuztürk, 1999).

Applying the finite volume method, the temperatures of the fluid can be calculated when there is a heat transfer rate applied to the borehole. Using a borehole resistance to describe the relationship between the fluid temperature and borehole temperature, the borehole temperature can then be subtracted from the fluid temperature.

$$T_b(t) = T_f(t) - q' \cdot R_b \quad (3.32)$$

where R_b can be calculated by Equation (3.18), which is a sum of the convective resistance of the fluid, calculated by Equation (3.19), the conductive resistance of the pipe, calculated by Equation (3.20) and the conductive resistance of the grout, calculated by Paul's method (Equation (3.25) and Equation (3.26)).

Consequently, Equation (3.31) is recast to solve for the g-function and modified to account for the borehole thermal resistance:

$$g(t/t_s, r_b/H) = \frac{2\pi k \cdot (T_f(t) - q' \cdot R_b - T_s)}{q'} \quad (3.33)$$

For typical ratios of borehole radius to borehole depth, the short time-step g-function data correspond to time steps between 2.5 minutes and 200 hours. The short time step g-function is implemented as an extension of the long time step g-function. The short time step g-function generated using the two-dimensional finite volume model extends the long time step g-function smoothly.

The model that combines the short time step g-function and the long time step g-function was implemented in GLHEPro (Spitler, 2000), a design software for vertical borehole ground loop heat exchangers, as well as in EnergyPlus (Crawley et al., 2001), a whole building energy simulation program. Figure 3.17 plots the imbedded short time step g-function and the long time step g-function for four different configurations of borehole fields in GLHEPro.

The biggest disadvantage of this model lies in the fact that the fluid inside the pipes was not explicitly modeled; instead it was treated by a heat flux boundary condition at the pipe wall. Therefore, the dynamics of the fluid transport along the pipe loop cannot be taken into account. In addition, the thermal mass of the fluid is not counted for.

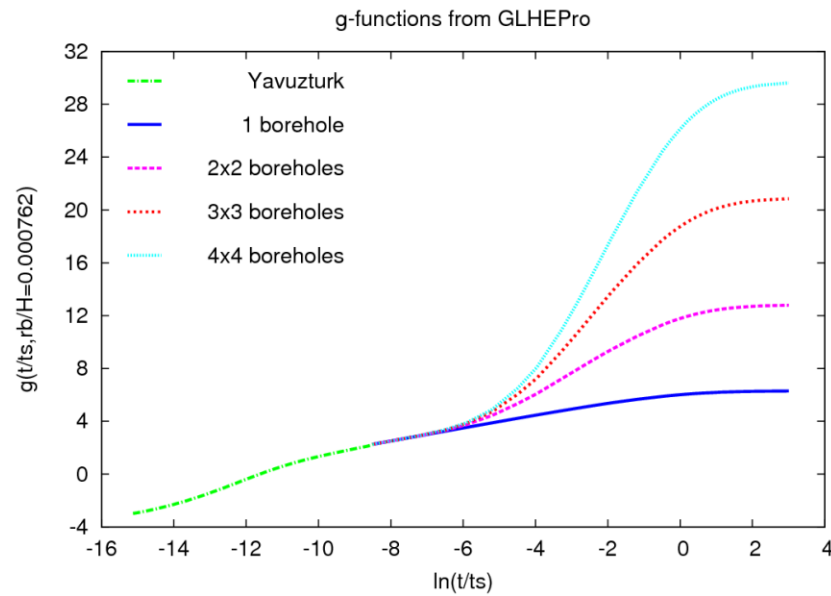


Figure 3.17 : The combined short time step g-function by Yavuztürk and long time step g-functions for four different configurations of borehole fields by Eskilson in GLHEPro.

3.7.3 DST model

Hellström (1991) developed the duct storage (DST) model for ground heat storage systems. The model divides the total volume of the ground for multiple BHEs into two regions: the local region and the global region. The local region is defined as the storage volume assigned to each individual borehole. Within the local region, there is a heat transfer process between the heat carrier fluid and the borehole wall, which is represented by a borehole resistance. In addition, the BHEs interact with the surrounding ground around each borehole. The global region, on the other hand, is concerned with the heat conduction problem between the bulk of storage volume (sum

of the storage volume assigned to each borehole) and the surrounding ground. Figure 3.18 shows a diagram of the division of the local and global region in the DST model.

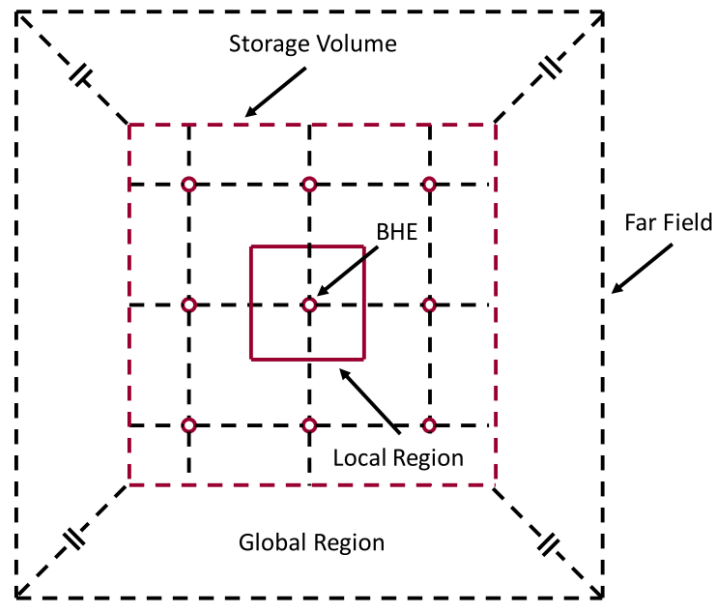


Figure 3.18 : The division of the local and global region in the DST model.

The heat transfer process of the local region is simplified as a one-dimensional heat transfer problem, and a one-dimensional radial mesh was constructed to simulate the transient heat transfer process for short timescales. By applying the finite difference method, the temperature change at any point at the local region due to the transient heat transfer process can be calculated. Over longer timescales, when the heat transfer within the local region reaches steady-state, analytical solutions can be applied to simulate this steady-flux process for a BHE with different borehole configurations, e.g. a single U-tube, double U-tube, concentric tubes. There is no heat flux across the border of the local region and the temperature at the border is calculated by superimposing the temperature change due to the two adjacent BHEs that share that border. The global solution is obtained by applying the finite difference method to a two-dimensional radial-axial mesh to simulate the three-dimensional heat transfer problem. The global thermal process includes three fundamental parts: a transient thermal build-up of the temperature field around the storage volume during the initial years, a steady-state heat loss through the surface of storage volume after the initial thermal build-up years, and a periodic heat loss for the annual storage cycle.

The analysis of the heat transfer process explained in the above paragraph is based on applying a constant heat pulse. For time-varying heat transfer, the load profile needs

to be decomposed into a series of heat pulses, and after obtaining the responses to each heat pulse, the results of these responses can then be superimposed in time in order to obtain the final solution.

The DST model is developed based on the assumption that the BHEs are densely packed, with a minimum surface area to volume ratio, which is typical for heat storage; however this assumption is not ideal for other applications, for example, multi-year simulations of ground source heat pump systems where the boreholes are more widely spaced.

The DST model has been implemented in TRNSYS, a transient systems simulation program with a modular component based structure. This model implementation is able to deal with relatively short time steps.

3.8 Numerical Models

3.8.1 1D finite difference borehole model

Shonder and Beck (2000) developed a parameter estimation based method, which is used in combination with a 1D numerical model. This model is similar to a cylinder-source representation, in that it represents the two legs of the U-tube as a single cylinder. However, it adds two additional features: a thin film, that adds a resistance without heat capacity; and a layer of grout, which may have a thermal conductivity and heat capacity different from the surrounding soil (Figure 3.19). In addition, unlike a standard cylinder heat source solution, this model accommodates time-varying heat input.

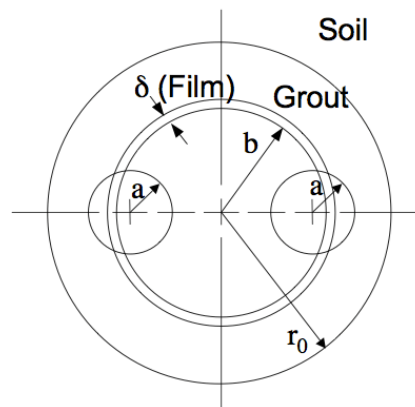


Figure 3.19 : One-dimensional numerical geometry (Shonder and Beck, 2000).

Altogether, the model presented above contains nine parameters: the thermal conductivities of the soil, grout, and thermal film; the volumetric heat capacities of the soil, grout, and thermal film, the thickness of the thermal film; the effective pipe radius, and the far-field temperature. In general, however, it will not be possible to estimate all of these parameters with a single experiment, as some may be dependent on others. For example, since the thin film is used to account for the thermal capacitance of the fluid and pipes, the thickness, δ , is related to the film's volumetric heat capacity.

It is noted that, if the thickness and the volumetric heat capacity of the thin film is zero and the thermal characteristics of the grout are same as the soil, this method is equivalent to a cylindrical heat source problem in an infinite medium.

3.8.2 2D finite volume borehole model

The procedure developed by Yavuztürk (1999) and Austin III et al. (2000) utilizes a parameter estimation technique, which adjusts the thermal conductivities of the grout and ground. A numerical model is used to obtain the best possible match to the experimentally determined temperature response. These thermal conductivities are the best estimates of actual thermal conductivities.

A two-dimensional (polar coordinates; r, θ) finite volume model is utilized. The inner part of the numerical domain is shown in Figure 3.20. For a typical borehole, a grid resolution of 100 finite volume cells in the angular direction and 150 to 200 cells in the radial direction is utilized. The exact grid resolution is a function of the borehole and U-tube pipe geometry and is determined by an automated parametric grid generation algorithm. The radius of the numerical domain is 3.6 m to allow for a reasonably long simulation time. The geometry of the circular U-tube pipes is approximated by “pie-sectors” over which a constant heat flux is assumed to be entering the numerical domain for each time step. The pie-sector approximation attempts to simulate the heat transfer conditions near the pipe walls. The heat flux at the pipe wall is time-dependent, which is determined from experimentally measured power input. Accordingly, the method has no problems associated with fluctuating power levels. The convection resistance due to the heat transfer fluid flow inside the U-tubes is accounted for through an adjustment on the conductivity of the pipe wall material by evaluating effective pipe resistance.

In this model, 60% of heat flux is assigned to one leg of U-tube (i.e. inlet), and 40% to the other leg (i.e. outlet). For the calculation, half of the geometry is used since it's symmetrical.

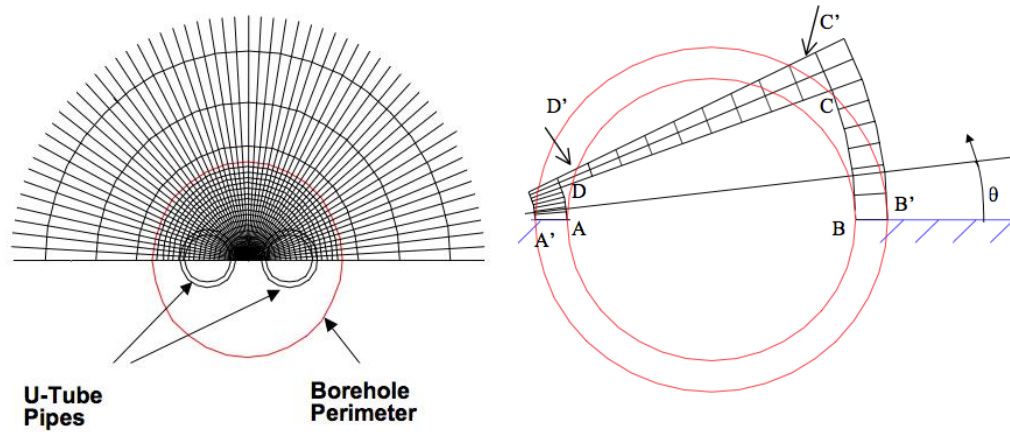


Figure 3.20 : Two-dimensional numerical grid used by Yavuztürk (1999) and Austin III et al. (2000).

The parameter estimation algorithm minimizes the sum of the squares of the errors between the numerical model and the experimentally determined temperature response. A number of optimization methods have been tested. For this problem, which involves searching a narrow turning valley, the Nelder-Mead Simplex method with O'Neill's modifications seems to be the best method (Jain, 1999).

This approach was further refined by a boundary-fitted coordinate grid, as shown in Figure 3.21, with finite volume method (Spitler et al., 2000). However, for real-world applications, there is a point of diminishing returns here, as the down-hole geometry is not known precisely, even under the best circumstances. Spacers that force the U-tube against the borehole wall may help achieving uniform borehole geometry along the axis.

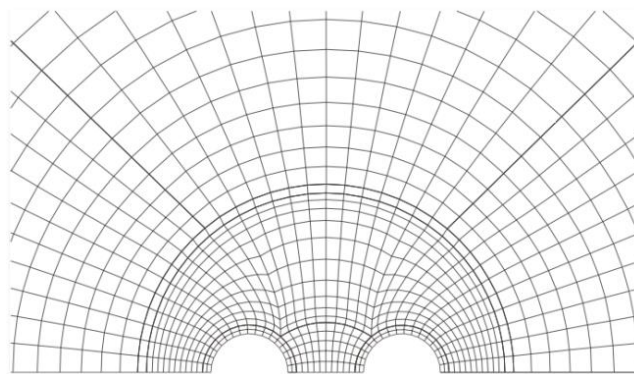


Figure 3.21 : Boundary fitted coordinate grid (Spitler et al., 2000).

3.8.3 3D numerical models in COMSOL Multiphysics™ environment

Marcotte and Pasquier (2008) and Schiavi (2009) developed three-dimensional numerical models of a thermal response test using COMSOL Multiphysics™ environment.

3.8.3.1 Numerical model of Marcotte and Pasquier (2008)

The model is constructed within the COMSOL environment and simulates the thermal behavior of a borehole heat exchanger. It comprises a geological media, a heat carrier fluid circulating inside a U-loop and a borehole filled with a conductive material (i.e. grout). Note that the fluid velocity inside the pipes is assumed constant and, therefore, it does not take into account the convective resistance occurring at the tube wall.

Figure 3.22 shows the finite element discretization of the model. The symmetry of the problem is used to model half of the domain, therefore considerably reducing the number of elements required. A no-flow condition is imposed on the vertical symmetry plane. The undisturbed ground temperature is imposed on the other vertical external boundaries. The mean fluid temperature is calculated via integration at the bottom horizontal face of the descending pipe. This temperature is imposed as a boundary condition for the lower face of the ascending pipe, allowing an identical temperature and heat flow on both sides of the U-bend. The same approach is used to compute the fluid temperature leaving the upper face of the ascending pipe (T_{out}). The fluid temperature increase (ΔT) is computed for a given heating power, fluid heat capacity and mass flow rate. The temperature of the fluid entering the U-loop is then computed as $T_{in} = T_{out} + \Delta T$.

With the exception of the U-loop horizontal faces, the model assumes thermal insulation on all the other horizontal faces. Thus, the axial effects are neglected, a reasonable assumption considering the model is used to obtain the fluid temperature for a few days only, i.e. the typical duration of a thermal test. Lamarche and Beauchamp (2007) indicate that axial effects could play a role only after a period much longer than the typical thermal test duration.

Also, the fluid mass flow rate is considered large enough to always remain in transition-turbulent regimes, the standard practice. To simulate the quick heat propagation to the borehole wall, expected in turbulent regimes, an anisotropic fluid

thermal conductivity tensor with an anisotropy ratio of 100 between horizontal plane and axial direction is used.

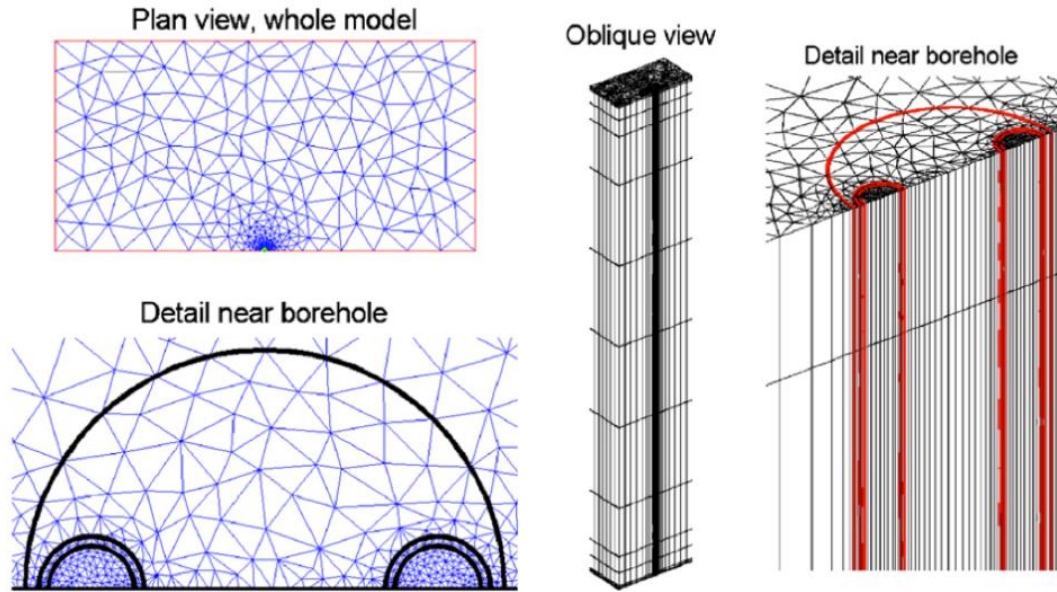


Figure 3.22 : Three-dimensional COMSOL numerical model geometry (Marcotte and Pasquier, 2008).

3.8.3.2 Numerical model of Schiavi (2009)

In order to evaluate the effective heat rates exchange during a real TCT, a tri-dimensional borehole heat exchanger was simulated, that couples the tri-dimensional transient conduction heat transfer problem within the soil, the borehole filling material and the HDPE tubes, with the one-dimensional convective problem within the carrier fluid.

The scheme of the geothermal heat exchanger and of the coupled energy storage system considered in the analysis is schematically shown in Figure 3.23a. It consists of a U-tube pipe having inner radius of 16.3 mm , with downward flow in the right-tube section and upward flow in the left-tube section, immersed in a filling material. The HDPE tubes, having a 3.7 mm thick wall, were also modeled, while for the sake of simplicity the wall thickness of the pipes was disregarded.

The system shows a symmetry plane and it is considered practically unlimited in the radial direction ($R \rightarrow \infty$), while in axial direction it is limited by two adiabatic surfaces placed at $z=0$ (soil surface) and $z=H$. (simulated depth of the heat exchanger). The borehole heat exchanger modeled in the COMSOL Multiphysics™ environment is reported in Figure 3.23b and Figure 3.24.

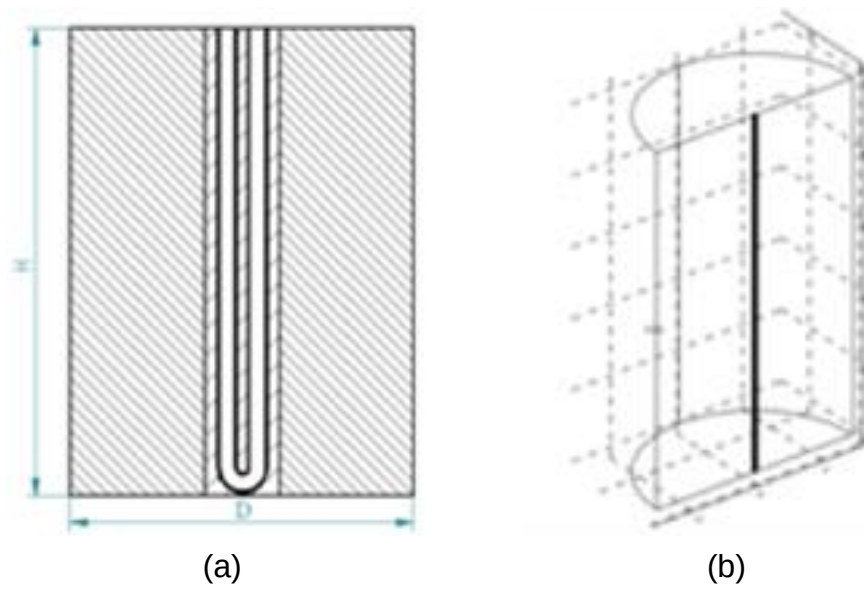


Figure 3.23 : Geometry of the borehole heat exchanger (Schiavi, 2009).

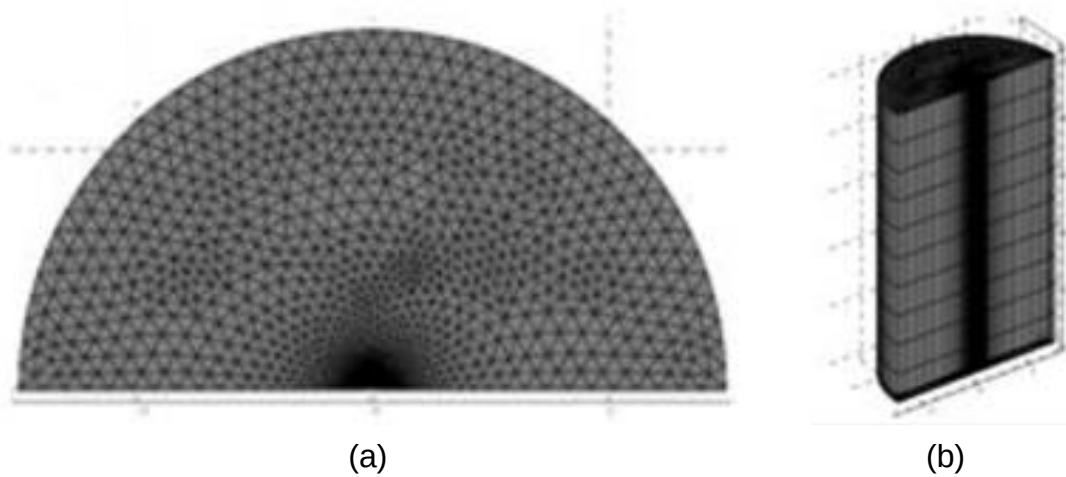


Figure 3.24 : Three-dimensional COMSOL numerical model geometry (Schiavi, 2009).

The geometry and methodology is similar to Marcotte and Pasquier's numerical model. However, in Schiavi's model, pipe wall thickness and the carrier fluid are not modeled in the three-dimensional domain.

4. NUMERICAL MODELING OF THERMAL PROCESSES IN VERTICAL GEOTHERMAL HEAT EXCHANGERS USING FINITE ELEMENT METHOD

In this chapter, three distinctive numerical modeling approaches are developed in order to simulate the ground source heat pump systems. The models utilize finite elements and were created using COMSOL Multiphysics™ (COMSOL, 2013). The proposed numerical models can be used both for assessment of thermal conductivity test and operation of ground coupled heat exchanger systems.

Discretizing the geometry and materials inside and outside the borehole has been proved to be an effective and accurate method that can explicitly simulate the complex geometry of GHEs and the thermal capacities of all GHE components – carrier fluid, pipes, filling material (grout, concrete, etc.) and the ground. This approach has been applied widely in recent development of GHE models, both 2D and 3D. Although 2D discretized numerical models are much more efficient in computation, 3D discretized numerical models offer a more general and accurate representation of the heat transfer processes.

The developed numerical models are used to simulate in-situ thermal conductivity tests with single and double loop configurations for a borehole and a heat exchanger pile, respectively. The analysis results of the numerical models are compared with each other; and an analytical method, i.e. the finite line source method is also used to evaluate the thermal conductivity tests, as well.

The details on the simulated thermal conductivity tests, the methodology for model development, the working principles of each proposed model, and the analyses results are presented in the following sections. The considered models are classified as follows:

- Model A: 3D Model (Ozudogru et al., 2012)
- Model B: 1D-3D Hybrid Model (Ozudogru et al., 2014)
- Model C: 2D Model (Olgun et al., 2014a)
- Model D: Finite Line Source Model (Zeng et al., 2002; Ozudogru et al., 2014)

The development of the models includes the discretization of the geometry, building the finite element mesh and assigning the appropriate initial and boundary conditions for all domains and boundaries. For the model extents, i.e., the side and the bottom model boundaries, either a prescribed temperature equal to the initial condition or a no heat flux (insulation) boundary condition can be specified. In the former case, it must be confirmed that there is no heat flux across the limit boundaries during the analysis, whereas in the latter case there must be no temperature change at the boundaries. Apart from the initial and boundary conditions that were used for modeling the selected tests presented here, further discussions of assigning the initial conditions for the model domains and the boundary condition for the ground surface are presented in Section 4.2, as there are some alternative ways to do so for advanced modeling.

4.1 Considered Modeling Approaches

4.1.1 3D model (Model A)

Heat transfer across different domains, such as between the pipes and the borehole, the borehole and the surrounding ground, or between various ground strata is provided by continuity along internal domain boundaries, which ensures the heat flux continuity in the normal direction across the considered boundary.

In the 3D numerical modeling approach, the geometry discretization consists of the geothermal heat exchanger, the pipes and the soil domains. All of the domains are generated using cylindrical elements. The number of the soil domains depends on the stratification and the model boundaries on the sides and on the bottom are selected so that they must be far from the thermal interaction zone. The width of this zone is a function of thermal parameters of materials, the applied heat rate and the duration of the transient analysis.

For the sake of simplicity and to reduce the number of finite elements greatly, the carrier fluid is not geometrically modeled. This approach saves great computational effort and also prevents anticipation of convergence problems, as the complex nature of turbulent flow would take long time to converge and may even lead to convergence errors. Although the fluid is not discretized in the model geometry, the fluid flow and the heat transfer between the fluid and the pipe inner wall are considered and explained below. Furthermore, the pipes are modeled as hollow cylindrical elements without a

U-bend and despite having a geometrical contact, they are equation wise connected (Figure 4.1).

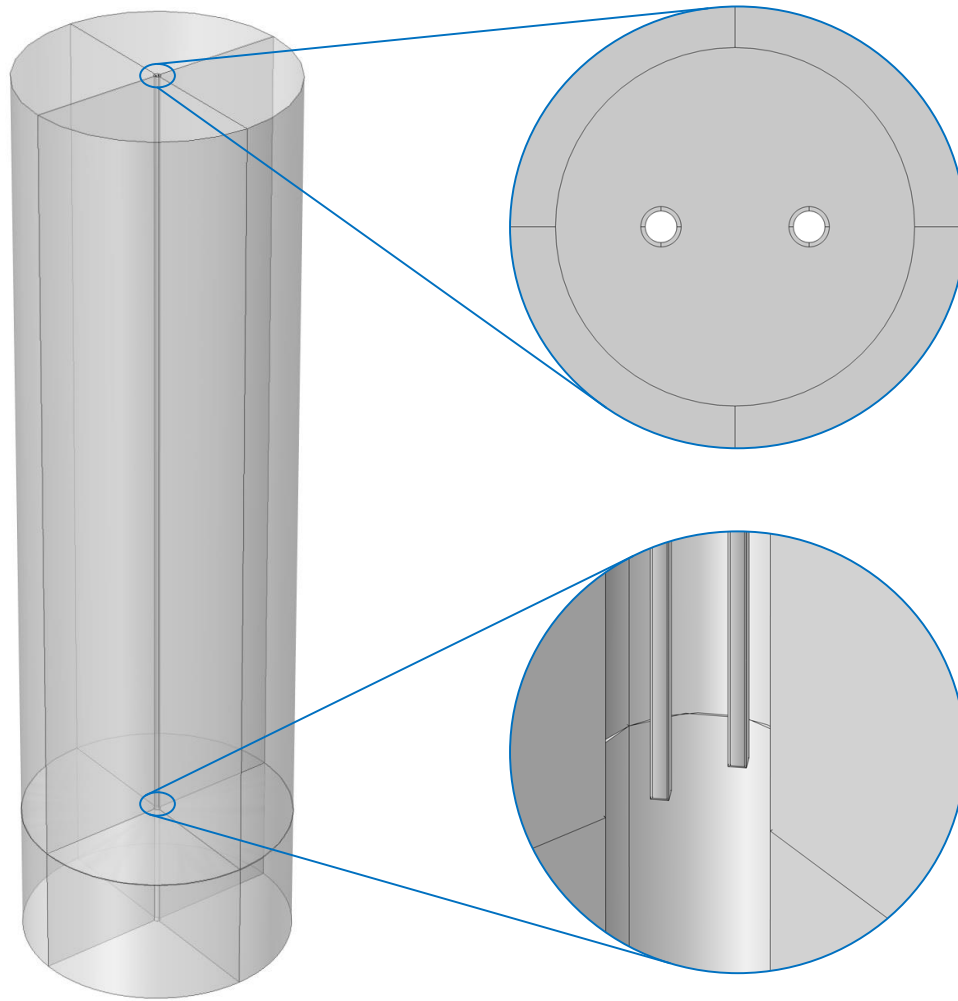


Figure 4.1 : Discretization of the geometry.

In cases where the heat exchanger has a symmetrical geometry, such as GHEs with a single U-tube at the center (Figure 4.1), GHEs with double U-tubes at equal distances from the center or for similar symmetrical configurations, only half geometry of the borehole and the surrounding ground (i.e. half cylinder) can be discretized. This approximation reduces computational time almost by half.

The meshing of the domains is carried out first by generating the mesh for the horizontal surface located at the bottom of the heat exchanger using triangular elements. This meshed surface is then swept towards the domain limits (Figure 4.2). Using swept mesh throughout the geometry saves significant computational effort while keeping the high accuracy as the vertical heat flow has diminished effect on the whole system.

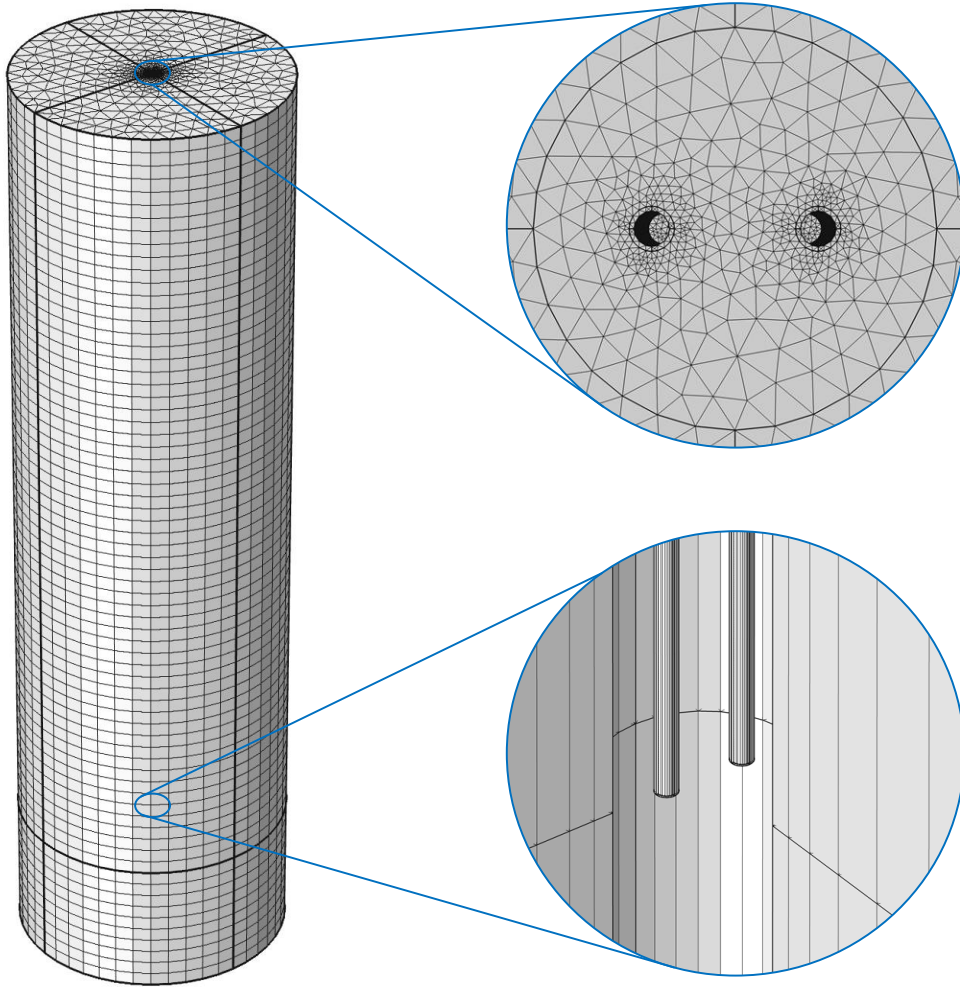


Figure 4.2 : Triangular swept mesh.

As mentioned before, the carrier fluid is not spatially discretized in the proposed model. However, since it is the most important factor affecting the heat transfer mechanisms inside the borehole, its physical representation must be explicitly considered.

In the numerical model, the carrier fluid is introduced as a thin conductive layer, which resides on the pipe inner walls and has geometrically zero thickness. The fluid flow is modeled by calculating the three-dimensional velocity profile along the pipe inner walls by running a stationary analysis on the thin conductive layer. In this analysis, prescribed temperature boundary condition is applied on the edge boundaries of the thin conductive layer, with 1 K on the inlet and 0 K on the outlet. In order to ensure the connection of the downwards and the upwards legs, another prescribed temperature boundary condition is used on the bottom edge of the upwards leg, such that it is equal to the bottom edge of the downwards leg (Figure 4.3). The obtained temperature

profile on the thin conductive layer using the stationary analysis is not a real temperature profile; however, it is an intermediate step to model the fluid flow. If the dependent temperature variable is defined as vel , the flow velocity components can be expressed as follows:

$$\begin{aligned}
 |vel| &= \sqrt{vel_{Tx}^2 + vel_{Ty}^2 + vel_{Tz}^2} \\
 u &= U \cdot vel_{Tx} / |vel| \\
 v &= U \cdot vel_{Ty} / |vel| \\
 w &= U \cdot vel_{Tz} / |vel|
 \end{aligned} \tag{4.1}$$

where,

$|vel|$: magnitude of the temperature gradient vector, vel ,

u, v, w : flow velocity components in x, y and z directions, respectively,

U : average flow velocity, which can be estimated as the ratio of volumetric flow rate to the inner pipe cross-section area,

$vel_{Tx}, vel_{Ty}, vel_{Tz}$: tangential derivatives of the dependent variable vel in x, y and z directions, respectively.

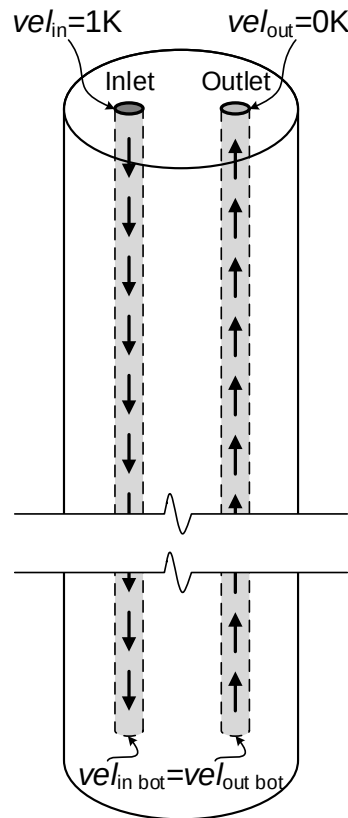


Figure 4.3 : Applied prescribed temperature boundary conditions in the stationary analysis.

Given a constant volumetric flow rate, the flow velocity components can be estimated in the stationary analysis using Equation (4.1). Moreover, if the flow rate varies over time, U will be calculated in each time step, and the velocity components of the carrier fluid will be updated, as well. The average flow velocity along the pipes is assumed to be uniform at each time step and this assumption is reliable since the flow in GHEs is fully developed and the flow regime is turbulent.

The first step of constituting the coupling of the carrier fluid and the pipe inner walls is to introduce the convective heat flow by applying a heat source on the boundaries of the thin conductive layer. This heat source is a function of thermal properties of the fluid, flow characteristics and the fluid temperature. Applied heat rate is estimated using the heat conduction-convection Equation (2.8) as follows:

$$q'' = -\rho_f c_{pf} (uT_{fTx} + vT_{fTy} + wT_{fTz}) \quad (4.2)$$

where, the subscript f denotes the carrier fluid and T_{fTx} , T_{fTy} and T_{fTz} are the tangential derivatives of the fluid temperature in x , y and z directions, respectively.

Next, two convective boundary conditions with different heat transfer coefficients are defined. One is defined on the pipe inner wall boundaries to characterize the heat flow from the fluid to the pipe, and the other is defined on the thin conductive layer, which denotes the heat transfer from the pipe to the fluid.

$$\begin{aligned} q''_{f \rightarrow p} &= h(T_f - T_{pwi}) \\ q''_{p \rightarrow f} &= h_{p \rightarrow f}(T_{pwi} - T_f) \end{aligned} \quad (4.3)$$

where, the $q''_{f \rightarrow p}$ and $q''_{p \rightarrow f}$ are fluid-to-pipe and pipe-to-fluid heat flow rates, respectively. The subscript pwi denotes the pipe inner wall.

The difference in the heat transfer coefficients arises because the carrier fluid is modeled as a thin conductive layer –a surface element– rather than a domain with volume. These two are inter-related with the volume-surface area ratio of the fluid as follows:

$$h_{p \rightarrow f} = h \frac{4}{d_{pi}} \quad (4.4)$$

where, d_{pi} is the pipe inner diameter.

The heat transfer coefficient, h is calculated using:

$$h = \text{Nu} \frac{k_f}{d_h} \quad (4.5)$$

where, d_h is the hydraulic diameter, defined as 4 times the ratio of the wetted area to the wetted perimeter. For circular pipes with pressurized flow, hydraulic diameter is equal to the pipe inner diameter. Nusselt number is estimated using Gnielinski's correlation and Churchill's friction factor equation, which are previously presented in Equation (2.22), Equation (2.23) and Equation (2.24).

In the solid domains pure conductive heat transfer is anticipated which is governed by the following equation assuming that there is no internal heat generation:

$$\rho c_p \frac{\partial T}{\partial t} + \nabla \cdot (-\mathbf{k} \nabla T) = 0 \quad (4.6)$$

Equation (4.6) is solved for temperature, T , making use of the assigned boundary conditions and the temperature coupling with thin conductive layer element. Boundary conditions designate the behavior of the numerical model during runtime and are used to build the sparse matrix solved thereafter for estimating the temperature changes. Either no heat flux (insulation – Neumann), or prescribed temperature (Dirichlet) boundary condition should be specified for the lateral and the bottom limits of the model. For the former case, it must be ensured that there is no temperature change at these boundaries while for the latter case, the heat flux on these boundaries must be zero throughout the runtime of the model. This can be achieved by setting the extent of the model at distances where heat exchange operations essentially have no effect. The Dirichlet boundary condition can only be used if the initial temperature of the ground is selected as a single value for all depths. For models with symmetrical geometry, no heat flux boundary condition is used on the symmetry plane.

The Neumann boundary condition with no heat flux can be expressed as:

$$q'' = (-\mathbf{k} \nabla T) = 0 \quad (4.7)$$

The heating or cooling of the fluid is modeled using a prescribed temperature boundary condition at the edge boundaries of the inlet. This is accomplished by applying a temperature difference on the inlet boundary and expressed as:

$$T_{\text{in}}(t) = T_{\text{out}}(t) + \Delta T(t) \quad (4.8)$$

where,

$T_{\text{in}}(t)$: inlet temperature at time t ,

$T_{\text{out}}(t)$: outlet temperature at time t ,

$\Delta T(t)$: applied temperature difference at time t .

Applied temperature difference, $\Delta T(t)$ maintains a temperature difference between the inlet and outlet throughout the analysis, thus simulates heat injection (heating) or heat extraction (cooling) in the system. $\Delta T(t)$ is a function of time; and it can be constant or can vary with time, to take into account the fluctuations in the power input, electric failures or pauses in heating/cooling.

$\Delta T(t)$ can be simply acquired from the field measurements of the inlet and outlet temperatures or can be estimated using the applied heat power and the fluid properties. The former method is more reliable, as it considers the additional heat input applied by the water pump, and furthermore, it is better to measure the inlet/outlet temperatures closer to the borehole to minimize the effect of the ambient air temperature. For the latter method, the applied temperature difference can be estimated by:

$$\begin{aligned} q(t) &= \dot{m}(t) c_{pf} \Delta T(t) = \dot{V}(t) \rho_f c_{pf} \Delta T(t) \\ \Delta T(t) &= q(t) / (\dot{V}(t) \rho_f c_{pf}) \end{aligned} \quad (4.9)$$

where,

$q(t)$: heat input power at time t , (W),

$\dot{m}(t)$: mass flow rate at time t , (kg/s),

$\dot{V}(t)$: volumetric flow rate at time t , (m^3/s).

4.1.2 1D-3D hybrid model (Model B)

The 1D-3D hybrid numerical model consists of several components. These include the pipes, the grout/backfill, and the soil/rock surrounding the heat exchanger. The lateral and bottom extents of the model are selected such that no thermal interaction occurs

across the external boundaries. The diameter of the pipe elements is taken equal to the pipe outside diameter and these elements are identified as ‘pseudo pipes’.

The actual physical representation of the pipes and the circulated fluid is modeled using a line element that crosses between the center-axes of pseudo pipes in each individual loop. On the other hand, pseudo pipes are used to compensate for the temperature coupling error which will be discussed in the following sections. The cross-sections of typical GHE configurations (single loop and double loop), showing the model components are schematically given in Figure 4.4. Symmetry could not be used in the discretization of the single loop model; because the linear pipe element cannot be divided. On the contrary, double loop models can benefit from symmetry if the operation of the GHE is performed in parallel connection, which greatly reduces the computation time.

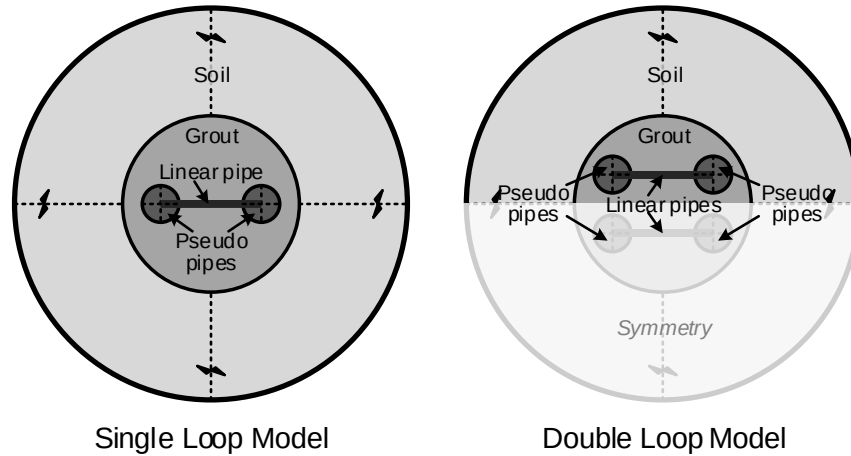


Figure 4.4 : Cross-sections and elements of single and double loop models.

Two dimensional elements are generated at the ground surface for all the model components and extruded downward. The extrusion is controlled by the toe of the heat exchanger borehole/pile, and the boundaries between various soil layers. These controlling locations horizontally divide the 3D domain into soil layers, thus various properties can be assigned for each layer accordingly. With the help of the linear pipe elements, the extrusion of the horizontal plane allows the creation of continuous line elements by establishing a connection between downward and upward pipes at the toe of the GHE. Figure 4.5 provides more insight into the linear pipes and the pseudo pipe approach, presenting the 3D extruded geometry.

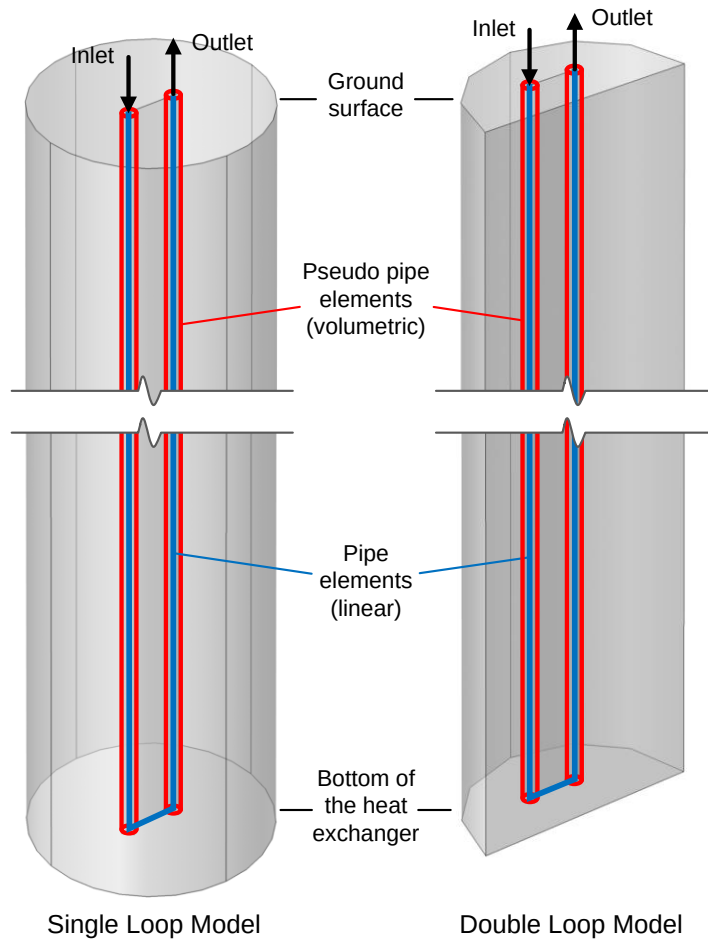


Figure 4.5 : Linear pipe elements and volumetric pseudo pipe elements.

The finite element mesh of the proposed model is generated in a similar way to the discretization of the geometry. First, the horizontal plane on the ground surface is meshed using triangular elements, and then the generated mesh on this plane is swept downward until the limits of the model, to obtain the overall finite element mesh which consists of triangular prism elements. 2D planar projections of the finite element meshes of the generic single and double loop models are shown in Figure 4.6.

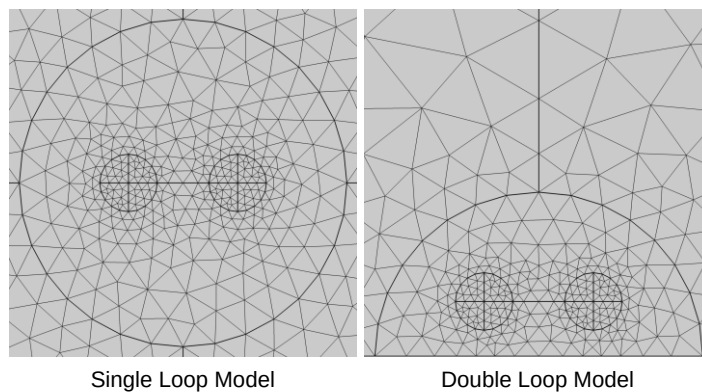


Figure 4.6 : 2D finite element meshes of the generic single and double loop models.

In vertical GHE applications, since the heat transfer is predominantly in the radial direction and the vertical heat flow in the system is insignificant, it is a reasonable approach to distribute the swept mesh on the vertical axis with refinements only close to layer interfaces, or the depths where the extrusion of the 2D geometry was stopped. This procedure makes the vertical mesh inside the soil layers coarser. There may be exceptions to this approach in certain cases, such that, the first several meters below the ground surface should also be refined if the vertical temperature gradient caused by climatic effects will be considered in the model. The overall finite element mesh of a generic heat exchanger pile model with double loop configuration is shown in Figure 4.7.

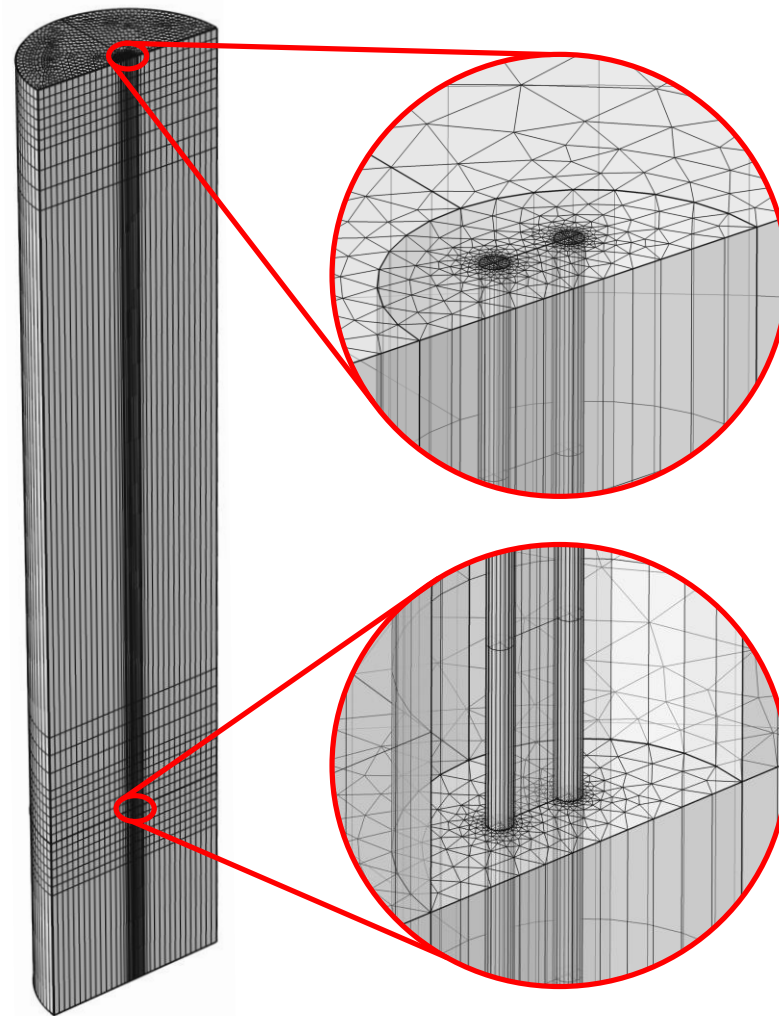


Figure 4.7 : Finite element mesh of a double loop heat exchanger pile model.

The physics of the utilized problems in the proposed numerical model in order to simulate the operation of a GHE system can be divided into two categories: (1) the

time dependent heat transfer problem in the volumetric domains, which is solved by calculating the temperature in each finite element mesh node; (2) the transient fluid flow and forced convection problems in the pipes, solved by evaluating the temperatures of the fluid and the pipe wall along the pipe axis.

In the solid domains such as pseudo pipes, grout, and soil, pure conductive heat transfer problem as given by Equation (4.6) is solved for temperature, T , making use of the assigned boundary conditions and the temperature coupling with the linear pipe elements. The fluid flow and heat transfer problem in the pipes is physically modeled using linear elements, reducing the 3D flow problem to 1D as shown in Figure 4.8. Modeling pipes as curves in 2D or 3D provides great advantage in computational efficiency over meshing and computing 3D pipes with finite diameter.

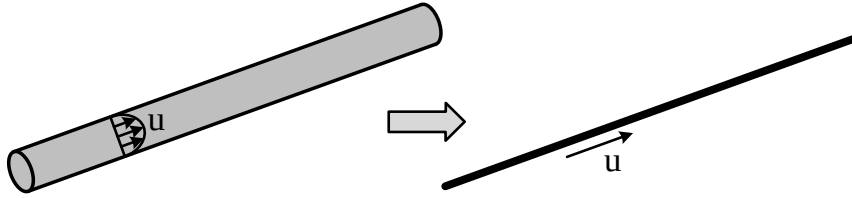


Figure 4.8 : Linear pipe elements reduce the 3D flow problem to 1D (after COMSOL, 2013).

The linear pipe element is an approximation of a heat exchanger pipe by modeling the internal fluid flow and the layered pipe wall; and solving the pipe flow and heat transfer problems simultaneously using built-in equations. The pipe flow problem is determined by solving the momentum and continuity equations given as (Barnard et al., 1966):

$$\rho_f \frac{\partial \mathbf{u}}{\partial t} = -\nabla p - f_D \frac{\rho_f}{2d_h} \mathbf{u} |\mathbf{u}| \quad (4.10)$$

and

$$\frac{\partial A_{pi} \rho_f}{\partial t} + \nabla \cdot (A_{pi} \rho_f \mathbf{u}) = 0 \quad (4.11)$$

The second term on the right-hand side in Equation (4.10) represents the pressure drop due to viscous shear. The Darcy friction factor, f_D in Equation (4.10) accounts for the continuous pressure drop along a pipe segment due to viscous shear, and is expressed

as a function of the Reynolds number, Re and the ratio of the surface roughness to the hydraulic diameter, e/d_h .

$$f_D = f\left(Re, \frac{e}{d_h}\right) \quad (4.12)$$

The Darcy friction factor can be estimated using the Churchill (1997) equation as follows:

$$f_D = 8 \left[\left(\frac{8}{Re} \right)^{12} + (C_A + C_B)^{-1.5} \right]^{1/12} \quad (4.13)$$

where C_A and C_B are factors given as:

$$C_A = \left[-2.457 \ln \left(\left(\frac{7}{Re} \right)^{0.9} + 0.27(e/d_h) \right) \right]^{16} \quad \text{and} \quad C_B = \left(\frac{37530}{Re} \right)^{16} \quad (4.14)$$

Equation (4.13) is valid for all flow conditions, i.e. laminar or turbulent. Moreover, e in Equation (4.14) denotes the absolute surface roughness of the pipe; and for plastic pipes, it is given as 0.0015 mm.

Reynolds number, Re is defined as the ratio of the inertial forces to the viscous forces, such that:

$$Re = \frac{\rho_f u d_h}{\mu} \quad (4.15)$$

Heat transfer in pipes problem is governed by the energy equation for an incompressible fluid flowing in a pipe:

$$\rho_f A_{pi} c_{pf} \frac{\partial T}{\partial t} + \rho_f A_{pi} c_{pf} \mathbf{u} \cdot \nabla T = \nabla \cdot A_{pi} k_f \nabla T + f_D \frac{\rho_f A_{pi}}{2d_h} |\mathbf{u}|^3 + q'_{wall} \quad (4.16)$$

The second term on the right hand side corresponds to friction heat dissipated due to viscous shear. The radial heat transfer from the surroundings into the pipe is given by:

$$q'_{wall} = (hZ)_{eff} (T_{ext} - T) \quad (4.17)$$

Figure 4.9 shows the cross-section of the pipe-fluid domains and the temperature distribution across the pipe wall (COMSOL, 2013). With reference to this figure, the effective overall heat transfer coefficient per unit length of the pipe, including the internal film resistance and the wall resistance can be estimated as:

$$(hZ)_{\text{eff}} = \frac{2\pi}{\frac{2}{d_{pi} h_{\text{int}}} + \frac{\ln(d_{po}/d_{pi})}{k_p}} \quad (4.18)$$

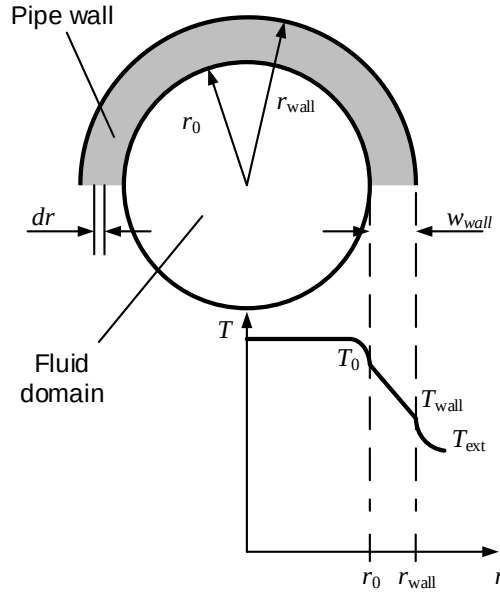


Figure 4.9 : Temperature distribution across the pipe wall (after COMSOL, 2013).

The internal film resistance can be calculated using:

$$h_{\text{int}} = \text{Nu} \frac{k_f}{d_h} \quad (4.19)$$

Nusselt number is defined as the ratio of convective to conductive heat transfer across a boundary. For turbulent flow conditions, Nusselt number can be estimated using the correlation developed by Gnielinski (1976) as follows:

$$\text{Nu} = \frac{(f_D/8)(\text{Re}-1000)\text{Pr}}{1+12.7(f_D/8)^{1/2}(\text{Pr}^{2/3}-1)}, \quad \begin{matrix} 0.5 < \text{Pr} < 2000 \\ 3000 < \text{Re} < 6 \times 10^6 \end{matrix} \quad (4.20)$$

In the numerical model, the external temperature outside of the pipes, T_{ext} corresponds to the temperature field computed in the 3D domains. This provides automatic heat transfer coupling to the volumetric domains, considering the pipes as a line heat source. The coupling between the two physics nodes is accomplished through outer pipe wall temperature. However, there is a limitation in this approach. Since the pipes are made of linear elements, the outer pipe wall temperature is coupled to the temperature field of the volumetric domains located at the pipe axis. This introduces

estimation errors, because the temperature field of the volumetric domains to be coupled should be located at a distance of pipe outside radius from the pipe axis. Another downside of this simplification is that, it does not account for the heat capacity of the pipes. A ‘pseudo pipe’ approach was developed as shown in Fig. 5 to overcome these issues. The diameter of these cylindrical solid domains is equal to the pipe outside diameter. The assigned material properties, which are presented in the next section, ensure the coupling of the temperature accurately, while accounting for the heat capacity of the pipes, as well.

4.1.3 2D horizontal cross-section model (Model C)

The 2D model uses a horizontal cross-section of the thermal pile–soil system and it consists of several components. These include the carrier fluid, circulation pipes, concrete, and the soil surrounding the thermal piles (Figure 4.10). The variation of temperature distribution along the vertical direction will be small as the predominant heat exchange takes place horizontally. Therefore any vertical process is assumed to have negligible influence on the long-term GHE operation and the 3D heat transfer problem is reduced to a 2D time-dependent heat conduction problem.

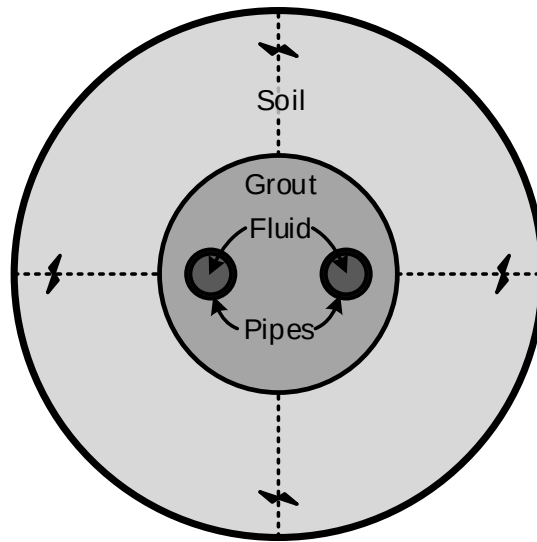


Figure 4.10 : Components of the 2D model.

Pure conductive heat transfer equation as given by Equation (4.6) is solved for temperature, T , making use of the assigned initial and boundary conditions. Fluid flow inside the pipes and the associated convective heat transfer is simulated by an equivalent solid, which has the same heat capacity (i.e., mass density and specific heat

capacity) as the circulation fluid. On the other hand, the thermal conductivity of the equivalent solid is selected to be very high (1000 W/m·K). With this approach, the fluid domain can be utilized to inject a specific temperature or to inject/extract heat per unit length of the thermal pile, at selected time periods during the analysis. High thermal conductivity also leads to a practically uniform temperature distribution within the cross-section of each pipe leg.

In this modeling approach, the thermal resistance due to the convective heat transfer between the circulation fluid and the pipe inner wall is neglected. On the contrary, the convective thermal resistance is taken into account as a conductive resistance in the pipe wall domain, by considering an effective thermal conductivity of the pipe material $k_{p\text{ eff}}$ in the analyses given by:

$$k_{p\text{ eff}} = \frac{\ln(d_{po}/d_{pi})}{\frac{2}{d_{pi} h_{int}} + \frac{\ln(d_{po}/d_{pi})}{k_p}} \quad (4.21)$$

where, h_{int} is the internal convective heat transfer coefficient, k_p is the actual thermal conductivity of the pipe material, d_{po} and d_{pi} are the outside and inside diameters of the pipe, respectively. The internal convective heat transfer coefficient depends on the flow characteristics and the thermal properties of the water and it is evaluated using the correlations developed by Gnielinski (1976) and Churchill (1977). The modeling technique involving the equivalent solid approach explained above is adopted from a study by Lazzari et al. (2010). It should also be noted that, since the axial direction is not considered in the model, the heat rate per unit length is equally distributed in all pipes without making any distinction between upcoming and downgoing pipes. Thus, the temperature of the equivalent solid corresponds to the mean temperature of the fluid inside the pipes.

Quarter symmetry can be used to reduce the computational effort in cases where the geometry of the heat exchanger is symmetrical, since the the fluid temperatures in all pipes are equal and thus the physical/thermal processes in the model are symmetrical. The finite element mesh discretization of the model is performed by using triangular elements. The meshed model domain and the quarter symmetry approach are shown in Figure 4.11.

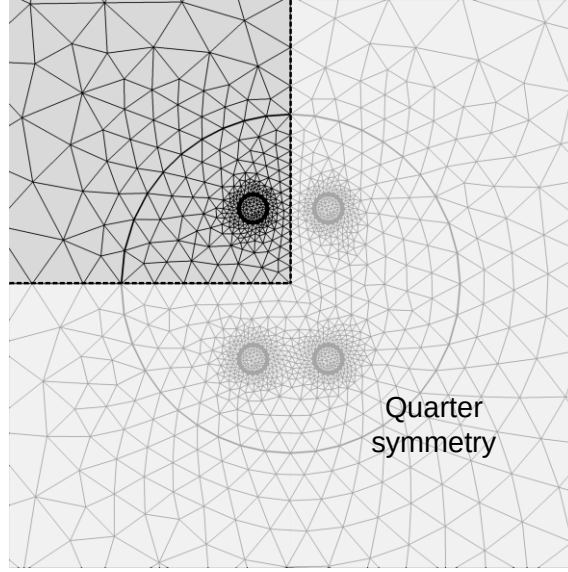


Figure 4.11 : Finite element mesh of the model and the use of quarter symmetry.

4.1.4 Finite line source model (Model D)

The theory of the FLS model has been presented in Section 3.5.2. For the sake of comparability of the results with the proposed numerical models, the conventional analytical method is expanded in order to account for variable heat flux. Yang et al. (2009) proposed a method to account for the varying heat rate in ILS, which uses step loading and utilizes the principle of superposition in the time domain. The details of this modification is presented in Section 4.2.3. Using this same principle, to account for the varying heat rate for a heating period of t_1 and afterwards a recovery period of $(t_2 - t_1)$, the modified FLS equation becomes:

$$\Delta T(t) = \left(\frac{q}{4\pi k} \right) \int_0^H \left[\frac{\operatorname{erfc} \left(\sqrt{r^2 + (z-h)^2} / 2\sqrt{\alpha t_2} \right) - \operatorname{erfc} \left(\sqrt{r^2 + (z-h)^2} / 2\sqrt{\alpha(t_2 - t_1)} \right)}{\sqrt{r^2 + (z-h)^2}} - \frac{\operatorname{erfc} \left(\sqrt{r^2 + (z+h)^2} / 2\sqrt{\alpha t_2} \right) - \operatorname{erfc} \left(\sqrt{r^2 + (z+h)^2} / 2\sqrt{\alpha(t_2 - t_1)} \right)}{\sqrt{r^2 + (z+h)^2}} \right] dh \quad (4.22)$$

Moreover, the analytical models such as the finite line source can be used to calculate the temperature changes in the ground, starting from the borehole/pile wall. In order to estimate the mean fluid temperature inside the tubes, borehole thermal resistance must be known. The relationship between the mean fluid temperature and the borehole wall temperature is given as:

$$T_f = T_{bw} + \frac{q}{H} R_{GHE} \quad (4.23)$$

Many factors have impact on the borehole resistance, including the diameter of the borehole, the thermal conductivity and the diameters of the U-tubes, their shank spacing, thermal conductivities of the backfill and the ground, flow rate of the fluid and its thermal properties, as well as the flow circuit arrangement. The borehole resistance is a transient parameter, as it starts from zero and approaches a constant value, which is called the steady state borehole resistance. It is common practice that the heat transfer inside the borehole is approximated as a steady-state process, as the temperature variation inside the borehole is usually slow and minor. Such simplification has been proved appropriate and convenient for most engineering practices except for the analysis dealing with dynamic responses considering time scales of a few hours (Yavuztürk, 1999).

Different methods to estimate the steady state borehole resistance have been proposed by various researchers and these include equivalent diameter expression (Gu and O'Neal, 1998), shape factors (Paul, 1996), multipole expansions approach (Bennet et al., 1987), line source approximation (Hellström, 1991). In this study, we have used Hellström's line source approach as presented by Zeng et al. (2003) because in addition to its simplicity, it provides results with reasonable accuracy and can be extended to double loop configurations.

For the borehole with double U-tube configuration given in Figure 4.12, the difference between fluid temperatures in each pipe and the borehole wall can be expressed in terms of heat fluxes per unit length in each pipe and a set of thermal resistances:

$$\begin{aligned}
 T_{f1} - T_{bw} &= R_{11}q_1 + R_{12}q_2 + R_{13}q_3 + R_{14}q_4 \\
 T_{f2} - T_{bw} &= R_{21}q_1 + R_{22}q_2 + R_{23}q_3 + R_{24}q_4 \\
 T_{f3} - T_{bw} &= R_{31}q_1 + R_{32}q_2 + R_{33}q_3 + R_{34}q_4 \\
 T_{f4} - T_{bw} &= R_{41}q_1 + R_{42}q_2 + R_{43}q_3 + R_{44}q_4
 \end{aligned} \tag{4.24}$$

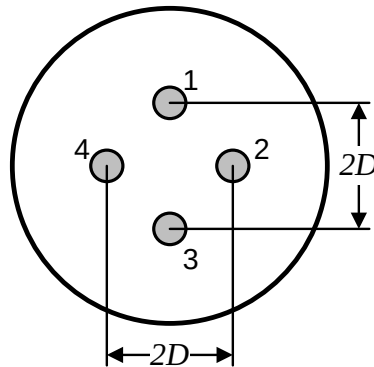


Figure 4.12 : Cross-section of a double U-tube borehole (after Zeng et al., 2003).

In Equation (4.24), R_{ii} denotes the thermal resistance between the circulating fluid in a certain U-tube leg and the borehole wall, and R_{ij} is the thermal resistance between two individual pipes. The solution of Hellström's line source assumption is expressed as:

$$\begin{aligned} R_{11} &= \frac{1}{2\pi k_b} \left[\ln \left(\frac{r_b}{r_{po}} \right) - \frac{k_b - k_g}{k_b + k_g} \ln \left(\frac{r_b^2 - D^2}{r_b^2} \right) \right] + R_p \\ R_{12} &= \frac{1}{2\pi k_b} \left[\ln \left(\frac{r_b}{\sqrt{2}D} \right) - \frac{k_b - k_g}{2(k_b + k_g)} \ln \left(\frac{r_b^4 + D^4}{r_b^4} \right) \right] \\ R_{13} &= \frac{1}{2\pi k_b} \left[\ln \left(\frac{r_b}{2D} \right) - \frac{k_b - k_g}{k_b + k_g} \ln \left(\frac{r_b^2 + D^2}{r_b^2} \right) \right] \end{aligned} \quad (4.25)$$

The pipe resistance R_p is a combination of the convective resistance of the fluid and the conduction resistance of the pipe wall. It is defined by the expression:

$$R_p = \frac{1}{2\pi r_{pi} h_{int}} + \frac{\ln(r_{po} / r_{pi})}{2\pi k_p} \quad (4.26)$$

Finally, the overall resistance of the GHE is evaluated as an average of thermal resistances between individual pipes:

$$R_{GHE} = (R_{11} + R_{12} + R_{23} + R_{13}) / 4 \quad (4.27)$$

4.2 Improvements on the Modeling and Analysis of TCT Data

The aim of this analysis is to model the BHE accurately, considering the three-dimensional effects and using the least possible assumptions.

There are several advantages of the proposed 3D numerical model compared to 2D and 1D models and analytical methods which can be summarized as follows:

- Stratified soil profiles can be introduced.
- The ground underneath the GHE is also simulated, allowing the heat transfer below the heat exchanger to be explicitly considered.
- Temperature variations along the depth can be obtained.
- Geothermal gradients can be imposed as initial ground temperature.
- Climate dependent boundary conditions at the ground surface can be applied.

In this section, various considerations that can be implemented in the numerical model will be presented. These approaches eliminate simplification assumptions; hence more accurate results are achieved. They consist of setting the initial conditions for the ground, defining the boundary conditions for the ground surface and taking into account the effect of ambient air temperature on the operation of the borehole heat exchanger system.

The thermal properties of the involved materials can also be defined as temperature dependent. However, since the effect of temperature on the thermal properties of solids is negligible and the wider range of temperature change occurs in the carrier fluid, it is good practice to define only the parameters of the fluid as temperature dependent.

4.2.1 Initial ground temperature

Undisturbed ground temperature is affected by seasonal change in the ambient air temperature and this variation reaches up to approximately 15 m below the ground surface as shown in Figure 4.13 (Ericsson, 1985). Moreover, the undisturbed ground temperature increases with depth due to the geothermal gradient. It varies over the world and typically is in the range of 5-30 K/km (Gehlin, 2002).

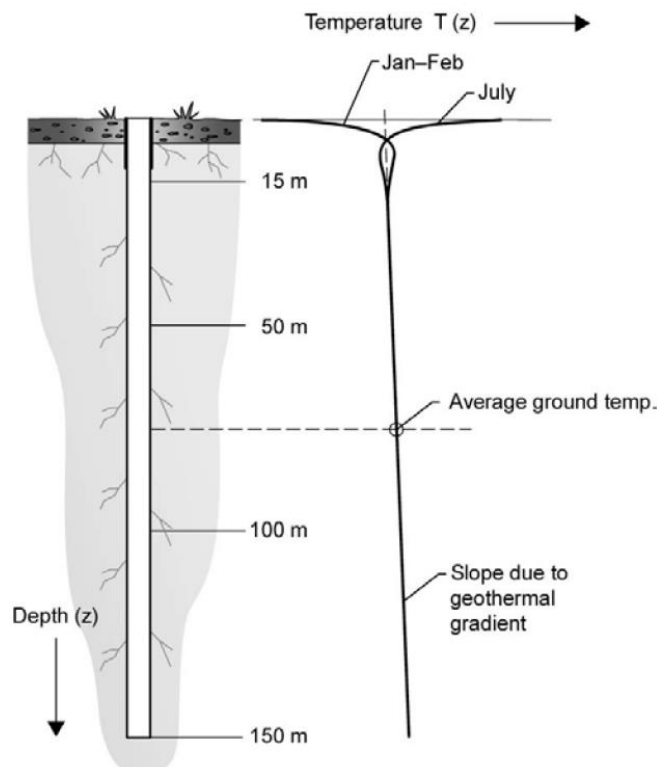


Figure 4.13 : Temperature profile in the ground. Seasonal temperature variations do not reach below 15 m from the ground surface (Gehlin, 2002; adapted from Ericsson, 1985).

The conventional and simple way of setting the initial ground temperature is to assign a constant average value, which is obtained from field measurements. A more accurate approach is to take into account the seasonal variation at shallow depths and the geothermal gradient. The former can affect the analysis results of shallow systems. Since the typical length of the borehole heat exchangers are in the range of 30-200 m, the effect of geothermal gradient on the undisturbed ground temperature for such systems cannot be neglected (Gehlin, 2002).

Kusuda and Achenbach (1965) provided an analytical model, which specifies the ground temperature as a function of time and depth using the daily mean surface temperature data for the whole year. It is given as:

$$T = T_{\text{mean}} - T_{\text{amp}} \cdot e^{-z \sqrt{\frac{\pi}{\alpha t_{\text{cycle}}}}} \cdot \cos \left[\frac{2\pi}{t_{\text{cycle}}} (t_{\text{current}} - t_{\text{shift}}) - z \sqrt{\frac{\pi}{\alpha t_{\text{cycle}}}} \right] \quad (4.28)$$

where,

T : temperature at depth of interest,

T_{mean} : mean surface temperature (average air temperature),

T_{amp} : amplitude of the surface temperature (difference between maximum and minimum air temperature),

D : depth of interest,

α : thermal diffusivity of the ground,

t_{cycle} : the period of soil temperature cycle (365 days),

t_{current} : current day of the year,

t_{shift} : day of the year with minimum surface temperature.

Using Equation (4.28) in the numerical model to assign the initial ground temperature with respect to depth will not account for the geothermal gradient, as it is a limitation of Kusuda-Achenbach's formulation. However, the geothermal gradient and the seasonal temperature variations can be handled simultaneously using a one-dimensional numerical model which provides the undisturbed ground temperature profile prior to any thermal operation of the BHE system. Such temperature profile can be used as the initial condition of the three-dimensional model that is considering the operational system or thermal conductivity tests.

The one-dimensional initialization model is the vertical representation of the ground profile. It has two boundary conditions; a prescribed temperature boundary condition on the ground surface, and a prescribed heat flux boundary condition on the bottom. The boundary condition on the top can be simply assigned using Equation (4.28) with taking the depth zero ($D=0$), as it simplifies to:

$$T = T_{\text{mean}} - T_{\text{amp}} \cdot \cos \left[\frac{2\pi}{t_{\text{cycle}}} (t_{\text{current}} - t_{\text{shift}}) \right] \quad (4.29)$$

On the other hand, the boundary condition on the bottom limit can be estimated using:

$$q'' = k \left(\frac{\partial T}{\partial z} \right)_{\text{geo}} \quad (4.30)$$

where, $(\partial T / \partial z)_{\text{geo}}$ is the geothermal gradient (K/m).

The initialization model is a time dependent analysis which is run for the period:

$$t_{\text{initial}} = t_{\text{current}} + n \cdot t_{\text{cycle}} \quad (4.31)$$

where, n is a whole number denoting the number of years. It should be large enough so that the undisturbed ground temperature profile reaches a steady state for the current day of the year.

4.2.2 Ground surface temperature

In cases where real-time ambient air temperature measurement is not available, the boundary condition on the ground surface can be set as a prescribed temperature, which is equal to the initial ground temperature. However, using a convective boundary condition with ambient air temperature is a more accurate approach. This boundary condition can be introduced as:

$$q'' = h_{\text{air}} (T_{\text{air}} - T_{\text{ground}}) \quad (4.32)$$

The convective heat transfer coefficient of air can be estimated using the correlation developed by Walton (1993). This equation is valid for natural convective flows between a fluid and a horizontal surface, such as the heat flow between the air and the ground surface. It is given as:

$$h_{\text{air}} = 4.7382 |\Delta T|^{1/3} \quad (4.33)$$

where ΔT is the temperature difference between the ambient air and the ground surface.

4.2.3 Variable heat flux modification for analytical methods

The analytical analysis methods such as line heat source methods and the cylindrical heat source method rely on the assumption of constant heat rate throughout the heating or cooling period. However, this theoretical approach does not conform to the practical case of GHE operation. During thermal conductivity testing, many factors may affect the injected/extracted heat, leading to variations in the applied heat flux, as discussed before.

Yang et al. (2009) proposed a modification for the ILS model in order to account for the varying heat flux, which uses step loading and utilizes principle of superposition in the time domain. The superposition principle is illustrated in Figure 4.14 for an example case with four step loading. Each time interval has a different heat rate denoted by q_1 , q_2 , q_3 and q_4 . The proposed method implies that, each variable heat rate contributes for the time interval starting from the time at which it was first applied until the end of entire loading period.

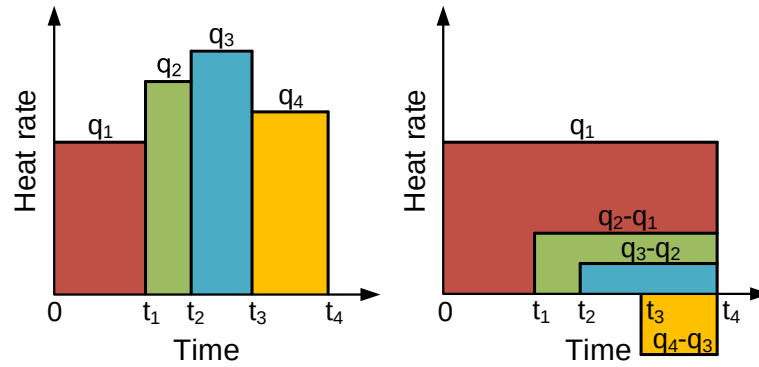


Figure 4.14 : The superposition principle for varying heat rate (after Yang et al., 2009).

For the case above, the temperature change at the end of thermal loading (at $t=t_4$) can be expressed as:

$$\begin{aligned} \Delta T(r, t) = & \frac{q_1}{4\pi k} \text{Ei} \left(-\frac{r^2}{4\alpha t_4} \right) + \frac{q_2 - q_1}{4\pi k} \text{Ei} \left[-\frac{r^2}{4\alpha (t_4 - t_1)} \right] \\ & + \frac{q_3 - q_2}{4\pi k} \text{Ei} \left[-\frac{r^2}{4\alpha (t_4 - t_2)} \right] + \frac{q_4 - q_3}{4\pi k} \text{Ei} \left[-\frac{r^2}{4\alpha (t_4 - t_3)} \right] \end{aligned} \quad (4.34)$$

The method can be generalized for n number of intervals as follows:

$$\Delta T(r, t) = \sum_{i=1}^n \frac{q_i - q_{i-1}}{4\pi k} E_i \left[-\frac{r^2}{4\alpha(t_n - t_{i-1})} \right] \quad (4.35)$$

Although Yang et al. (2009) initially proposed this model for the infinite line heat source method; the same procedure can be used with other analytical methods, as presented for the FLS method in Section 4.1.4.

4.2.4 Thermal recovery period

Thermal response tests are analogous to standard pumping tests conducted in hydrogeology, because a system that is initially assumed at equilibrium is perturbed and the response is monitored in time, to assess the system's properties with inverse modeling. The concepts borrowed from pumping tests allows the planning of the duration of the TRTs and the analysis of variable heat injection rate tests accounting for external heat transfer and temperature recovery, which reduces the uncertainty in the estimation of thermal properties.

The typical duration of a constant heat injection test is between 30 and 60 hours, but there is no prescribed guideline for the test duration. Raymond (2011) proposes that the duration of TRTs should be sufficiently long to allow the temperature perturbation to reach a specific radius of influence. The test duration can further be increased to measure the water temperature recovery after heat injection is stopped. In that case, water circulation is maintained in the heat exchanger once heat injection stops, and temperature is measured until it approaches the initial ground temperature. The flow rate is monitored during recovery to compute the heat injection rates and to verify if external heat transfer affects temperature, such as during a circulation test used to measure the undisturbed subsurface temperature.

The factorial sensitivity analysis indicates that monitoring temperature recovery following heat injection provides additional data to estimate thermal conductivity (k) independently from thermal resistance of the geothermal heat exchanger (R_{GHE}) and increase the accuracy in estimating k (Figure 4.15). It is therefore recommended that a TCT should be analyzed with the line-source model by initially adjusting k using recovery data and afterwards adjusting R_{GHE} using heat injection data with the estimate of k obtained previously to independently determine these two parameters.

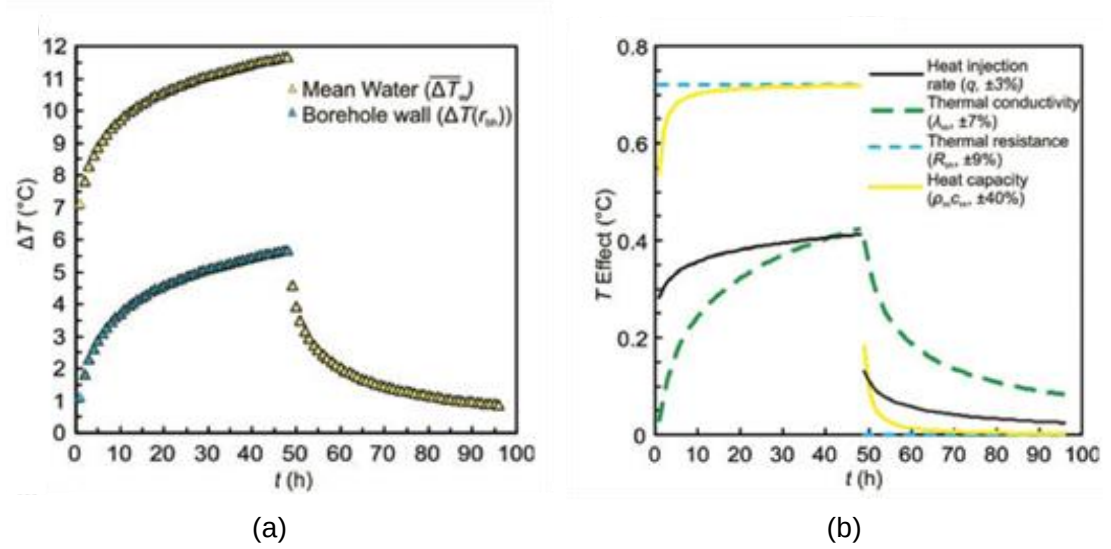


Figure 4.15 : (a) Base case temperature increments calculated with the line-source model. (b) Results of the factorial analysis used to determine parameter sensitivities to the line-source model. The effect of each parameter on computed temperature, which is identical for a positive or a negative variance is plotted as function of time (Raymond et al., 2008).

4.2.5 Effect of ambient air temperature variations

In a typical in-situ thermal conductivity test setup, heat exchange between the ambient air and the carrier fluid is inevitable since it is very difficult to achieve perfect thermal insulation along the pipes above ground, which connect the borehole loop and the test rig (Figure 4.16). The circulating fluid can gain or lose heat depending on the ambient air temperature as the amount of ambient interaction depends on the pipe length between the heating unit and the GHE top, the degree of thermal insulation and temperature difference between the heating fluid and outside environment.

Moreover, experiments have demonstrated that the evaluation of the thermal conductivity is affected by ambient air temperature variations. The influence of diurnal temperature changes on the measured fluid temperature has been reported several times by different researchers (Austin III, 1998; Esen and Inalli, 2009; Florides and Kalogirou, 2008; Fujii et al., 2009; Signorelli et al., 2007).

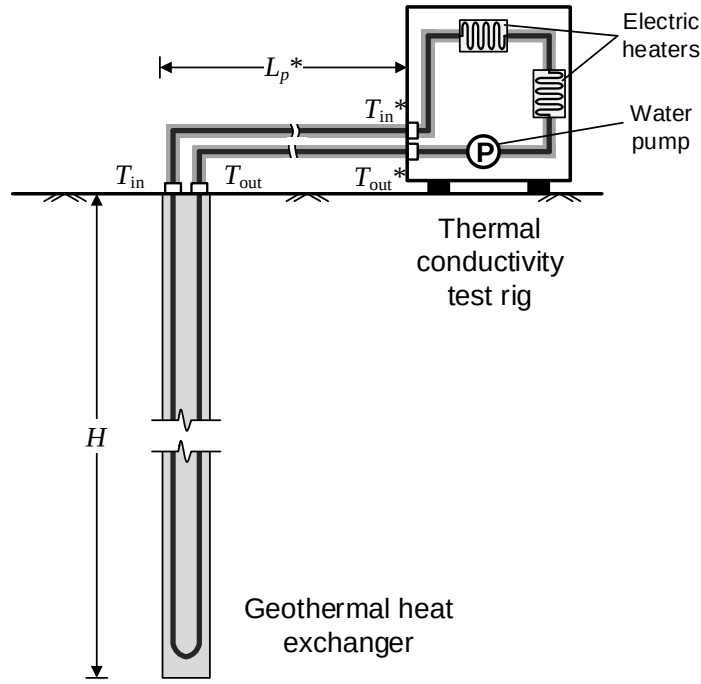


Figure 4.16 : Schematic view of the borehole thermal conductivity test (after Bandos et al., 2011).

Figure 4.16 shows a schematic view of a typical borehole thermal conductivity test setup with the above ground piping. T_{in}^* and T_{out}^* denote the measured temperatures during the test, whereas T_{in} and T_{out} are the actual temperatures interacting with the geothermal heat exchanger. L_p^* stands for the length of the pipe between the measured and the actual inlet and outlet fluid temperatures.

This issue has been addressed by Bandos et al. (2011) with a proposed methodology to evaluate the fluid temperatures entering and exiting the borehole heat exchanger (T_{in} and T_{out}) in reference to the fluid temperatures (T_{in}^* and T_{out}^*) measured in the heater unit within the test truck. In this approach, the heat gain or loss is characterized by a temperature offset with respect to the experimentally measured fluid temperatures (T_{in}^* and T_{out}^*). Additional heat into the system when the outside air is warmer than the fluid temperature results in an increase in the entering fluid temperature with respect to the fluid temperature measured in the test truck. On the contrary, heat loss during episodes colder than the fluid temperature results in a decrease in the entering fluid temperature. The basis of this approach is incorporating heat losses or gains through the length of the above ground piping between the test truck and the test pile. The following equations were developed for applying a correction factor that takes into account the effect of ambient air temperature variations:

$$\begin{aligned} T_{\text{in}} &= T_{\text{in}}^* e^{-\eta} + T_{\text{air}} (1 - e^{-\eta}) \\ T_{\text{out}} &= T_{\text{out}}^* e^{\eta} + T_{\text{air}} (1 - e^{\eta}) \end{aligned} \quad (4.36)$$

where, η is a dimensionless parameter, defined as:

$$\eta = \frac{L_p^*}{\dot{V} \rho_f c_{pf} R_{\text{air}}} \quad (4.37)$$

In Equation (4.37), R_{air} ($m \cdot K/W$) indicates the thermal resistance between the air and the carrier fluid. It is challenging to estimate this parameter precisely and even making an assumption can be very difficult. Hence, a relationship for calculating the dimensionless parameter was provided:

$$\eta = p \frac{\langle T_{\text{in}}^*(t) - T_{\text{out}}^*(t) \rangle_t}{\langle T_{\text{in}}^*(t) + T_{\text{out}}^*(t) - 2T_{\text{air}}(t) \rangle_t} \quad (4.38)$$

where, p denotes percentage of heat exchange between the air and the fluid with respect to the total exchanged heat, and the brackets indicate time averaging during the test period. Bandos et al. (2011) suggest an initial guess of $p=5.0\%$ to be used in Equation (4.38) and using a parameter matching technique until the correct value of p is obtained.

4.3 Model Validation Study

The validation of the proposed finite element modeling approaches is carried out by modeling field thermal conductivity test. The tests to be modeled are selected as tests performed on a heat exchanger borehole and a heat exchanger pile. The borehole is integrated with a single ground loop and 15 cm in diameter and 100 m in length, whereas the pile set-up is selected to be 45 cm diameter and 20 m in length with a double loop configuration. The plan views of field test set-ups for each test, along with the loop placement and pipe leg spacing are given in Figure 4.17.

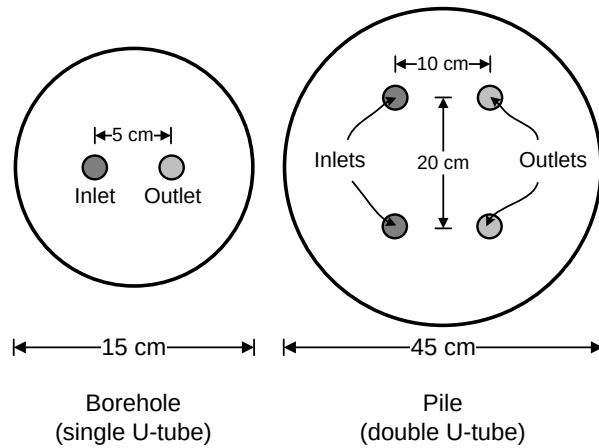


Figure 4.17 : Thermal conductivity test configurations for borehole and pile set-ups.

The geothermal borehole is integrated with 3/4" HDPE pipes and the heat exchanger pile with 1 1/4" pipes, both with a standard dimension ratio (SDR) of 11. The geometrical parameters of the borehole and the pile are given in Table 4.1.

Table 4.1 : Dimensions of the borehole and the heat exchanger pile.

Parameter	Borehole	Pile	Unit
Diameter	15	45	cm
Length	100	20	m
Pipe inner diameter	21.5	34.0	mm
Pipe wall thickness	2.4	3.8	mm
Pipe shank spacing	5	10	cm
Loop spacing	—	20	cm

The undisturbed ground temperature is selected to be 15°C, which represents a moderate climate region. The thermal conductivity tests are performed by applying constant heat rates of 50 W/m and 100 W/m for the borehole and the pile, respectively. The magnitude of the selected heat rate is within the range recommended by Kavanaugh et al. (2001). The duration of the thermal loading is 100 hours and after this period, the temperatures are recorded for a recovery period of 150 hours. For the 3D numerical models and the finite line source model, the ground surface is kept at a constant temperature, which is equal to the undisturbed temperature. A mixture of water and propylene glycol with a concentration of 20% by volume is used as the heat exchange fluid. The thermal and rheological properties of the water-antifreeze solution are taken from the ASHRAE Handbook Fundamentals (ASHRAE, 2009). The thermal properties of the ground and the heat exchanger material are selected by referring to the studies by Farouki (1981) and Salomone et al. (1989). The input parameters used in the numerical analyses are listed in Table 4.2.

Table 4.2 : Parameters used in the numerical analyses.

Parameter	Value	Unit
Global parameters		
Initial ground temperature	15	°C
Heat rate per depth (borehole)	50	W m ⁻¹
Heat rate per depth (pile)	100	W m ⁻¹
Circulation fluid (20% propylene glycol – water solution)		
Flow rate	20	dm ³ min ⁻¹
Dynamic viscosity	2.02	mPa s
Thermal conductivity	0.48	W m ⁻¹ K ⁻¹
Thermal conductivity (for 2D Model)	1000	W m ⁻¹ K ⁻¹
Specific heat capacity	3962	J kg ⁻¹ K ⁻¹
Density	1020.91	kg m ⁻³
Pipe (HDPE)		
Thermal conductivity	0.39	W m ⁻¹ K ⁻¹
Thermal conductivity (for 2D Model)	0.36	W m ⁻¹ K ⁻¹
Specific heat capacity	2300	J kg ⁻¹ K ⁻¹
Density (material)	960	kg m ⁻³
Pseudo pipe (for 1D-3D Hybrid Model)		
Thermal conductivity	{1000, 1000, 0}	W m ⁻¹ K ⁻¹
Density (effective, 3/4")	319.28	kg m ⁻³
Density (effective, 1 1/2")	320.66	kg m ⁻³
Borehole (sand-bentonite)		
Thermal conductivity	1.00	W m ⁻¹ K ⁻¹
Specific heat capacity	1600	J kg ⁻¹ K ⁻¹
Density	1500	kg m ⁻³
Pile (reinforced concrete)		
Thermal conductivity	1.50	W m ⁻¹ K ⁻¹
Specific heat capacity	1000	J kg ⁻¹ K ⁻¹
Density	2500	kg m ⁻³
Ground		
Thermal conductivity	2.00	W m ⁻¹ K ⁻¹
Specific heat capacity	1500	J kg ⁻¹ K ⁻¹
Density	2000	kg m ⁻³

4.4 Analysis Results and Comparison of the Models

During the heating period (0–100 h), the borehole is heated with a constant heat rate of 50 W/m and the injected heat rate for the pile is 100 W/m. These heat rates per unit length of the GHEs correspond to total heat powers of 5000 W and 2000 W for the borehole and the pile, respectively. During the recovery period (100 h–250 h), there is no heat injection. Figure 4.18 shows the estimated total heat output with respect to time for each model. For the 3D Model, the heat output is evaluated by integrating over the pipe inner surface along the depth of the GHE. For the 1D-3D Hybrid Model, it is calculated by integrating the 1D pipe elements along the depth of the GHE.

Similarly, for the 2D Model, it is estimated by line integration over the pipe inner boundary. On the other hand, for the FLS model, the heat output is constant during the heat injection period and zero during the recovery period. The results show that each model has a good agreement with the analytical method, which is the base case, having less than 1% difference.

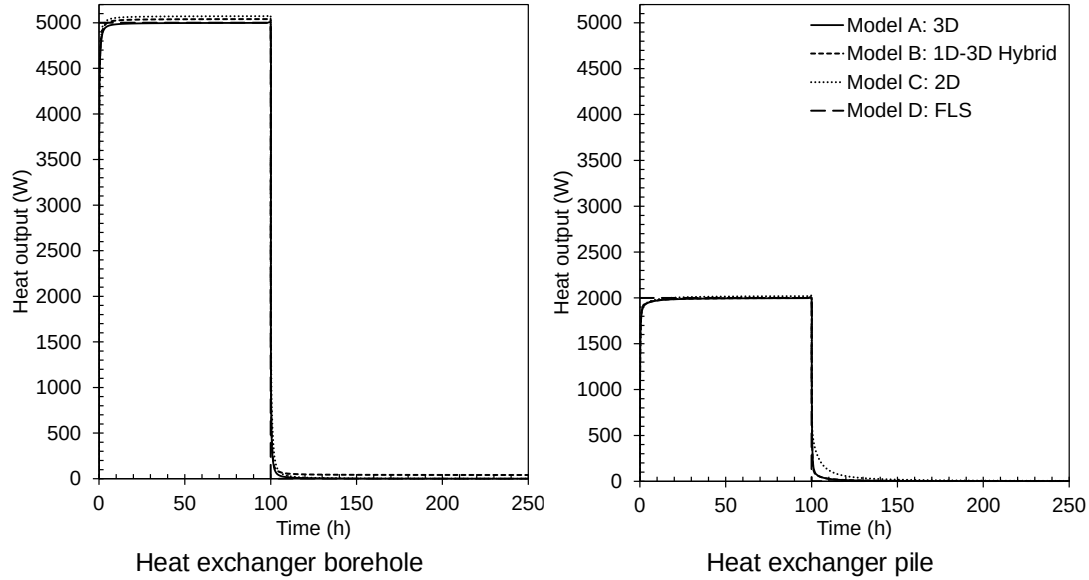


Figure 4.18 : Comparison of total heat output from the carrier fluid of the borehole and the pile estimated using proposed numerical models and the analytical FLS model.

The steady state thermal resistance of the GHE is calculated using Hellström's (1991) line source approximation as given by Equation (4.27). The steady state thermal resistance has a constant value during the heating period and it is zero during the recovery period. The obtained average temperature of the GHE wall is then used to evaluate the mean fluid temperatures during the analysis using Equation (4.23). On the other hand, GHE wall temperature in the 2D model is evaluated by integrating along the GHE wall boundary. For the 3D and 1D-3D hybrid models, it is estimated using a surface integration technique along the depth of the GHE. In the numerical models, overall thermal resistance of the GHE is calculated similarly as in the analytical method, using Equation (4.23). The estimated thermal resistance is transient in nature, since mean fluid temperature and the average wall temperature are time dependent. The comparison of the thermal resistances for the borehole and the pile estimated using the analytical method and the numerical models are given in Figure 4.19. Despite its smaller diameter, the thermal resistance of the borehole is almost twice of the energy

pile. This is caused by the fact that the pile material has a higher thermal conductivity and it is integrated with double U-tubes.

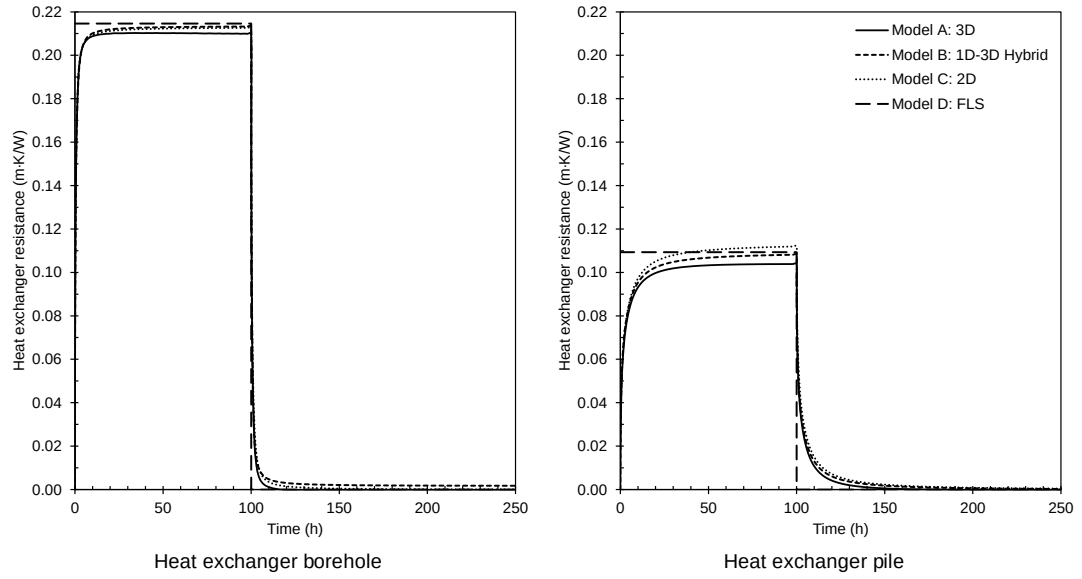


Figure 4.19 : Comparison of overall thermal resistances of the borehole and the pile estimated using proposed numerical models and the analytical FLS model.

The mean fluid temperatures and the borehole/pile wall temperatures estimated using all of the modeling approaches are presented in Figure 4.20. From Figure 4.19 and Figure 4.20, it can be seen that the analytical model and the numerical models result in similar overall thermal resistances and temperatures. For the borehole, the difference in the estimated thermal resistances is less than 2% and the fluid temperatures differ less than 1% compared to the analytical method. On the other hand, the deviation of the results is more prominent for the heat exchanger pile as the analytical method is selected as the base case. The thermal resistance and the fluid temperature evaluated by the 3D model differs approximately 5% and 2%, respectively. From these results, it is inferred that for both simulated cases, 1D-3D Hybrid Model (Model B) has the best prediction capabilities, with a relative error of less than 1%.

The formulation in Equation (4.22) is utilized for the analytical model and it is used to estimate the temperature variations along the depth of the GHE at various time/radial distance combinations, starting from the borehole/pile wall. These variations are computed at the borehole/pile wall, and at radial distances of 25 cm, 50 cm and 1 m from the GHE wall. The temperature evaluations are performed for the whole analysis

period with 5 hour time step increments. For the sake of simplicity, temperature profiles at 5 h, 10 h, 25 h, 50 h, 100 h, 150 h and 250 h are presented here.

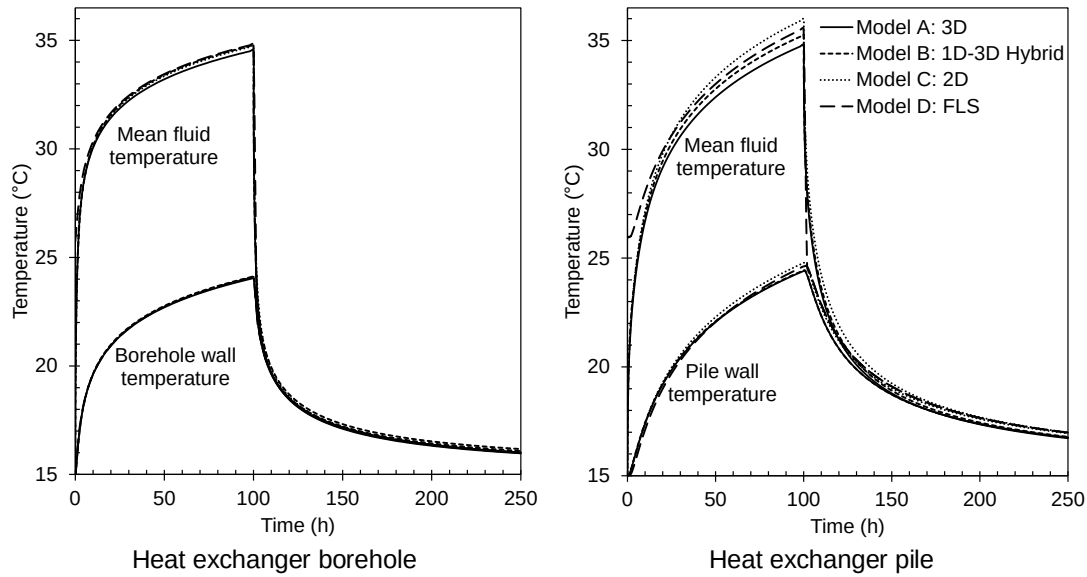


Figure 4.20 : Comparison of mean fluid temperature and wall temperature of the borehole and the pile estimated using proposed numerical models and the analytical FLS model.

Moreover, for some distance/time combinations, temperature plots of various times are not shown on the figures, because they either overlap with each other or the temperature changes at these times are insignificant. The temperature profile plots at various times for the borehole and the energy pile are shown in Figure 4.21 and Figure 4.22, respectively.

From Figure 4.21 and Figure 4.22, it can be inferred that the temperature profiles calculated using analytical and numerical models for the heat exchanger borehole have slightly better agreement than the heat exchanger pile. This behavior is expected due to the fundamental assumption of the finite line source model, where the diameter of the heat exchanger is in fact infinitely small. Thus, the prediction of the analytical method is more accurate for small diameter GHEs.

The temperatures at different times with respect to radial distance from the GHE wall are also evaluated at the mid-depth of the GHE. Figure 4.23 shows the comparison plots between two models for the borehole and the energy pile at 5 h, 25 h, 50 h, 100 h, 150 h and 250 h. Similar to the temperature profiles, analytical and numerical results of temperature vs. radial distance for the borehole are slightly closer. Nevertheless, all three numerical modeling approaches proposed here result in very similar values with

the analytical solution. The models serve as useful tools to better understand the performance of GHE systems with different diameters and tube configurations.

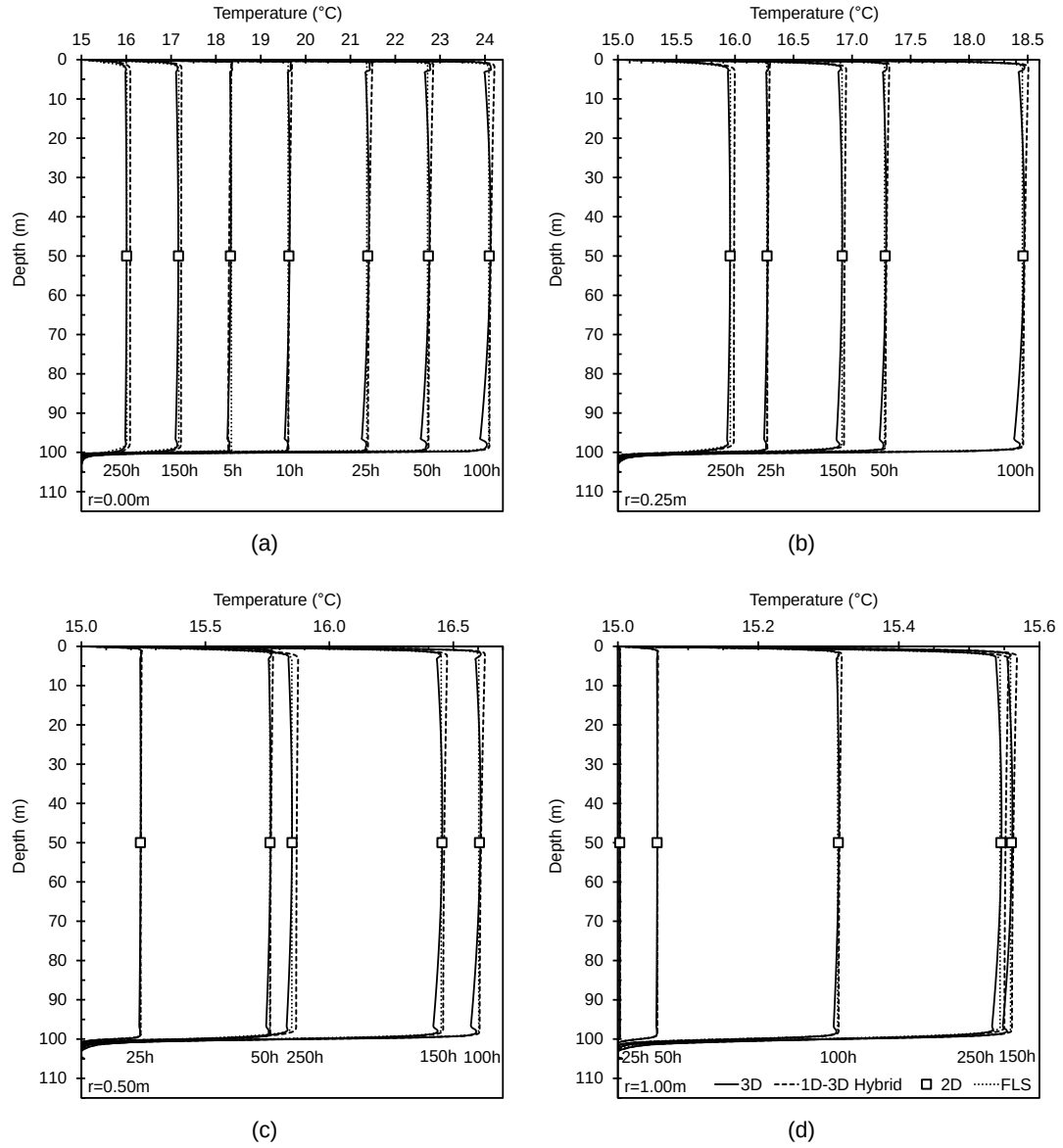


Figure 4.21 : Comparison between the temperature variations along the borehole depth estimated by the proposed numerical models and the analytical FLS model at: (a) borehole wall; (b) 25 cm away from the borehole wall; (c) 50 cm away from the borehole wall; (d) 1 m away from the borehole wall.

Furthermore, the numerical results are used to develop g-functions for the simulated borehole and the energy pile (Figure 4.24). The g-function model provides an efficient solution to simulate a single GHE or GHE fields with defined configurations over long timescales. The g-function is a normalized step response function that describes the temperature change at the GHE wall in response to a step heat input for a time step. It can be expressed as:

$$g\left(\frac{t}{t_s}, \frac{r_b}{H}\right) = \frac{2\pi k \cdot \Delta T_{bw}}{q'}, \quad t_s = H^2/9\alpha \quad (4.39)$$

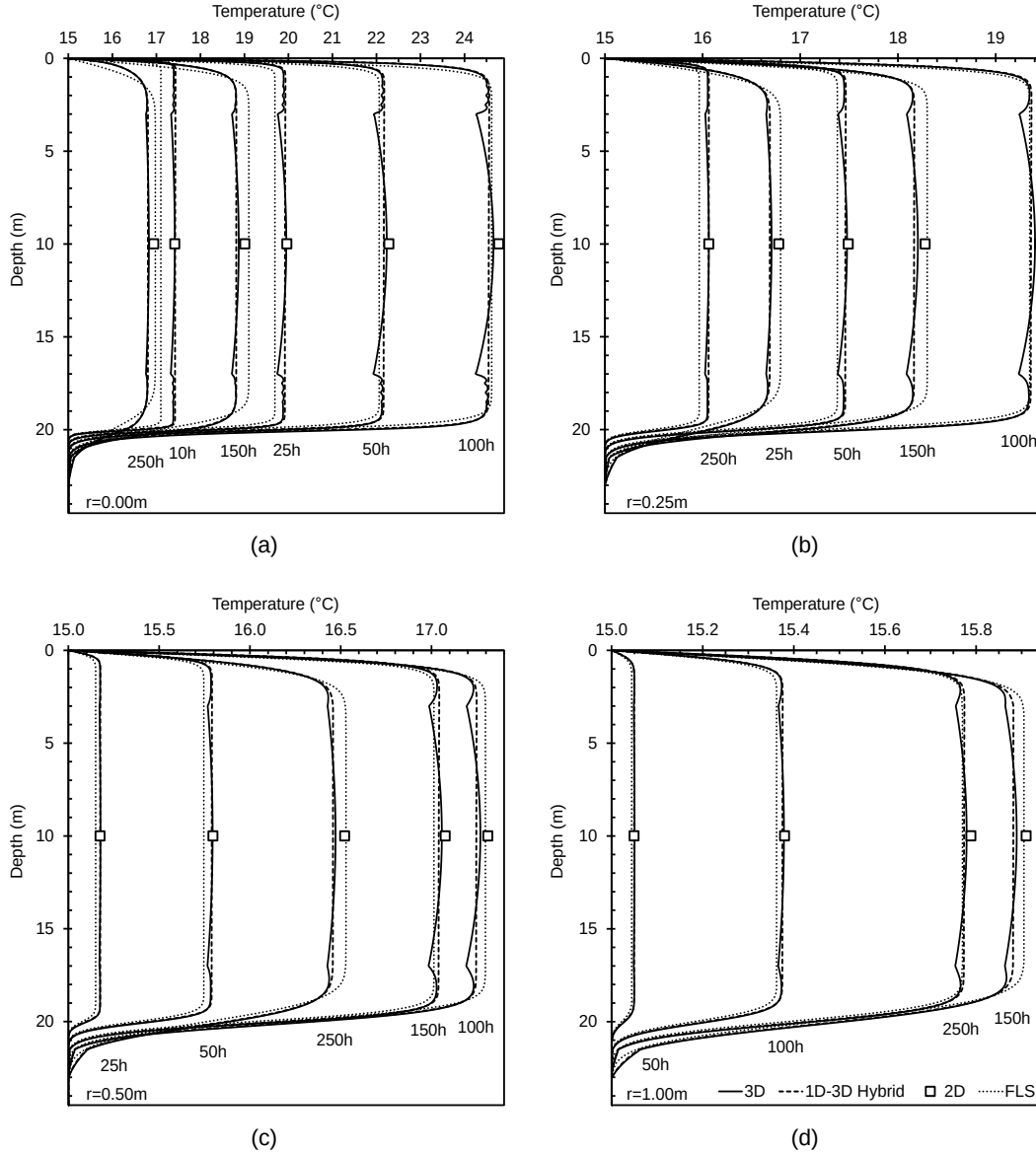


Figure 4.22 : Comparison between the temperature variations along the energy pile depth estimated by the proposed numerical models and the analytical FLS model at: (a) pile wall; (b) 25 cm away from the pile wall; (c) 50 cm away from the pile wall; (d) 1 m away from the pile wall.

Eskilson (1987) developed hundreds of long-step g-functions for multiple borehole heat exchangers arranged in different configurations. In this approach, the thermal conductivities and capacities of the materials inside the heat exchanger are neglected and as a result, the solution is only valid after a certain time, defined as:

$$t > \left(\frac{5r_b^2}{\alpha} \right) \quad (4.40)$$

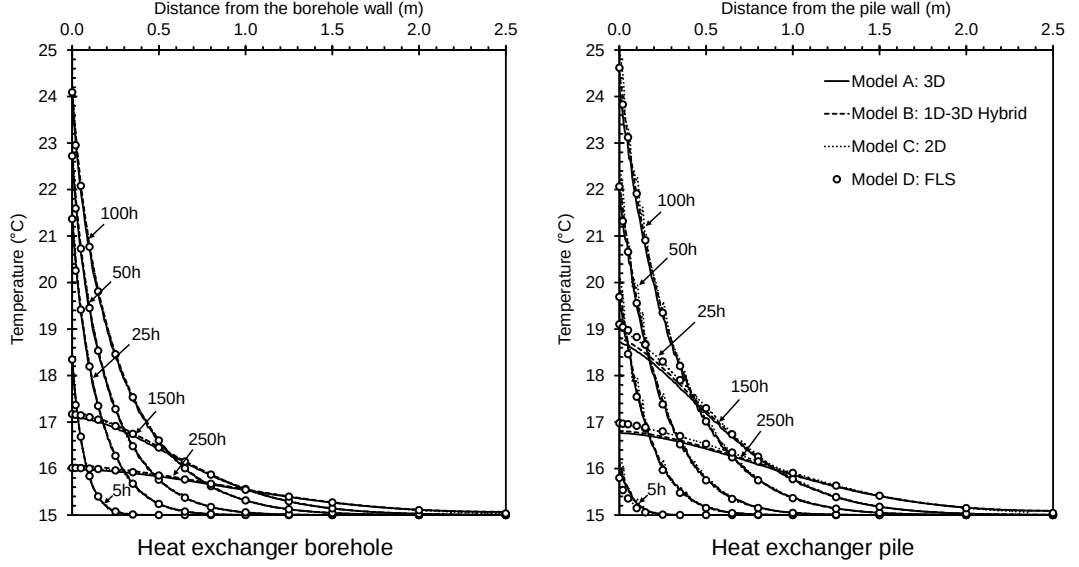


Figure 4.23 : Comparison between the temperature versus radial distance at mid-depth of the GHE estimated by the numerical models and the analytical FLS model.

Figure 4.24 shows the comparison of g-functions of the borehole and the energy pile computed using the analytical and numerical models. In order to investigate a wider range, the analytical method results are extended to 20 years of heating in both cases. Using Equation (4.40), the time limits indicating the validity of FLS model are computed as 11.7 hours and 105.5 hours for the borehole and the pile, respectively. It can be concluded that the g-functions computed with either model are in good agreement after the mentioned time limits in both cases.

Unlike the analytical solutions, the proposed 3D numerical models allow estimating the fluid temperatures along the pipes and also the temperature distribution inside the GHE. The fluid temperature profiles for the borehole and the energy pile along the depth at 5 h, 25 h, 50 h, 100 h, 150 h and 250 h are presented in Figure 4.25. Temperature contour plots within a cross-section at mid-depth of GHE at the end of heat injection period (at 100 h) are shown in Figure 4.26.

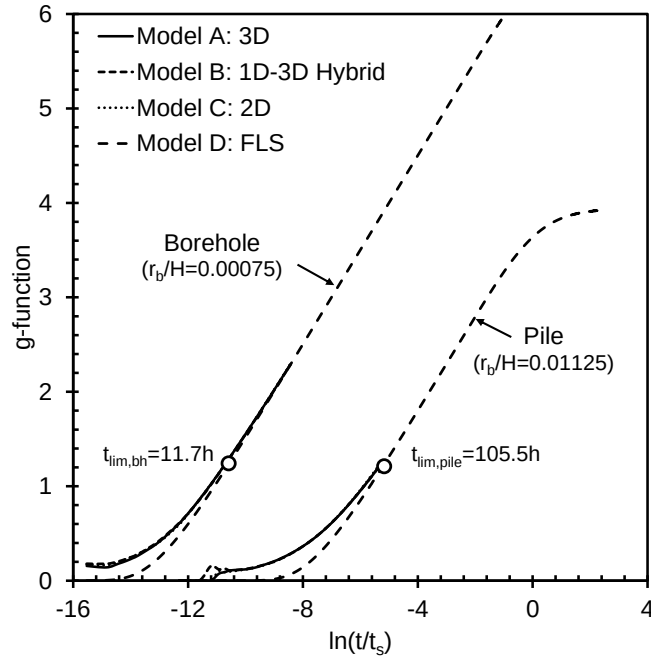


Figure 4.24 : Dimensionless g-functions for the borehole and the energy pile computed using the numerical models and the analytical FLS model.

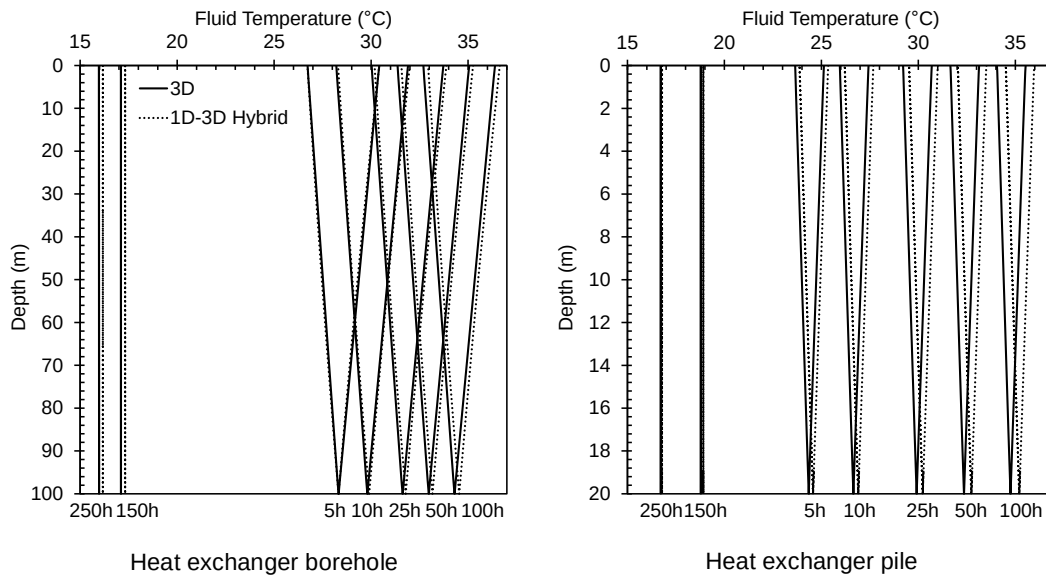


Figure 4.25 : Fluid temperature profiles along the depth of GHE estimated by the 3D Model and the 1D-3D Hybrid Model.

In addition to comparing the results of the proposed finite element models, it is evident that they differ in terms of definition of the numerical problem, such as the number of degrees of freedom and the number of finite elements. This is due to the differences in the complexities of the models and the symmetry conditions. These features also have an impact on the memory usage of the utilized computer and the solution time of each

problem. The comparison of the above mentioned properties are summarized in Table 4.3 for each numerical modeling approach.

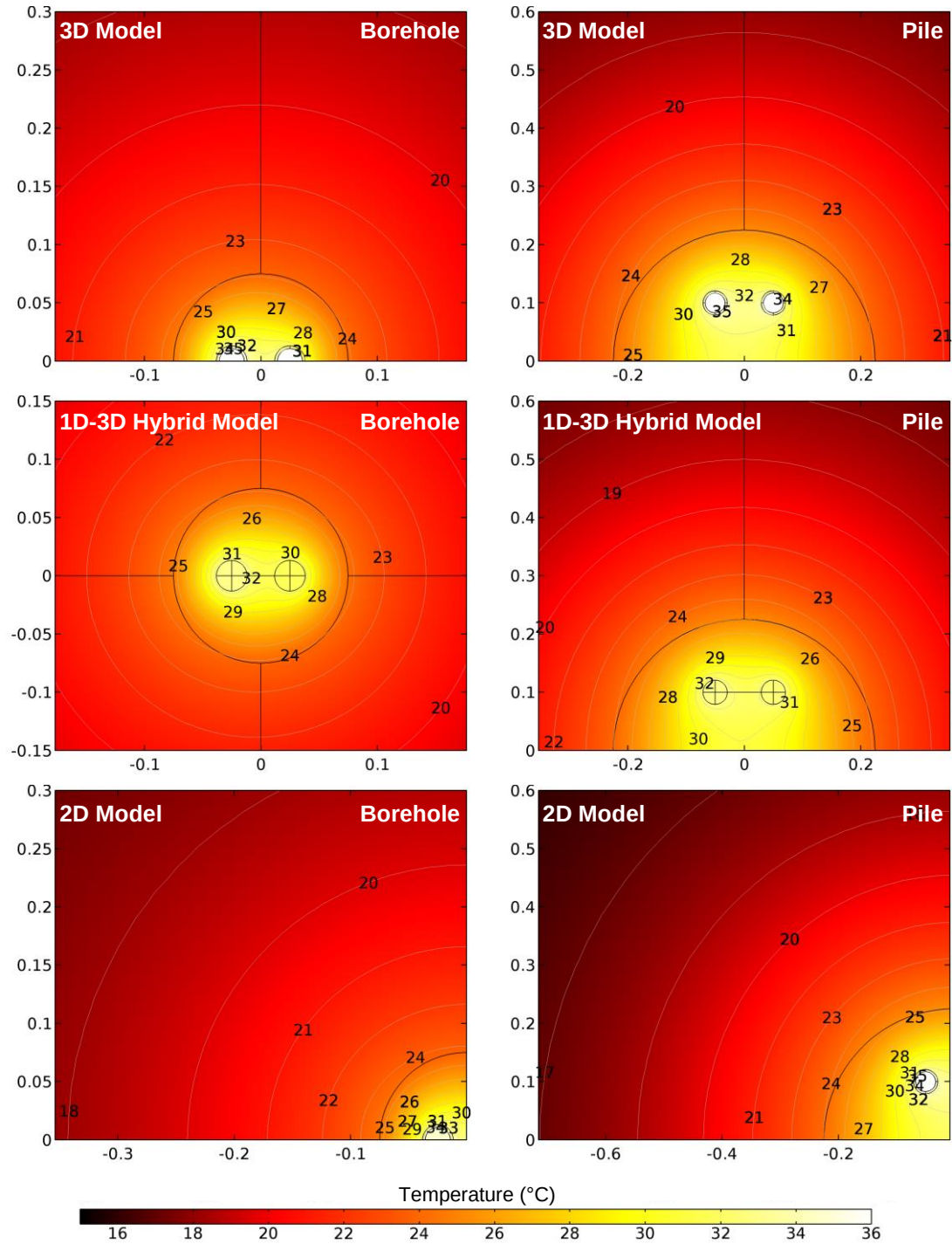


Figure 4.26 : Temperature contour plots within a cross-section at mid-depth of GHE at the end of heat injection period (at 100 h).

The solution time also greatly depends on the specifications of the computer. The PC that is used for solving the numerical models has a quad core processor with 8 available threads and 24 GB memory. The model and make of the CPU is Intel® Core™ i7-930 Processor. It has 8M cache, 2.80 GHz clock speed and 4.80 GT/s Intel® QPI speed.

The PC has six memory slots and each of them have the identical memory stick, which is 4 GB Kingston PC3-10700 DDR3-1333 SDRAM, running at 667 MHz speed.

Table 4.3 : Comparison of the proposed finite element modeling approaches in terms of degrees of freedom, number of finite elements, memory usage and solution time.

Description	Heat exchanger borehole			Heat exchanger pile		
	Model A	Model B	Model C	Model A	Model B	Model C
Symmetry	Half	–	Quarter	Half	Half	Quarter
Degrees of freedom	170297	308111	5099	206374	186912	5623
Domain elements	37586	74204	2420	44322	44538	2646
Boundary elements	9304	15634	191	10455	10157	200
Edge elements	953	1284	–	1037	995	–
Memory usage (GB)	2.33	5.95	0.51	2.23	2.92	0.50
Solution time (h)	216.79	6.18	0.01	7.64	1.79	0.01

5. CASE STUDIES OF FIELD THERMAL CONDUCTIVITY TESTS

In this chapter, case studies of several field thermal conductivity tests are presented. The first section discusses a field TCT performed at Virginia Tech test site (Ozudogru et al., 2015a). The test is simulated using a 2D finite difference (FD) model (Ghasemi-Fare and Basu, 2013a; Ghasemi-Fare and Basu, 2013b) and the proposed 1D-3D hybrid finite element (FE) model discussed in Section 4.1.2. In addition to modeling the field test, both numerical modeling approaches are used to simulate constant rate heat injection and constant temperature fluid injection scenerios for a generic vertical geothermal heat exchanger. For the former scenario, the models are used to back-calculate the soil thermal conductivity. The results obtained from these two numerical modeling approaches are compared with each other.

In the second section, the proposed 1D-3D hybrid FE model is used for simulating five thermal conductivity tests performed on three heat exchanger piles at Berkel site located in Richmond, TX. The detailed description of the test site and ground conditions is presented along with the analysis results, which are compared with the field measurements and discussed, as well.

5.1 Case Study 1: Comparison of Finite Difference and Finite Element Modeling Techniques: Numerical Modeling of a Field TCT Performed at Virginia Tech

The predictive capabilities of a 2D FD modelling approach and the proposed 1D-3D hybrid FE numerical model are demonstrated by simulating a field thermal conductivity test. The novelty of FD model lies in the idealization of governing mechanism for heat transfer through a vertical heat exchanger with embedded circulation tubes and in the implementation of relevant heat transfer equations within an axisymmetric analysis framework. Previous FD models that were developed to solve similar problems used a purely two-dimensional framework, which cannot accurately capture heat transfer through cylindrical heat sources (Yavuztürk et al. 1999; Rouissi et al. 2012); thermal resistance of the circulation pipes was also neglected in those studies. On the other hand, the FE model distinctively uses one-

dimensional (1D) linear elements for simulating the flow and heat transfer inside the circulation pipes, which is fully coupled with the 3D geometry using the temperature field at the pipe exterior surface. Both models can simulate the transient- and steady-state heat and mass transport processes involved in thermal operations of vertical GHEs with satisfactory accuracy and minimal computational effort.

The theory governing the thermal processes, element discretization and the selection of the appropriate boundary conditions have been briefly explained. The numerical models are verified through prediction of results from a field thermal conductivity test performed at Virginia Tech geotechnical field test site. Furthermore, both models are used to simulate the performance of a single U-tube borehole heat exchanger at a generic site. Two different thermal operation scenarios are considered; (1) constant rate heat injection, simulating a field thermal response test; and (2) constant-temperature fluid injection, simulating heat pump operation during building cooling. In each case, heat injection through a constant heat rate or a constant inlet temperature is followed by a recovery period (i.e., no injection). Predictions of FD and FE models are compared and a validation study is carried out by comparing the numerical results for the constant rate heat injection case with finite line source (FLS) solution.

5.1.1 Finite difference model

Heat transfer between vertical geothermal heat exchangers (GHE) and the surrounding soil is investigated by considering a vertical plane that passes through the center of the heat exchanger and contains both branches of the U-shaped circulation loop embedded within the heat exchanger. Heat transfer between the circulation fluid and the surrounding media (pipe, concrete/grout and soil) is quantified using a formulation approach using the finite difference (FD) framework. A schematic view of the FD analysis domain is shown in Figure 5.1. The FD model simultaneously solves partial differential equations for (1) heat balance within the circulation tube and (2) heat conduction within the heat exchanger and the soil. Time-dependent evolutions of temperature within the fluid, grout and the soil surrounding the heat exchanger are evaluated considering heat flow continuities at the material boundaries. Left and right branches of the circulation tube (i.e. downward and upwards legs) are considered as individual heat sources; however, fluid temperature at any depth of these heat sources is evaluated from the coupled solution of heat balance heat conduction equations.

Neglecting radial convection resistance of the fluid circulating inside the tube, heat balance equation for any element within a branch of the U-tube can be expressed as:

$$\frac{\partial T}{\partial t} = v \frac{\partial T}{\partial z} + \frac{2k_p}{\rho_f C_{pf} r_p} \frac{\partial T}{\partial r} \Big|_{r=r_w} \quad (5.1)$$

where ρ_f is mass density of the circulation fluid, k_p is the thermal conductivity of the circulation pipe, v is fluid circulation velocity, r_p is radius of the circulation pipe, C_{pf} is specific heat of the circulation fluid, and dT is an average increment in an element temperature within a small time interval dt .

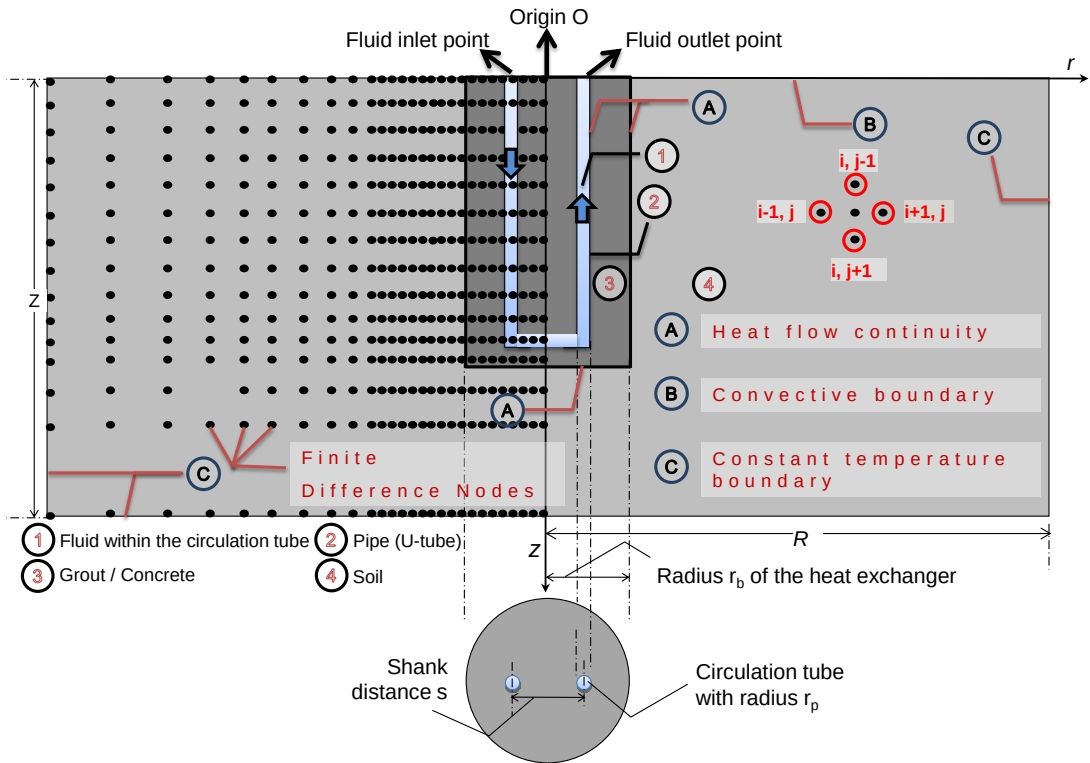


Figure 5.1 : Analysis domain for the finite difference model – finite difference grid and boundary conditions.

Temperature increment for any element within the media due to each of the U-tube branches (heat sources) can be calculated using the heat balance equation given by:

$$\frac{1}{\alpha} \frac{\partial T}{\partial t} = \frac{\partial^2 T}{\partial z^2} + \frac{\partial^2 T}{\partial r^2} + \frac{1}{r} \frac{\partial T}{\partial r} \quad (5.2)$$

where α is thermal diffusivity, ρ is mass density, t is time, and r and z are the radial and vertical coordinates, respectively. Coupled solution of Equation (5.1) and Equation (5.2) can capture heat transfer along the length of the circulation tube

branches and the heat conduction within the soil and the grout. However, heat flux continuity equation is needed to couple Equation (5.1) and Equation (5.2).

Using an explicit solution scheme for two consecutive time steps, finite difference forms of the governing partial differential equations Equation (5.1) and Equation (5.2) can be written as:

$$T_{i,j}^{t+1} - T_{i,j}^t = v \frac{\Delta t}{\Delta z_j} (T_{i,j-1}^t - T_{i,j}^t) + \frac{2k_t \Delta t}{\rho_f C_{pf} r_p \Delta r_i} (T_{i+1,j}^t - T_{i,j}^t) \quad (5.3)$$

$$\frac{T_{i,j}^{t+1} - T_{i,j}^t}{\alpha_{i,j} \Delta t} = \left[\frac{-2r_{i,j}^L \frac{\partial T}{\partial r} \Big|_{r=r_{i,j}^L} + 2r_{i+1,j}^L \frac{\partial T}{\partial r} \Big|_{r=r_{i+1,j}^L} - 2r_{i,j}^R \frac{\partial T}{\partial r} \Big|_{r=r_{i,j}^R} + 2r_{i+1,j}^R \frac{\partial T}{\partial r} \Big|_{r=r_{i+1,j}^R}}{\left[(r_{i+1,j}^L)^2 - (r_{i,j}^L)^2 \right] + \left[(r_{i+1,j}^R)^2 - (r_{i,j}^R)^2 \right]} \right] + \left(\frac{\partial^2 T}{\partial z^2} \right) \Big|_{(i,j)} \quad (5.4)$$

where, i and j are nodal indices, r^L and r^R are, respectively, radial distances measured from the centers of the left and right branches of the circulation tube, and Δt is the time step interval.

As the values of thermal conductivity and thermal diffusivity of soil, grout and pipes are different, heat flow continuity conditions are required at the pipe-grout interface (i.e. at $r = \pm (0.5s \pm r_p)$; r_p is the radius of the pipe and s is the shank spacing) and at the grout-soil interface (i.e. at $r = \pm r_b$; r_b is the radius of the borehole heat exchanger). The finite difference form of the heat flow continuity condition at the grout-soil interface is given in Equation (5.5). Similar continuity condition is used at the pipe-grout interface as well.

$$\left(\frac{T^{t+1} - T^t}{\Delta t} \right) \left[\frac{(r_i^2 - r_{i-1}^2) \left(\frac{k_g}{\alpha_g} \right) + (r_{i+1}^2 - r_i^2) \left(\frac{k_s}{\alpha_s} \right)}{(r_{i+1}^2 - r_{i-1}^2)} \right] = \frac{1}{r_i} \left[\frac{k_s r_{i+1} \left(\frac{T_{i+2}^t - T_{i+1}^t}{r_{i+2} - r_{i+1}} \right) - k_g r_{i-1} \left(\frac{T_i^t - T_{i-1}^t}{r_{i+2} - r_{i+1}} \right)}{(r_{i+1} - r_{i-1})} \right] + k_s \left(\frac{r_{i+1}^2 - r_i^2}{r_{i+1}^2 - r_{i-1}^2} \right) \frac{\partial^2 T}{\partial z^2} + k_g \left(\frac{r_i^2 - r_{i-1}^2}{r_{i+1}^2 - r_{i-1}^2} \right) \left(\frac{\partial^2 T}{\partial z^2} \right) \quad (5.5)$$

Convective boundary condition is used at the top boundary of the analysis domain, and constant temperature boundary condition is used for the left, right and bottom boundaries (Figure 5.1). Convective boundary condition for the ground surface and constant temperature boundary for the bottom side are defined in Equation (5.6) and Equation (5.7), respectively.

$$T_{i,j}^{t+1} - T_{i,j}^t = \alpha_{i,j} \Delta t \left[\frac{\partial^2 T}{\partial r^2} + \frac{1}{r_i} \frac{(T_{i+1,j}^t - T_{i,j}^t)}{\Delta r_{i+1}} - \frac{1}{\Delta z_{j+1}} \left(\frac{(T_{i,j+1}^t - T_{i,j}^t)}{\Delta z_{j+1}} + h_{\text{top}} (T_{i,j}^t - T_{\text{ag}}) \right) \right] \quad \text{for } -R < r < R, z = 0 \quad (5.6)$$

$$T = T_{\text{initial}} \quad \text{for } -R \leq r \leq R, z = Z \quad (5.7)$$

where h_{top} is a convective heat transfer coefficient through the top boundary of the analysis domain and T_{ag} is the temperature of the medium above ground surface.

Time step size Δt that can be used in the FD formulation depends on the grid density. To avoid solution instability, maximum time step used in the analyses was restricted to that derived using Courant-Friedrichs-Lewy stability criterion (Courant et al., 1967).

5.1.2 Verification of the numerical models

Both FD and FE models are validated using data recorded during one of the field thermal conductivity tests presented by Abdelaziz (2013). The test site was located at the Virginia Tech geotechnical testing facility. Four heat exchanger piles and a reaction pile were installed at the site and several thermal conductivity and thermo-mechanical loading tests were performed. Two types of thermal conductivity tests were conducted at the test site: (1) in accordance with ASHRAE method (Kavanaugh et al., 2001) and (2) following the Dutch perspective (Witte et al., 2002).

During the ASHRAE tests, the circulation fluid was heated with a constant heat rate using the electric heaters located at the testing trailer. Even though insulation was present, the piping above ground, (i.e. the pipes between the top of the pile and the trailer), was affected by variations in the ambient air temperature. This resulted in the application of a variable heat rate in the pile and requires advanced analysis techniques for estimating the thermal conductivity (Bandos et al., 2011). On the other hand, the Dutch test was conducted with a temperature controller device. The temperature difference between the inlet and outlet fluids was maintained at a constant value

throughout the testing period, which provides a constant heat rate. The thermal conductivity test that followed the Dutch approach is considered in the verification study because such tests can be interpreted using available analytical heat transfer solutions, and thus provide a higher level of confidence in the verification study.

The test pile had a diameter of 25.4 cm and a length of 30.48 m. The pile was integrated with a single U-pipe, which had shank spacing (center-to-center) of 7.5 cm. The drilling log indicated that the first 12.8 m of the soil profile consists of a silty clayey sand layer, underlain by a shale layer. The shale layer extends beyond the depth of pile base. The subsurface profile at the test site and geometry of the test pile, as used in both numerical models, are shown in Figure 5.2.

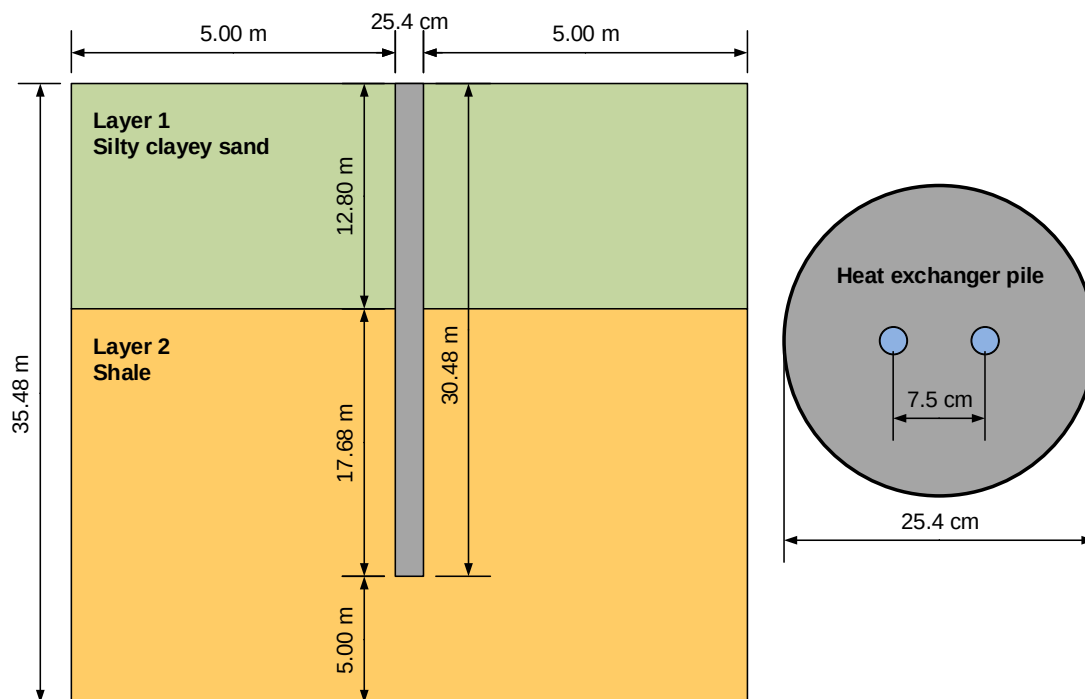


Figure 5.2 : Subsurface profile and layout at Virginia Tech field test site and the dimensions used in the analyses.

The undisturbed ground temperature was measured as 14.7°C. The duration of the test was approximately 50 hours. Water was used as the circulation fluid and the temperature difference between the inlet and outlet was kept as 5.56°C during the entire testing period. This temperature change corresponds to a constant heat rate of 72 W/m. The input parameters used in the numerical models are given in Table 5.1. A discussion on the estimation of relevant material properties is provided in Abdelaziz (2013).

Table 5.1 : Input parameters for numerical analyses to simulate field test at Virginia Tech (adopted from Abdelaziz, 2013).

Parameter	Value	Unit
Global parameters		
Initial ground temperature	14.7	°C
Heat rate per depth	72	W m ⁻¹
Test duration	50	h
Circulation fluid (water)		
Flow rate	5.68	dm ³ min ⁻¹
Dynamic viscosity	0.9772	mPa s
Thermal conductivity	0.6048	W m ⁻¹ K ⁻¹
Specific heat capacity	4180	J kg ⁻¹ K ⁻¹
Density	997.8	kg m ⁻³
Pipes		
Thermal conductivity	0.40	W m ⁻¹ K ⁻¹
Specific heat capacity	2300	J kg ⁻¹ K ⁻¹
Density	940	kg m ⁻³
Pipe inner diameter	21.844	mm
Pipe wall thickness	2.413	mm
Shank spacing (center-to-center)	7.5	cm
Borehole		
Thermal conductivity	1.28	W m ⁻¹ K ⁻¹
Specific heat capacity	880	J kg ⁻¹ K ⁻¹
Density	1600	kg m ⁻³
Diameter	25.4	cm
Length	30.48	m
Ground layer 1 (silty clayey sand)		
Thermal conductivity	1.0	W m ⁻¹ K ⁻¹
Specific heat capacity	1500	J kg ⁻¹ K ⁻¹
Density	1900	kg m ⁻³
Layer thickness	12.80	m
Ground layer 2 (shale)		
Thermal conductivity	2.90	W m ⁻¹ K ⁻¹
Specific heat capacity	1200	J kg ⁻¹ K ⁻¹
Density	2400	kg m ⁻³
Layer thickness	22.68	m

The inlet and outlet temperatures were recorded during the field test. The results obtained from the numerical analyses are presented in Figure 5.3, along with the field measurements. The result of the verification study shows that both FD and FE models predicted the measured fluid temperatures with good accuracy, while the FE model worked slightly better. The coefficient of determination (R^2) of the FE and FD models in comparison to the field measurements are evaluated as 0.995 and 0.950, respectively.

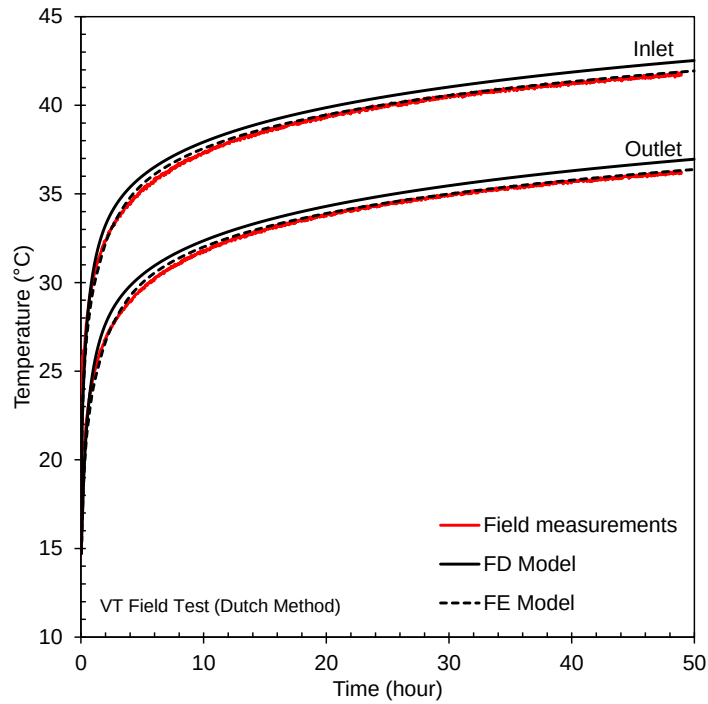


Figure 5.3 : Comparison of inlet and outlet temperatures predicted by FD and FE models with the measurements recorded during the field thermal conductivity test.

5.1.3 Analysis results

Two different thermal operation scenarios are studied using both the 2D finite difference and 1D-3D hybrid finite element models. First, a constant rate heat injection operation of a geothermal borehole (i.e., a thermal response test) is simulated using both models. The results obtained by FD and FE models are compared with the FLS analytical solution. After validating both models, additional analyses are performed where GHE operation with constant-temperature fluid injection and recovery is simulated.

100-m-long geothermal borehole with a diameter of 0.15 m is simulated in this study. The borehole is integrated with a single U-tube heat exchanger element with 21 mm inside diameter of and 2.4 mm wall thickness. The shank spacing between the two branches of the U-tube is equal to 50 mm. For the first 100 hours of the operation, either a constant heat (5000 W) or constant temperature (35°C) was injected to the ground through the heat exchange fluid. A recovery phase of 150 hours immediately after the constant rate heat and constant-temperature fluid injection phases is also simulated using both FD and FE models. During the recovery phase, the fluid temperature drastically decreases to reach back to its initial value since there is no heat injection into the system. Initial fluid temperature was considered to be equal to the

initial ground temperature, which is 15°C. A mixture of water and propylene glycol with a concentration of 20% by volume is used as the heat carrier fluid. The thermal and rheological properties of the water-antifreeze solution are taken from the ASHRAE Handbook Fundamentals (ASHRAE, 2009). The thermal properties of the ground and borehole material are selected by referring to the studies by Farouki (1981) and Salomone et al. (1989). The input parameters used in the numerical models are presented in Table 5.2.

Table 5.2 : Input parameters used in the numerical analyses to simulate the constant rate heat injection and constant-temperature fluid injection scenarios.

Parameter	Value	Unit
Global parameters		
Initial ground temperature	15	°C
Heat rate per depth (constant heating case)	50	W m ⁻¹
Injection temperature (constant temperature case)	35	°C
Circulation fluid		
Flow rate	20	dm ³ min ⁻¹
Dynamic viscosity	2.02	mPa s
Thermal conductivity	0.48	W m ⁻¹ K ⁻¹
Specific heat capacity	3962	J kg ⁻¹ K ⁻¹
Density	1020.9 1	kg m ⁻³
Pipes		
Thermal conductivity	0.39	W m ⁻¹ K ⁻¹
Specific heat capacity	2300	J kg ⁻¹ K ⁻¹
Density	960	kg m ⁻³
Pipe inner diameter	21.5	mm
Pipe wall thickness	2.4	mm
Borehole		
Thermal conductivity	1.00	W m ⁻¹ K ⁻¹
Specific heat capacity	1600	J kg ⁻¹ K ⁻¹
Density	1500	kg m ⁻³
Diameter	0.15	m
Length	100	m
Ground		
Thermal conductivity	2.00	W m ⁻¹ K ⁻¹
Specific heat capacity	1500	J kg ⁻¹ K ⁻¹
Density	2000	kg m ⁻³

5.1.3.1 Constant rate heat injection scenario (field thermal response test)

FD and FE analyses are used to model a constant rate heat injection of 5000 W, thus simulating a thermal response test. For modeling the constant heat injection, the fluid temperature at the inlet is increased by applying a temperature difference ΔT which is expressed as:

$$T_{\text{in}}(t) = T_{\text{out}}(t) + \Delta T(t) \quad (5.8)$$

The temperature difference between the circulation tube inlet and outlet points as well as the mass flow rate of the circulation fluid are maintained constant throughout the heating period to simulate a constant rate of heat injection into the system. On the other hand, the temperature difference ΔT is zero during the recovery period. For the heating period, ΔT for a given heat injection rate is calculated using:

$$q = \dot{V} \rho_f c_{pf} \Delta T$$

$$\Delta T = q / (\dot{V} \rho_f c_{pf}) \quad (5.9)$$

Figure 5.4 shows the variation of heat flux along both legs of the circulation tube at three different time periods computed from the FD and FE simulations of 5000 W constant rate heat injection into the 100 m long geothermal borehole. As expected, heat flux along the circulation tube is nearly constant with time, showing very small variation between the considered simulation periods. This indicates that the circulation tube induces very little resistance to heat transfer.

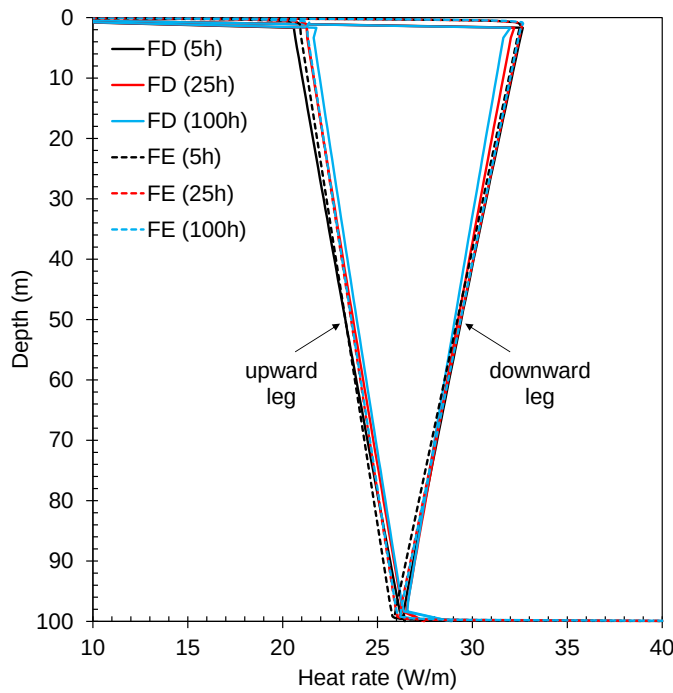


Figure 5.4 : Variation of heat flux along both legs of the circulation tube for the constant rate heat injection scenario at three different time steps.

On the other hand, as it can be seen in the figure, heat flux is different between the different legs of the circulation tube along borehole depth. The heat flux along the downward leg leading from the fluid inlet is generally higher than the heat flux along the upward leg leading toward the fluid outlet from the borehole. However, the total

heat flux at each elevation is somewhat constant as the total heat flux from the downward and upward legs is nearly equal to 50 W/m throughout the borehole depth. Therefore, a constant heat rate can also be applied along the length of the borehole to simulate a thermal response test using the FLS solution. It is also instructive to note that the FD and FE simulations can resolve the GHE behavior to detail levels not available with the analytical solutions. Since the length of the geothermal borehole is 100 m , the results obtained from FD and FE analyses are compared with an idealized FLS solution considering 50 W/m heat injection rate per unit length of the line source model.

Comparison of the ground temperature responses obtained by the FD and FE models and the calculated ground temperature increments by the FLS solution are presented in Figure 5.5. Average of the computed temperatures at the inlet (i.e. downward leg) and outlet (i.e. upward leg) sides of the FD model are shown in the figure. Similarly, the radial averages of the computed temperatures at different distances within the 1D-3D hybrid FE model are shown. It is seen that the results obtained by both FD and FE models are in good agreement with the analytical solution. Therefore, it can be concluded that both models can predict the results accurately and they can be used to simulate the performance of geothermal boreholes during heat exchange operations.

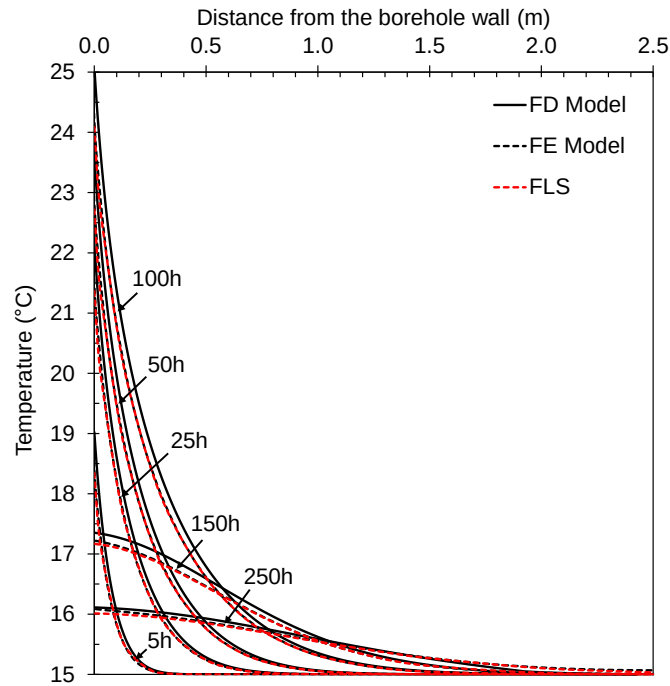


Figure 5.5 : Comparison of ground temperatures calculated by FD (average of inlet and outlet sides) and FE (radial average) models with the analytical FLS solution for the constant rate heat injection scenario.

Heat fluxes in both models computed from the temperature gradients along the borehole wall are compared in Figure 5.6. Heat flux along the borehole wall at the inlet side (i.e. downward leg) is higher compared to the heat flux at the outlet side (i.e. upward leg). Thus, ground temperature increment at the inlet side is expected to be higher than that at the outlet side especially at close distances. This difference is expected to decrease with distance away from the borehole. It is also worth mentioning that the heat fluxes at the inlet and outlet sides for the considered time periods are both higher than the average heat injection rate (50 W/m). This is due to the close proximity of these locations to these heat source (i.e. circulation tube). The average heat flux around the borehole wall computed from the 1D-3D hybrid FE model is also shown in Figure 5.6. As seen, the average heat flux around the borehole is about 45 W/m heat flux at 5 hours and increases to about 50 W/m at 100 hours of operation. Although the total injected heat rate was constant during the whole operational time (100 hours), the heat flux at the borehole wall is lower at 5 hours compared to 25 and 100 hours of the heat injection period. This can be explained by the thermal resistance of the borehole, as it takes some time for the heat flux inside the borehole to overcome the borehole resistance and reach the ground. Initially the total heat injected from the borehole to the ground will be lower than 5000 W, and as time passes the heat transfer inside the borehole reaches to a nearly steady-state condition approaching the heat injection rate.

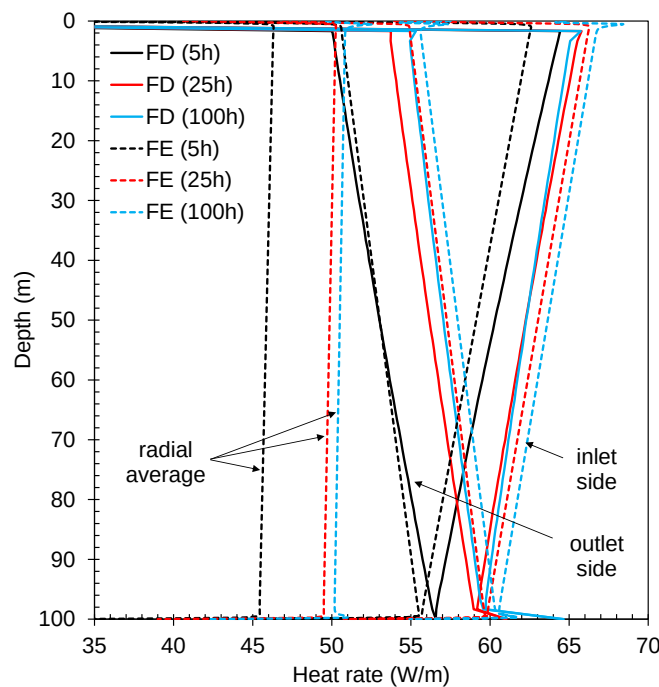


Figure 5.6 : Variation of heat flux along the borehole wall for the constant rate heat injection scenario.

Apart from the heat injection rate and the ground temperature response, variations of fluid temperatures obtained by FD and FE analyses are also compared in this study. Figure 5.7a shows that average of the inlet and outlet fluid temperatures computed by both models are in good agreement. The maximum difference between the FD and FE predictions is 0.5°C. It can be also inferred that the fluid temperatures obtained by both models are in complete agreement during the recovery phase, which starts after 100 hours or operation. Figure 5.7b shows the variation of fluid temperatures along both legs of the circulation tubes at different times of the simulations during the operation and recovery phases. Similarly, the fluid temperatures from FD and FE models are in close agreement.

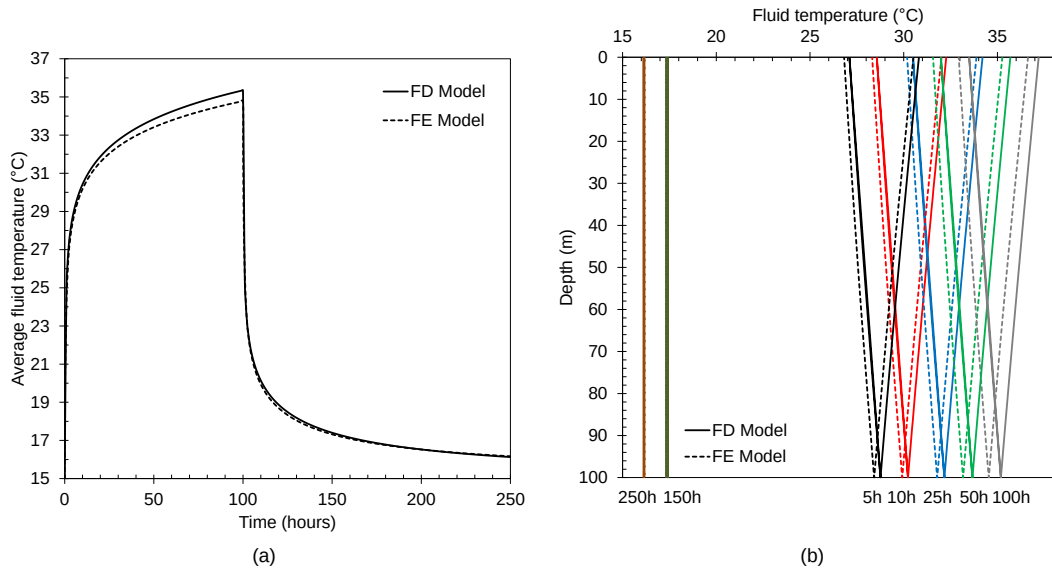


Figure 5.7 : Fluid temperature variation for the constant rate heat injection scenario (a) average fluid temperature with time and (b) variation of fluid temperature with depth along the circulation tube.

Since FD and FE models are utilizing two completely different computational platforms and model idealization approaches, it is interesting to note that the results obtained from both models (e.g., heat flux, ground temperature response, and fluid temperature) are in good agreement. On the other hand, the slight difference in the prediction of the numerical models can be explained from the basic modeling assumptions adopted in these models. The analysis domain in the FD model is a two-dimensional axisymmetric cross section passing through centerline axis of the geothermal heat exchanger (Figure 5.1), whereas, the FE model considers a 3D domain surrounding the geothermal borehole. Furthermore, the temperature profiles presented in Figure 5.5 are evaluated by averaging the temperature values obtained at mirror

points on the inlet and outlet sides for the FD model; however, the results from the FE model represents an average of temperature evaluated along a circle with radius of the distance of concern, relative to the borehole wall.

5.1.3.2 Constant-temperature fluid injection scenario (heat exchanger operation)

The FD and FE models are also used to analyze a constant-temperature fluid injection scenario which is representative of the thermal performance of a GHE during building cooling operation. Same thermal properties and dimensions which were used for the thermal response test analysis are also used for the constant-temperature fluid injection case (Table 5.2). However, instead of applying a temperature difference at the inlet to simulate the constant heat rate, a constant inlet temperature of 35°C is used to simulate a constant-temperature heat injection operation. Evolution of the outlet fluid temperature with time is shown in Figure 5.8a. It is seen that both models yield very similar values as the outlet fluid temperature rapidly increases from the initial value (15°C) towards the injection fluid temperature (35°C). Fluid temperature rapidly drops after heat injection is stopped. Similarly, temperature variations along the borehole depth obtained by FE and FD models at different times of the simulations during the operation and recovery phases are presented in Figure 5.8b respectively. It can be seen that the fluid temperatures obtained by both models (FE and FD) are in complete agreement.

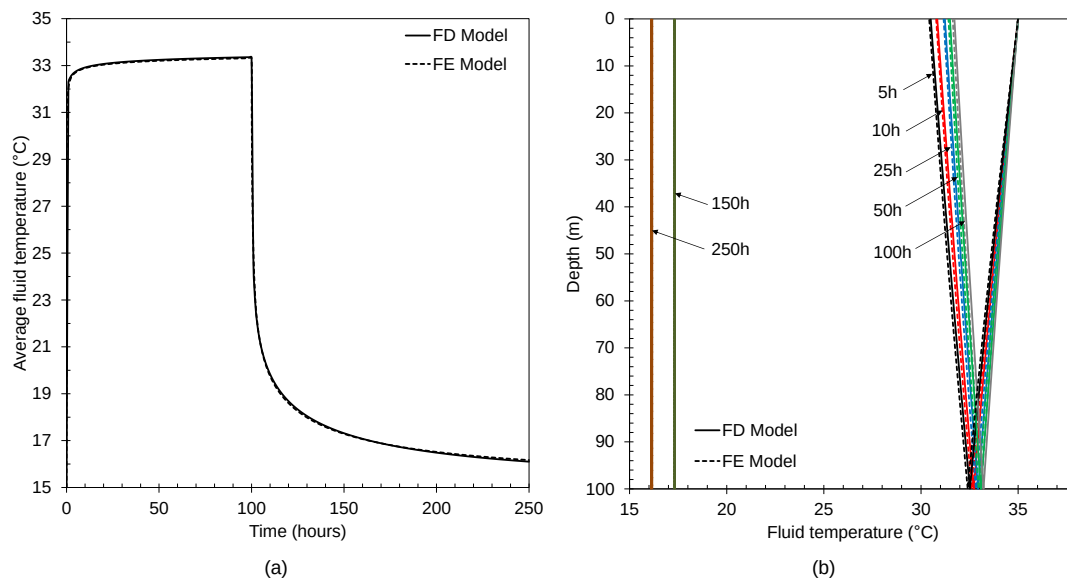


Figure 5.8 : Fluid temperature variation for the constant-temperature fluid injection scenario (a) average fluid temperature with time and (b) variation of fluid temperature with depth along the circulation tube.

Figure 5.9 shows the heat fluxes along the both legs of the circulation tube calculated by both models and it is seen that the results are in good agreement. Heat flux from the downward leg is higher than the heat flux from the upward leg. In contrast with the constant rate heat injection scenario (Figure 5.4), heat flux during the constant-temperature injection scenario decreases by 25% from 5 hours to 100 hours of operation. This means that the temperature difference between the inlet and outlet fluid decreases with time, as the temperature of the heat injection fluid is kept constant. This is also indicative that the rate of heat energy injection to the ground will gradually reduce during building cooling operations using GHE. However, typical heat injection lasts about 6-8 hours during these times of operations and then the system is turned off allowing the system to recover. Therefore any loss in system capacity during typical GHE operation is expected to be minimal compared to the 100 hour simulation presented herein. Finally, ground temperature responses obtained from the FE and FD models are also compared for the constant-temperature fluid injection scenario. The comparison is presented in Figure 5.10 and it is seen that both models are in good agreement as the maximum ground temperature difference between FE and FD models is 0.5°C.

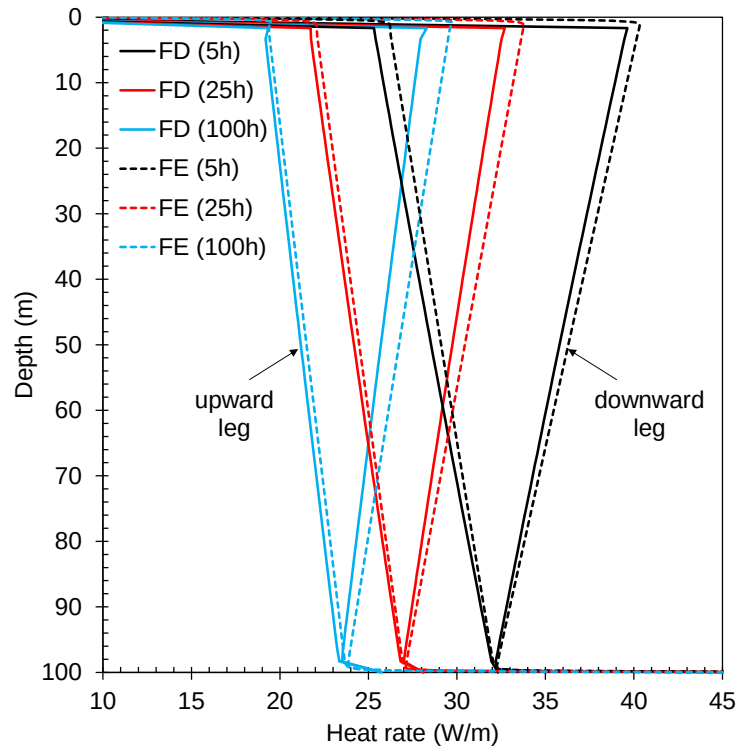


Figure 5.9 : Variation of heat flux along both legs of the circulation tube for the constant-temperature fluid injection scenario at three different time steps.

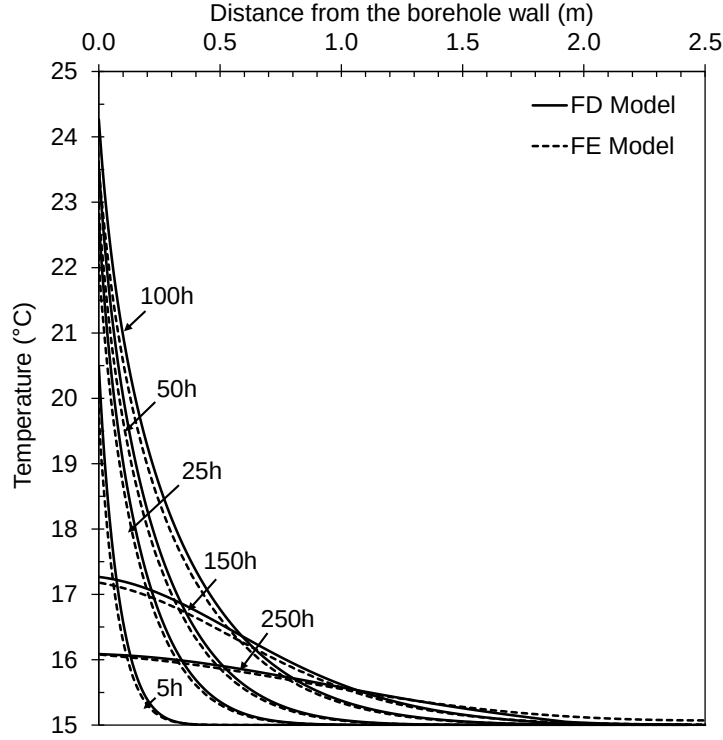


Figure 5.10 : Ground temperatures calculated by FD (average of inlet and outlet sides) and FE (radial average) models for the constant-temperature fluid injection scenario.

5.1.3.3 Back-calculation of soil thermal conductivity

Results obtained from the numerical analyses of the constant rate heat injection scenario are used to calculate effective soil thermal conductivity. A transient method presented by Gehlin (1998) and Austin (1998) which considers the FLS solution is used to estimate the effective soil thermal conductivity. The effective thermal conductivity is obtained using the following expression:

$$k_{eff} = \frac{q}{4\pi KH} \quad (5.10)$$

where K is the slope of the average fluid temperature versus log time curve, and it is estimated by fitting a regression line for the considered period. In this study, 10 different time intervals were selected, ranging from 5-100 hours to 90-100 hours. Back-calculated effective soil thermal conductivity from both FE and FD analyses are compared with the soil thermal conductivity considered in the analyses. Figure 5.11 shows the calculated effective thermal conductivity (k_{eff}) values for these different time periods in comparison to the value used in the numerical analyses ($k = 2.0 \text{ W/mK}$). Similarly, the ratios of the calculated effective thermal conductivity values over the

actual soil thermal conductivity are also presented in the same figure. Estimated effective soil thermal conductivity from both finite element and finite difference models are quite accurate as the relative difference between effective thermal conductivity calculated from the results and the actual soil thermal conductivity used in the models is less than 9%. On the other hand, effective thermal conductivity obtained from the FE model shows a better match (within 1.5%) with the actual soil thermal conductivity. Nonetheless, it is important to point out that the FD model consumes lower computational time when compared to the 1D-3D hybrid FE model used in this study. For analyses presented in this section, computational time for FD analyses were around one-third to one-fourth of that needed for the FE analyses.

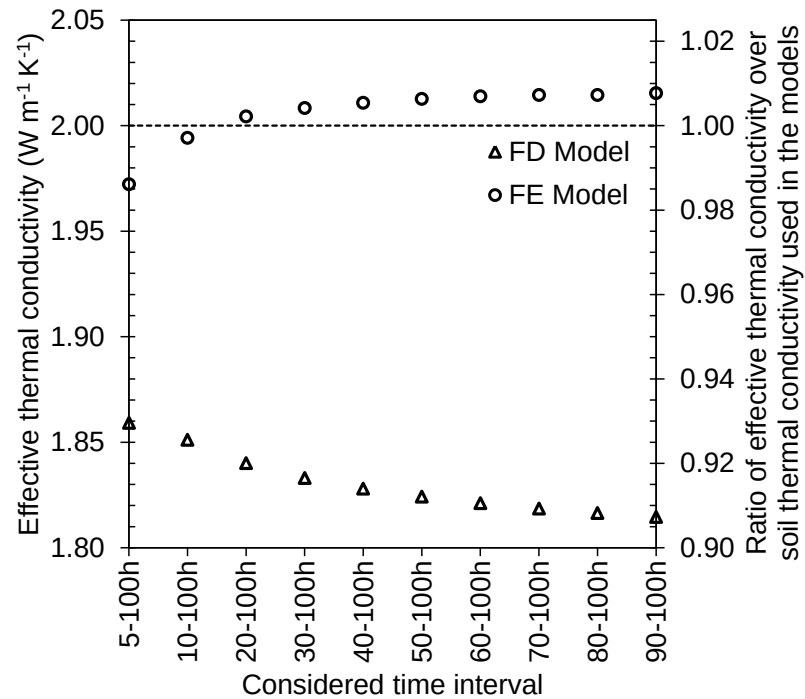


Figure 5.11 : Back-calculated effective soil thermal conductivity (k_{eff}) and the ratio of the back-calculated thermal conductivity over the actual soil thermal conductivity in the models.

5.1.4 Discussion

Verification of both models are successfully performed using the results from a full-scale field thermal conductivity test conducted at Virginia Tech. The models are further utilized to analyze borehole heat exchanger at a generic site. Two different operational scenarios are considered. First, a constant rate heat injection scenario, which corresponds to a field thermal response test, was simulated. Numerical analysis results of constant rate heat injection scenario are validated in comparison with the

finite line source analytical solution. The second scenario considered a constant-temperature fluid injection, simulating the operation of a geothermal heat pump during building cooling. In each scenario, a thermal operation period of 100 hours followed by a recovery period of 150 hours, during which heating was stopped, was considered. Numerical analysis results were compared through the quantification of average fluid temperature, fluid temperature profiles at several times, heat fluxes along the circulation tube and along the borehole, and ground temperature profiles at mid-depth of the heat exchanger at various times. Furthermore, following a transient method proposed by Gehlin (1998) and Austin (1998), effective ground thermal conductivity is back-calculated using the numerical simulation results for the constant rate heat injection case. Comparison between predictions using FD and FE models and validation of these models against field test data and analytical solution confirms that both models can simulate thermal performance of vertical geothermal heat exchangers with reasonable accuracy.

The capabilities of the numerical modeling approaches presented here extend beyond the limitations of the analytical methods including simplified geometries and heat injection rates. Both models can simulate temperature induced thermal operations (i.e., injection of fluid at a desired temperature), a desired capability missing in the available analytical solutions. Moreover, analytical models do not provide information about the thermal interactions between the legs of circulation tubes embedded within geothermal heat exchangers. It is also not possible to estimate the transient borehole resistance, fluid temperatures or the temperatures within the GHE using analytical solutions. It is worthwhile to point out that both numerical models are easy to implement as the input values are parameterized and the geometry is automatically generated based on user input.

The FD model has relatively higher computational efficiency compared to the FE. Part of the higher computational expense for the FE analysis originates from the large number of elements required to model the domains. On the downside, the FD model cannot capture the full out-of-plane response, for which one must have a three-dimensional model. Moreover, the FE model can be utilized to model GHEs integrated with multiple circulation tubes, while the FD model can only work with a single loop.

5.2 Case Study 2: Numerical Evaluation of Ambient Interaction during Field TCTs Performed at Berkel Site

In-situ TCT was originally developed to assess the heat exchange capacity of geothermal boreholes. The conventional evaluation approach is based on the infinite line source theory (Ingersoll and Plass, 1948), which is applicable to boreholes which are long elements with small diameters. As heat exchanger piles are being installed more frequently, it is becoming necessary to develop suitable testing procedures and analysis methods to evaluate the thermal conductivity of such systems, which are geometrically different than boreholes. Heat exchanger piles typically have larger diameters and shorter lengths (diameter ~ 20-90 cm and length ~7-30 m). Inherent limitations prevent the direct application of existing testing and evaluation methods for heat exchanger piles. For example, American Society of Heating, Refrigeration, and Air Conditioning (ASHRAE) limits the TCT to a maximum borehole diameter of 15 cm (6 in), which makes it practically inapplicable for heat exchanger piles with much larger diameters. It is also recommended to disregard the first few hours (5 to 10) of test data when analyzing the results to allow stabilization of temperatures within the geothermal borehole (Kavanaugh et al., 2001). It is conceivable that a longer duration will be required for initial temperature stabilization when TCT is performed on larger diameter geothermal elements such as heat exchanger piles. Another important aspect related to the shorter length of heat exchanger piles is the more pronounced influence of ambient interactions. A smaller thermal mass is excited during the TCT of a heat exchanger pile and therefore any nominal effect from the ambient interaction through the above-ground connection pipes has a proportionally larger effect on the TCT measurements and interpretations.

In order to address these issues, five in-situ thermal conductivity tests were performed on three individual heat exchanger piles. Field test installation located in Richmond, Texas consists of three piles with 30 cm and 45 cm (12 in and 18 in) diameters extending to a depth of 20 m, each equipped with double HDPE circulation loops. Two of the piles were tested utilizing single and double loop configurations at respective tests and the other pile was tested only using the double loop (Brettmann et al., 2010). These tests provide the opportunity to gauge and benchmark the specific effects such as large diameter, short length and ambient interaction. The measurements from these tests are evaluated using 1D-3D hybrid finite element models presented herein. The

numerical models to analyze these field tests are developed using a commercially available finite element simulation environment, COMSOL Multiphysics™ (COMSOL, 2013). The models can simulate the 3D transient heat and mass transport processes in and around the heat exchanger pile with satisfactory accuracy and minimal computational effort (Ozudogru et al., 2014). A review of in-situ thermal conductivity testing is also presented, along with a methodology to estimate the effect of ambient air temperature variations on the TCT as previously presented by Bandos et al. (2011). Using this technique, a parametric study is performed to account for the effect of different percentages of heat exchange between the ambient air and the fluid in the connection pipes with respect to the total exchanged heat. Moreover, a numerical methodology that utilizes the ground temperature profile formulations by Kusuda and Achenbach (1965) is presented to calculate the undisturbed ground temperature before each test. The calibration of the models is carried out by comparing the numerical results with the measured data. For each test, a best fit parameter is selected by utilizing a statistical evaluation based on sum of squared residuals and average relative errors.

5.2.1 Berkel field test site and the testing program

The thermal conductivity test site is located at Berkel regional office in Richmond, Texas, within the Beaumont geologic formation along the Gulf Coast of Texas. The soils at the site consist of Pleistocene aged deposits of shallow coastal river channels and flood plains. There is a complex stratification of sands, silts and clays at the site and the vicinity as a result of the frequent changes in the courses of the river channels during this period. The clays at the site are overconsolidated due to desiccation that resulted when the water table dropped during the Second Wisconsin Ice Age. These clayey formations are predominantly composed of montmorillonite, kaolinite, and illite as well as fine ground quartz. The sandy formations are composed of quartz, feldspar and occasional hornblende. The sands vary in density from medium dense to very dense, and are sometimes lightly cemented (Brettmann et al., 2010).

Subsoil investigations performed at the test site reveal a stratified soil profile where a stiff to very stiff clay layer extends from the ground surface to a depth about 9.8 *m*. A layer of very dense sand is located below extending to a depth of 17.4 *m*. Beneath the sand layer, there is a 1.5 *m* thick layer of medium stiff clay which is underlain by a medium dense to very dense sand extending to the depth of boring completion at 21.3 *m*. The groundwater at the site is at a depth of 3.3 *m* below ground surface. There

is not any noticeable groundwater flow at the site. A sketch of the soil profile at the test site is shown in Figure 5.12.

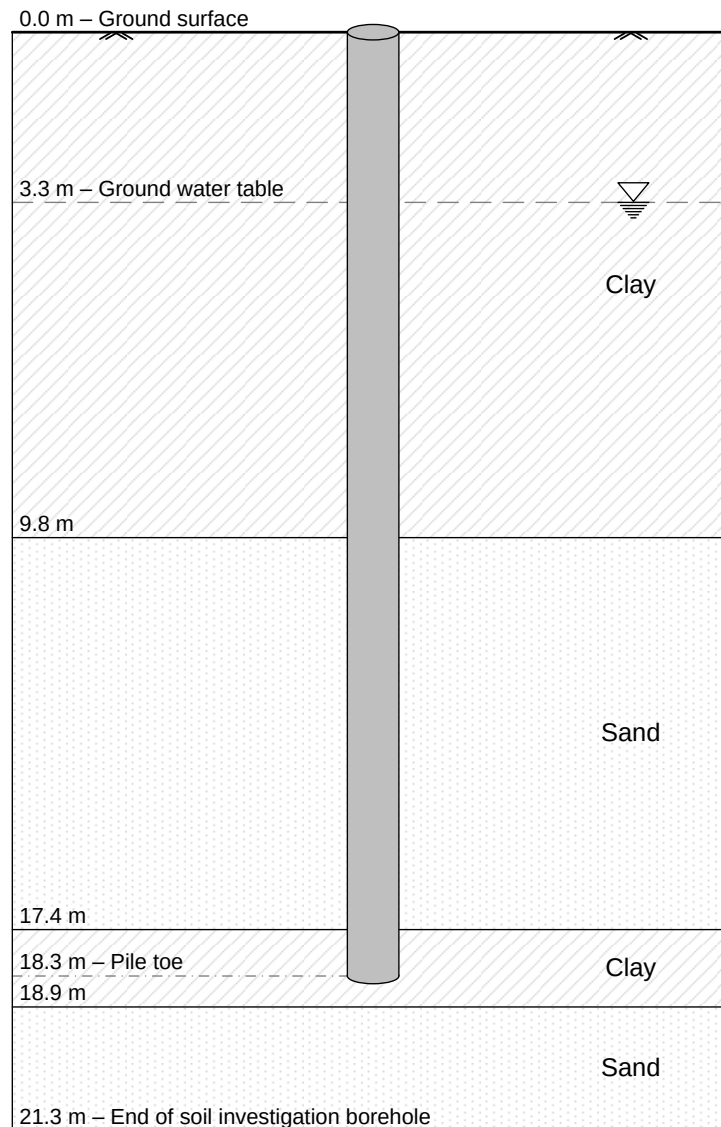


Figure 5.12 : Soil profile at the test site (after Brettmann et al., 2010).

A total of three heat exchanger piles were installed for the testing program. One of the test piles was installed using bentonite/silica sand thermal grout and the other two were installed using auger pressure grouted (APG) pile installation technique with cement/flyash/sand grout mix. APG piles are installed by rotating a hollow stem, continuous flight auger into the ground down to the desired pile base elevation. When the required depth is reached, a high strength, fluid grout is pumped under pressure through the hollow stem auger. The auger is withdrawn slowly by rotating clockwise as the pumping continues to both maintain the head of grout and avoid any intrusion of water or soil into the grout column. APG piles installed with geothermal pipe loops

are called auger pressure grouted energy (APGE) piles (Brettmann et al., 2010). The APGE piles at the test site were installed using a standard grout-mix design that would result in a compressive strength of 27.5 MN/m^2 in 28 days. A non-shrink additive was also included in the mix design and no grout shrinkage was observed in any of the piles.

Laboratory tests were performed on several soil samples and on both of the grout materials. Laboratory thermal conductivity testing was performed using the thermal needle test conforming to the guidelines of ASTM D5334 (2014). The tested soil sample in the upper clay layer was retrieved from depths below the groundwater table. Thermal conductivity at the upper levels has not been measured separately. The results of the laboratory tests are summarized in Table 5.3. The laboratory results show that the measured thermal conductivities of the thermal grout and the cement-mix grout used for the APGE piles are of the same value, even though the parent materials, densities and the moisture contents are much different.

Table 5.3 : Summary of the laboratory test results (after Brettman et al., 2010).

Sample	Depth (<i>m</i>)	Moisture content <i>w</i> (%)	Dry density ρ_{dry} (kg/m^3)	Thermal conductivity <i>k</i> ($\text{W/m}\cdot\text{K}$)
Clay	6.1	21.1	1730	2.22
Sand	13.7	14.0	1740	4.05
Clay	18.3	28.0	1540	2.09
APGE pile grout	—	7.3	1910	1.35
Thermal grout	—	64.5	930	1.35

The test piles were installed in a triangular layout with a 4.57 *m* center-to-center spacing between each pile as shown in Figure 5.13 (Brettmann et al., 2010). An exploratory soil boring was drilled at the centroid of the triangle, 2.64 *m* away from each test pile to monitor the ground temperatures during testing. The test piles are extending to a depth of 18.3 *m* and are equipped with two pairs of 25 *mm* diameter pipe loops with U-bend couplers at the bottom. The high density polyethylene (HDPE) pipe loops were attached to a 25 *mm* diameter full length center steel bar using 12.7 *cm* diameter plastic spacers. The center-to-center spacing of the pipes and the steel bar is about 7.6 *cm*, and the shank spacing of each loop is approximately 7.6 *cm*. The pipe loops were installed on diametrically opposite sides of the pile as shown in Figure 5.14. Three thermistors were placed at the center of each test pile at depths of 6.1 *m*, 13.7 *m* and 18.3 *m*.

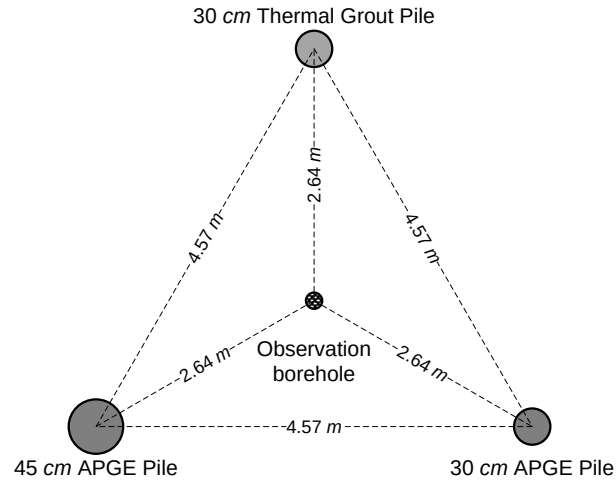


Figure 5.13 : Thermal conductivity test site layout (after Brettmann et al., 2010).

Several individual thermal conductivity tests were performed on the test piles with different loop configurations such that, utilizing single pipe loop or double pipe loop as reported by Brettmann et al. (2010). The double loop tests were performed in series connection. The pipe connections and the flow route of the carrier fluid in both single loop and double loop configurations are shown in Figure 5.14.

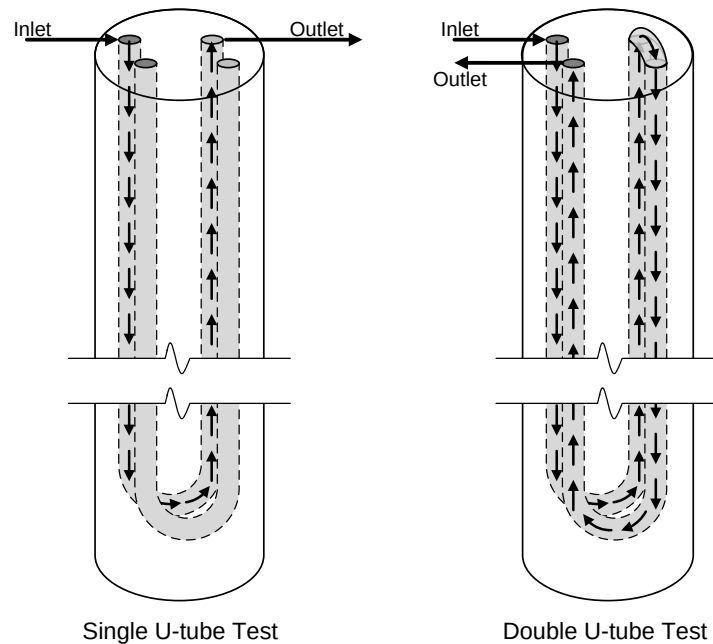


Figure 5.14 : Sketch of the connections in single and double loop tests.

The tests were performed for a duration of approximately 4 to 6 days. The heat injection rate, the inlet and the outlet temperature of the water, the temperatures inside the piles at different depths and the temperatures in the soil boring at different depths were measured at 5 minute intervals. The ambient air temperature was also monitored at 15 minute intervals during the tests. The details of the thermal conductivity test

program are summarized in Table 5.4. The measured temperatures of the average fluid, soil boring and the ambient air during the tests are shown in Figure 5.15.

Table 5.4 : Summary of the field thermal conductivity testing program.

Test no	Start date/time	End date/time	Duration	Test pile	Loop type
1	06/26/09 08:20	06/30/09 08:30	4 days	30 cm APGE	Single loop
2	06/30/09 09:30	07/06/09 08:20	5 days, 22 hours	45 cm APGE	Single loop
3	07/06/09 10:10	07/10/09 09:10	3 days, 23 hours	30 cm Thermal	Double loop
4	07/14/09 10:30	07/20/09 08:50	5 days, 22 hours	45 cm APGE	Double loop
5	07/20/09 09:20	07/24/09 09:20	4 days	30 cm APGE	Double loop

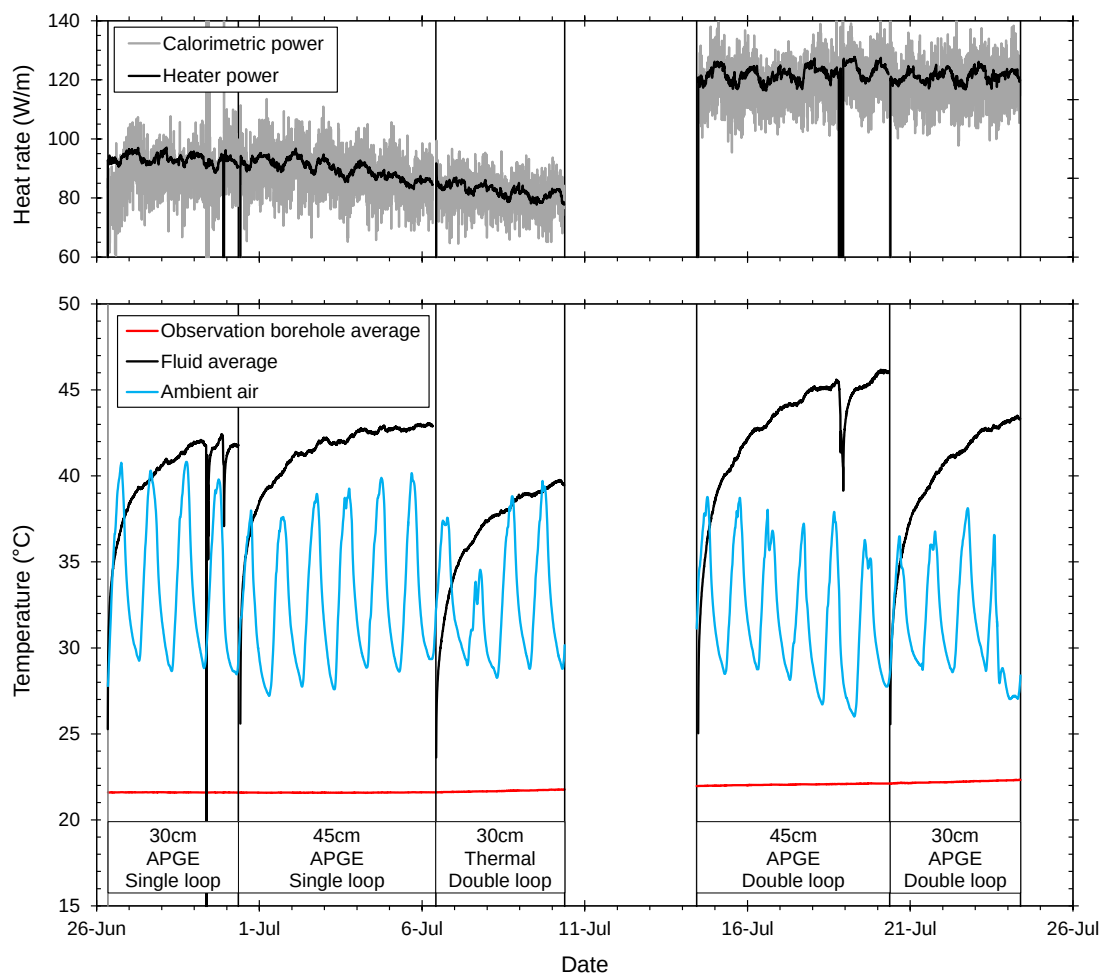


Figure 5.15 : Measured calorimetric and heater power, and average observation borehole, circulation fluid and ambient air temperatures during field thermal conductivity testing.

The average initial water temperature from the local source was 25°C. The average initial soil temperature measured from the thermistors was 22°C, which is consistent with reported mean ground temperatures and the seasonal climate data from this area (Turner and Doty, 2012). The change in the temperature of the soil boring was less than 1°C within the one month period of the whole thermal conductivity testing program. This minimal change could be triggered by the climatic temperature differences and hence, it was observed that there was no temperature change at the central observation borehole at a radial distance of 2.64 m from the test piles during the thermal conductivity testing.

5.2.2 Numerical analyses

A commercially available finite element simulation environment, COMSOL Multiphysics™ (COMSOL, 2013) was used to develop 1D-3D hybrid numerical models, in which the circulation pipes are modeled as line elements. The numerical model used in this study is described in detail by Ozudogru et al. (2014) in which the modeling methodology is discussed along with the validation of the modeling approach. The TCTs on the heat exchanger piles were analyzed using this modeling approach. These analyses collectively considered the different pile diameters, circulation loop configurations of the installed test piles and ambient interaction assumptions during testing. The numerical modeling approach also considered the initial ground temperature profile as affected by seasonal temperature variations

The numerical analyses were performed using a time dependent setting, where time steps were selected as 5 minute intervals from the start of the test until 30 minutes, and then 30 minute intervals until the end of the testing period. The last time step of each numerical analysis corresponds to the duration of each test, except for Test #1, in which the analysis was stopped at 70 hours. In this particular test, there were errors in the measurements due to a power failure after 70 hours.

All of the numerical models are discretized similarly since the soil profile is the same for all the piles. There is a difference in the geometrical setting for different pile diameters (30 cm vs. 45 cm) and different boundary conditions are used for single loop and double loop tests. Symmetry could not be utilized in any of the models; because the single loop tests are not practically symmetrical and the circulation pipes were connected in series in the double loop tests (Figure 5.14).

The overall model geometry consists of 120 volumetric domains, such that; 30 of them are soil, 26 are pile and 64 are pipe domains. Geometry discretization is studied carefully, allowing proper material assignment for each domain and efficient finite element mesh generation. The circulation pipes and the test pile are divided into domains along their length, where each domain corresponds to a different soil layer. The soil layers and the elevation of the ground water table are inferred from the soil borehole information (Figure 5.12). The physical representation of the pipes was modeled using line elements. However, there is a limitation of this approach in terms of the continuity and coupling of the temperature field along the pipe cross-section and the solid domain around it (i.e. grout). Since the pipes are made of line elements, the outer pipe wall temperature is coupled to the temperature field of the volumetric domains located along the pipe axis. This introduces estimation errors because it is necessary to couple the temperature at the pipe-grout boundary, which is located at a distance equal to the outside radius from the pipe axis. Another downside of this simplification is that, it does not account for the heat capacity of the pipes. Therefore, a ‘pseudo pipe’ approach was developed to overcome these issues (Ozudogru et al., 2014). The diameter of these cylindrical solid domains is set equal to the pipe outside diameter. The assigned material properties are selected to ensure an accurate temperature coupling, while accounting for the heat capacity of the pipes, as well.

The radius and the depth of the model limits are selected to be 5.0 *m* and 25.0 *m*, respectively. These dimensions are adequate, as it is known from the field measurements that there is no change in the undisturbed soil temperature 2.64 *m* from the heat exchanger piles during testing (Brettmann et al., 2010).

The finite element mesh of the model is generated in two steps; (1) triangular elements are used to mesh the horizontal plane at the depth of the pile toe, where the bottom of pseudo pipe elements resides; (2) the generated planar mesh is swept upwards and downwards until the upper and lower limits of the model.

The swept mesh is distributed along the vertical axis with applying refinements close to layer interfaces, generating coarser mesh elements in mid-layers. This optimization greatly reduces the number of mesh elements and is practically applicable since the heat flow is predominantly in the radial direction and the vertical heat flow is insignificant due to small thermal gradient in this direction. A similar meshing technique was used in a study by Marcotte and Pasquier (2008). The only exception to

this approach is the first 10 meters below the ground surface, where the vertical mesh was also refined to consider the vertical temperature gradient caused by climatic effects. The finite element mesh discretization of the models with 30 cm diameter piles consists of 63,112 elements and the models with 45 cm diameter pile are made of 64,064 elements. The finite element mesh and the soil domains are shown in Figure 5.16.

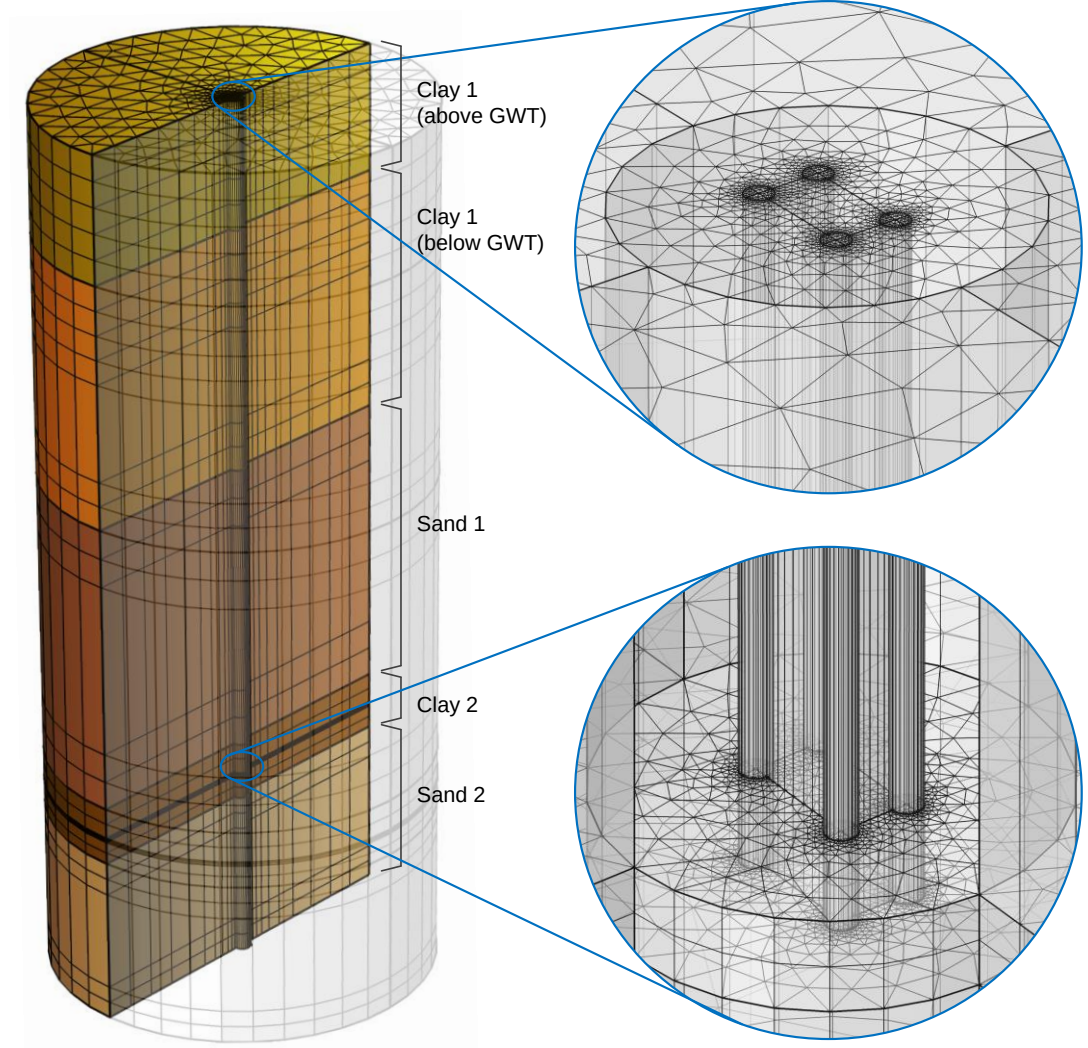


Figure 5.16 : Finite element mesh and the domains of the numerical model.

The material properties of soil and heat exchanger pile are assigned with the help of laboratory thermal conductivity tests conducted on the soil samples and the grout mixes. Specific heat capacities of the materials are calculated by local volume averaging using (Rees et al., 2000; Al-Khoury, 2011):

$$(\rho c_p)_{\text{eff}} = n(\rho c_p)_f + (1-n)(\rho c_p)_s \quad (5.11)$$

Two different material sets are used for modeling the pipes. Real thermal conductivity of the HDPE pipe material ($0.39 \text{ W/m}\cdot\text{K}$) is assigned to the line elements. These elements cannot have heat capacities since they are not volumetric elements. To overcome this issue, specific heat capacity of the pipes is assigned to the pseudo pipe elements surrounding these line elements. The effective density of the pseudo pipe elements is estimated using the pipe volume ratio as a correction factor:

$$\rho_{p \text{ eff}} = \rho_p \left(\frac{d_{po}^2 - d_{pi}^2}{d_{po}^2} \right) \quad (5.12)$$

Anisotropic thermal conductivity is assigned to the pseudo pipe elements and it is selected to be very high ($1000 \text{ W/m}\cdot\text{K}$) in the horizontal plane and zero in the vertical direction to correct for the temperature coupling error. Thermal properties of the materials assigned to corresponding domains and line elements in the numerical model are summarized in Table 5.5. Soil properties were obtained from the laboratory tests performed on saturated samples below the groundwater table (Table 5.3). Properties of the clay layer above the ground water table were estimated using relationships that account for the effect of moisture content (Farouki, 1981). Similarly the properties of the thermal grout used in the model were obtained from thermal needle tests performed on reconstituted samples.

On the other hand, initial simulations not reported here in which utilized thermal conductivity values from the laboratory tests performed on the APGE grout samples resulted in thermal response beyond which can be explained by ambient air interaction. According to ASTM D5334 (2014), the thermal needle probe procedure is suitable for soils and soft rocks. This method is not appropriate for determining the thermal conductivity of concrete after the curing period and it can result in unreliable results. One of the main shortcomings of this test especially in cementitious materials is the drilling and placement of the thermal needle in the sample in which a non-smooth contact between the sample and the needle. This can result in underestimation of the thermal conductivity. Allan (1997) reported that the thermal conductivity of cement/fly ash/sand grout mixes after drying is in the range of $1.6\text{-}2.4 \text{ W/(m}\cdot\text{K)}$. Hence, in the numerical models, the thermal conductivity of APGE pile grout was chosen as $2.0 \text{ W/(m}\cdot\text{K)}$ as an average value as opposed to the value measured in the laboratory test.

Table 5.5 : Summary of assigned material properties.

Material	Depth (m)	Density ρ (kg/m ³)	Thermal conductivity k (W/m·K)	Specific heat capacity c_p (J/kg·K)	Thermal diffusivity α (m ² /s)
Clay 1 (above GWT)	0.0–3.3	2006	1.70	1435	5.90e-7
Clay 1 (below GWT)	3.3–9.8	2095	2.22	1596	6.64e-7
Sand 1	9.8–17.4	2212	4.05	1303	1.40e-6
Clay 2	17.4–18.9	1971	2.09	1735	6.11e-7
Sand 2	18.9–25.0	2212	4.05	1303	1.40e-6
APGE pile grout	0.0–18.3	2049	2.00	1200	8.13e-7
Thermal pile grout	0.0–18.3	1530	1.35	2150	4.10e-7
HDPE pipe ¹	0.0–18.2	—	0.39	—	—
Pseudo pipe ²	0.0–18.2	312	1000	2300	1.39e-3

¹ HDPE pipe material properties are assigned to line elements.

² Pseudo pipe material properties are assigned to volumetric elements. Effective density is calculated using pipe volume ratio and anisotropic thermal conductivity is assigned as 1000 W/m·K in the horizontal plane and zero for the vertical direction.

Temperature dependent parameters are used for the thermal properties and flow characteristics of water. Basic four parameters of water are: (1) dynamic viscosity; (2) thermal conductivity; (3) density; and (4) specific heat capacity (Haynes, 2014). Other flow and heat transfer related parameters such as Reynolds number, Nusselt number and heat transfer coefficient are evaluated during runtime, using the water temperature and the temperature dependent functions defining these parameters.

The initial conditions of the model were configured prior to the analyses. The temperature profile of the model was initialized to establish the seasonally-compatible temperature profile of the near surface soils as well as the near constant temperature of the lower levels in the model. This temperature corresponds to the undisturbed ground temperature as it is in thermal equilibrium prior to the TCT. The common approach in these types of simulations is to use an undisturbed ground temperature profile that is constant with depth. This approach ignores the variation of ground temperature with depth along the upper value for the. However this is a simplification, which does not consider the effect of seasonal temperature variations of the ground temperature near the ground surface. This assumption can potentially introduce significant errors especially in relatively short GHEs such as heat exchanger piles that span a depth of soil with temperatures.

The initial ground temperature profile is estimated using the analytical method developed by Kusuda and Achenbach (1965) and further confirmed with the thermistors installed in the ground at the test site. This approach specifies the

undisturbed ground temperature as a function of time and depth by using the daily mean surface temperature data for the entire year and is expressed as:

$$T = T_{\text{mean}} - T_{\text{amp}} \cdot e^{-z \sqrt{\frac{\pi}{\alpha_{\text{avg}} t_{\text{cycle}}}}} \cdot \cos \left[\frac{2\pi}{t_{\text{cycle}}} (t_{\text{current}} - t_{\text{shift}}) - z \sqrt{\frac{\pi}{\alpha_{\text{avg}} t_{\text{cycle}}}} \right] \quad (5.13)$$

where, T is the temperature at the depth of interest, T_{mean} is the mean surface temperature (average air temperature), T_{amp} is the amplitude of the surface temperature (difference between maximum and minimum air temperature), z is the depth of interest, α_{avg} is the average thermal diffusivity of the ground, t_{cycle} is the period of soil temperature cycle (365 days), t_{current} is the current day of the year, and t_{shift} is the day of the year with minimum surface temperature. While this is an analytical expression, it considers the seasonal temperature equilibrium within the soil profile in response to ambient temperature variations and average thermal properties of the ground.

Undisturbed ground temperature profile of the site is estimated for each day of the year using the temperature data from the weather station located in Richmond, TX. Figure 5.17 shows the average daily ambient temperatures for the whole year. From the ambient temperature data, it can be inferred that the amplitude of the surface temperature is 8.86°C and the day of the year with the minimum surface temperature is the 15th day of the year.

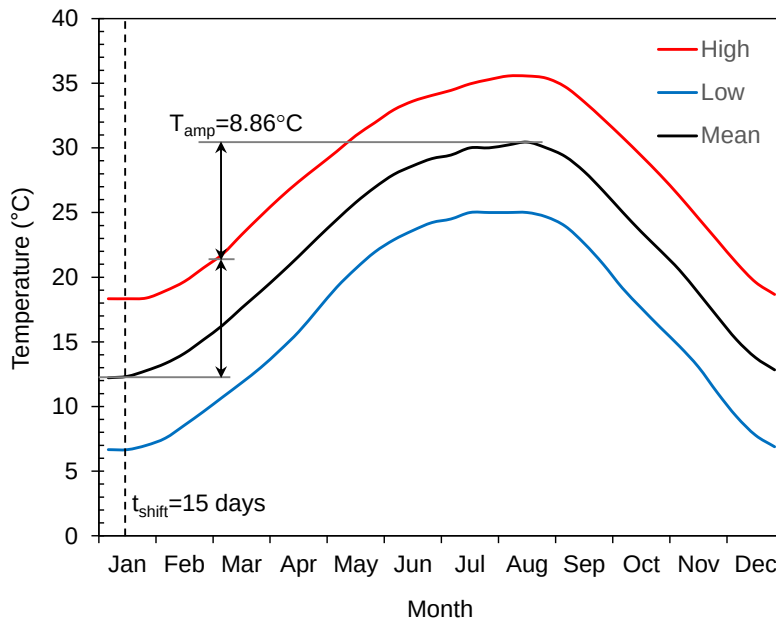


Figure 5.17 : The amplitude of the surface temperature and average daily ambient temperatures of Richmond, TX for the whole year.

Average thermal diffusivity of the ground is required to calculate the undisturbed ground temperature profile using Equation (5.13). Thermal diffusivity is evaluated by a weighted average approach with respect to the layer thicknesses by:

$$\alpha_{\text{avg}} = \frac{\sum_{i=1}^N \alpha_i H_i}{\sum_{i=1}^N H_i} \quad (5.14)$$

Substituting the material properties and the layer thicknesses given in Table 5.5 into Equation (5.14), the average thermal diffusivity is estimated as:

$$\alpha_{\text{avg}} = \frac{(3.3 \times 5.90 + 6.5 \times 6.64 + 7.6 \times 14.05 + 0.9 \times 6.11) \times 10^{-7}}{18.3} = 9.57 \times 10^{-7} \text{ m}^2 / \text{s} \quad (5.15)$$

This analytical method is utilized to calculate the undisturbed ground temperature profile of the test site for a given day of the year. Figure 5.18a shows the seasonal temperature profiles of the test site. For each test, the initial ground temperature profile with respect to depth are estimated for the day that test was started and these temperature profiles are used as initial conditions in the numerical models. Measured temperatures from the soil boring at depths of 6.1 m, 13.7 m and 18.3 m and estimated initial temperature profile for each test are presented in Figure 5.18b. The estimated initial temperatures are consistent with the field measurements.

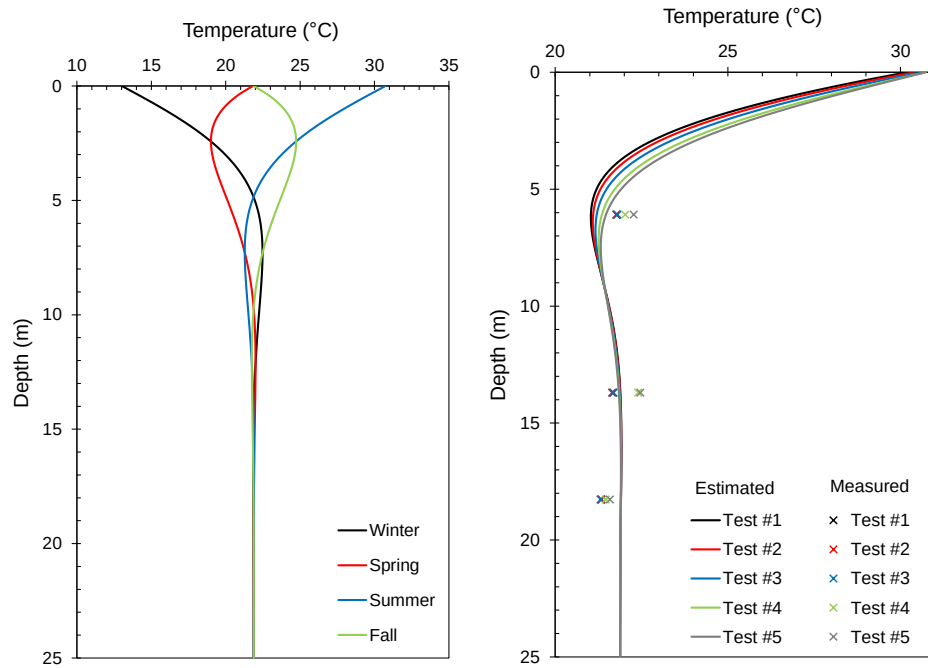


Figure 5.18 : (a) Undisturbed ground temperature profile of the test site for different seasons throughout a year; (b) Initial temperature profiles for each test used in the numerical analyses.

Boundary conditions designate the behavior of the numerical models during runtime and are used to build the sparse matrix solved thereafter for estimating the temperature changes. Two boundary conditions are defined on the inlet point: (1) volumetric flow rate; (2) prescribed inlet fluid temperature. The former is used to define the flow of water inside the pipes, and the latter is used to define the heating of the water. Heat injection is accomplished by applying an added temperature difference on the inlet boundary:

$$T_{in}(t) = T_{out}(t) + \Delta T(t) \quad (5.16)$$

Applied temperature difference, $\Delta T(t)$ maintains a temperature difference between the inlet and the outlet throughout the analysis, thus simulates heat injection (heating) in the system. $\Delta T(t)$ is a function of time, which is acquired from the field measurements of the inlet and the outlet temperatures. The measured inlet and outlet temperatures during testing are averaged over 30 minute periods and used as a time-dependent boundary condition to apply the heat injection using Equation (5.16) for each model.

For optimizing the convergence of the model, a continuous step function is used as a multiplier while assigning the volumetric flow rate and the applied temperature difference, allowing a smooth transition from zero to the desired value in 0.1 seconds. For the numerical models, average parameters corresponding to the flow characteristics and heat transfer of the water inside the pipes are presented in Table 5.6. The flow rate of the water is kept constant during each test, while other parameters are temperature dependent.

Table 5.6 : Average parameters of the pipe flow and heat transfer in pipes.

Test no	$\dot{V}_{avg}$ (l/min)	ΔT_{avg} (°C)	Re_{avg}	Nu_{avg}	$h_{int\ avg}$ (W/m ² ·K)
1	41.64	0.59	50600	267.67	6146.73
2	42.78	0.55	54679	278.83	6442.26
3	39.37	0.55	44035	247.73	5632.25
4	34.07	0.92	41690	224.57	5168.30
5	33.69	0.94	40866	224.24	5141.35

No heat flux (insulation) boundary condition is specified along the side and the bottom limits of the model. It was ensured that there is no temperature change along these boundaries throughout the runtime of the models. The boundary condition can be expressed as:

$$q'' = (-\mathbf{k}\nabla T) = 0 \quad (5.17)$$

A convective cooling boundary condition is used on the ground surface boundary with ambient air temperature measurements. Hence, the numerical model accounts for the effect of ambient air temperature changes on the ground during testing, at depths close to the ground surface. This boundary condition is defined as:

$$q'' = (-\mathbf{k}\nabla T) = h_{\text{air}} (T_{\text{air}} - T_{\text{ground}}) \quad (5.18)$$

The convective heat transfer coefficient of the air can be estimated using the correlation developed by Walton (1983). This equation is valid for natural convective flows between a fluid and a horizontal surface, such as the heat flow between the air and the ground surface. The correlation is given as:

$$h_{\text{air}} = 4.7382 |\Delta T|^{1/3} \quad (5.19)$$

where ΔT is the temperature difference between the ambient air and the ground surface.

Furthermore, the analyses incorporated ambient interaction methodology described above to account for the heat losses or gains imposed on the system through the piping system between the heating system (i.e. test truck) and the test pile.

5.2.3 Results and discussion

A series of numerical analyses were performed to investigate the effect of ambient interaction during TCT. The analyses considered different levels of ambient interaction in which the rates of heat exchange between the ambient air and the circulating water, ranged from 0.0% to 7.5% with 2.5% increments in reference to the parameter p described in Equation (4.38) above. A total of 20 analyses were performed with 4 ambient interaction assumptions for each of the 5 field TCT conducted on 3 heat exchanger piles. The analysis results for each test are presented in Figure B.1 through Figure B.5. These figures show the average measured and computed temperatures and heat rates of all analyses for each test.

The computed temperature data are analyzed by comparing the results with the measured data. This comparison is carried out by calculating the sum of squared residuals (SSR) and the average relative error. SSR is a measure of the discrepancy

between the measured data and the estimation model. A small SSR indicates a tight fit of the estimated data to the measured data. It can be calculated using:

$$SSR = \sum_{i=1}^n (Y_i - \hat{Y}_i)^2 \quad (5.20)$$

The average relative error is calculated as follows:

$$\delta_{\text{avg}} = \frac{\sum_{i=1}^n \left(\frac{|Y_i - \hat{Y}_i|}{|Y_i|} \right)}{N} \quad (5.21)$$

In Equation (5.20) and Equation (5.21), each data point corresponds to a temperature value in the time series. Thus, i^{th} value indicates the temperature at i^{th} time step, $t=t_i$. The results of the statistical analysis are presented in Table 5.7 and the best predicting models are shown in bold typeface.

Table 5.7 : Statistical evaluation of water and pile temperatures for each numerical model.

Test no	Effect of ambient air, p (%)	Water temperatures		Pile temperatures	
		Sum of squared residuals	Average relative error (%)	Sum of squared residuals	Average relative error (%)
1	0.0%	4.30	4.94%	0.53	2.09%
	2.5%	2.66	3.97%	0.24	1.44%
	5.0%	1.52	3.00%	0.10	0.82%
	7.5%	0.90	2.09%	0.11	0.93%
2	0.0%	9.39	7.26%	2.71	5.10%
	2.5%	6.37	6.01%	1.88	4.27%
	5.0%	3.98	4.77%	1.22	3.43%
	7.5%	2.23	3.52%	0.74	2.60%
3	0.0%	0.27	1.23%	0.77	1.93%
	2.5%	0.64	1.93%	0.93	2.33%
	5.0%	1.38	2.84%	1.32	2.95%
	7.5%	2.49	3.89%	1.93	3.67%
4	0.0%	0.79	1.65%	0.64	1.60%
	2.5%	1.30	2.41%	1.04	2.31%
	5.0%	2.35	3.36%	1.78	3.12%
	7.5%	3.92	4.36%	2.88	3.98%
5	0.0%	0.19	1.02%	0.22	1.25%
	2.5%	0.04	0.40%	0.06	0.52%
	5.0%	0.34	1.19%	0.18	0.94%
	7.5%	1.09	2.17%	0.57	1.73%

The best fit models have effect of ambient air correction of $p=7.5\%$ for Test 1 and Test 2; $p=0.0\%$ for Test 3 and Test 4; and $p=2.5\%$ for Test 5. These results clearly indicate that the heat exchange between the ambient air and the circulating water is more prominent in the tests where single loop was utilized (Tests 1 and 2). The lower the volume of the water circulating in the system, the higher this effect will manifest itself relatively. Another important aspect is related to the magnitude of the heat rate applied during testing. Higher heating was applied in the double loop tests and therefore any interaction between the ambient air and the circulating fluid during these tests is expected to be proportionally lower.

The statistical results are consistent with both water and pile temperatures, except for Test 1. For that test, the model with $p=5.0\%$ has better agreement with respect to pile temperatures. On the other hand, the statistical parameters are very close for both criteria and since the water temperature data is more sensitive, the model with $p=7.5\%$ is selected as the best fit. The maximum average relative error is found to be 3.52% in Test 2; and Test 5 gave the minimum average relative error, which is 0.40%. Temperature and heat rate plots of best-fit analysis for each test are shown in Figure B.6 through Figure B.10.

The figures and the statistical analysis suggest that Tests 1 and 2, both single loop tests, have poorer agreement with the numerical models compared to the other tests even at $p=7.5\%$. This discrepancy in the results could be due to power fluctuations during testing. It is also conceivable that single loop tests are more sensitive to measurement errors. Apart from that, the numerical models are successfully calibrated and they are able to predict the water temperatures and the temperatures in the piles accurately. Overall, this study also serves as a verification of the approach proposed by Bandos et al. (2011).

Temperature profile along the depth of the borehole for various times are shown for the best fit analysis of each test in Figure B.11 through Figure B.15. Similarly, temperature vs. radial distance at various times and temperature vs. time at several radial distances from the borehole wall are plotted for the horizontal cross-sections at the mid-depth of saturated clay and sand layers showing the best-fit analysis of each test. These plots are presented in Figure B.16 through Figure B.25. Finally, Figure B.26 through Figure B.35 show the temperature contour plots of best fit analysis of each test, for the same cross-sections and times.

6. THERMO-MECHANICAL PROCESSES IN HEAT EXCHANGER PILES

Relatively constant temperature of the ground and its thermal storage capacity can be exploited for heating and cooling of buildings. Traditionally, geothermal boreholes have utilized this concept for space heating and cooling. Recently, this ground coupling concept has been expanded from geothermal borehole systems to the use of building foundation elements as heat exchangers.

Ground temperature remains constant after a depth of about 6 meters as the near surface soils act as a thermal insulator as the ambient temperatures vary seasonally. The value of the ground temperature depends on the regional climate and reflects the mean annual temperature at that specific location (Kusuda and Achenbach, 1965). In most regions this constant temperature is about 10-24°Celsius (50-75°Fahrenheit).

Heat exchanger piles are designed to access and exploit the relative constant temperature of the ground and its energy storage capacity for efficient heating and cooling of buildings. Heat exchanger piles were first developed in Austria in the 1980s, but have only recently begun seeing common usage worldwide (Brandl, 2006). In this hybrid system, geothermal circulation loops are integrated into the deep foundation elements, such as piles, piers, or drilled shafts that are already designed to provide structural support. Fluid is circulated through the tubes in the heat exchanger piles. The heat energy is fed into the ground for cooling in the summer and withdrawn from the ground for heating in the winter. The temperature differential between the ground and the outside temperature acts as an energy pathway to harvest stored ground energy for bringing the building temperature to comfort zone levels.

Over the years, a number of analytical and numerical models of varying complexities have been developed and used as a design tool for vertical heat exchangers. These models were used to predict temperatures in and around the heat exchangers utilizing the heat transfer mechanisms inside a borehole, the conductive heat transfer from the borehole and the thermal interferences between boreholes. A number of design tools based on finite element or finite volume programs were used to develop fully discretized borehole heat exchanger models to include transient effects, as well as the

correct borehole geometry. Some of the most noteworthy models include the studies by Al-Khoury et al. (2005), Al-Khoury and Bonnier (2006), Signorelli et al. (2007), Marcotte and Pasquier (2008), Lamarche et al. (2010), He (2012) and Ozudogru et al. (2014).

In a heat exchanger pile, the temperature gradient between the ground and the pile can be utilized for heat transfer between the ground and the deep foundation. Thermal energy is extracted from the ground in heating mode, and is injected into the ground in cooling mode with the use of the heat carrier fluid circulating in the tubing system embedded in the pile (Brandl, 2006; Adam and Markiewicz, 2009). Heat transfer around a heat exchanger pile can be modeled in the same way as thermal diffusion around a geothermal borehole (Arson et al., 2013; Ozudogru et al., 2014), however thermo-mechanical couplings need to be accounted for to predict the interactions between a heat exchanger pile and the surrounding soil. So far, thermo-mechanical couplings were modeled separately for the pile and for the surrounding soil. Temperature changes induced by heat exchange operations can result in a variety of thermo-hydro-mechanical processes that can influence the serviceability and limit state of heat exchanger piles. Processes such as pore pressure development and thermal consolidation in soft saturated clays as well as thermally induced moisture migration followed by increased effective stress in unsaturated soils may play a role in pile behavior in response to increased temperatures.

This chapter discusses the thermo-mechanical processes in heat exchanger piles induced by temperature changes. Amongst the investigated phenomena are the possible effects on pile axial stresses, contact pressures, and the behavior of the pile-soil interface, i.e., slip displacements, friction stresses, and change in bearing capacity of the pile. Parametric analyses with respect to soil strength and pile end constraints are performed in order to assess the effect of confinement on the results.

6.1 Radial Expansion of Heat Exchanger Piles and Possible Effects on Contact Pressures at Pile-Soil Interface

The operation of heat exchanger piles creates differential temperatures between the pile and the soil around the pile. The thermo-elastic radial expansion of the heat exchanger pile and consequent increased contact pressures at the pile-soil interface is thought of as one of the possible mechanisms underlying changes in shaft resistance

measured in several thermo-mechanical pile load tests. The main purpose of this study is to investigate if increased pile capacity at higher temperatures can be attributed to thermal expansion (Olgun et al., 2014b). A summary of thermo-hydro-mechanical soil-structure interaction processes is provided in the context of temperature effects on heat exchanger pile response. In addition, a review of experimental studies on the load-displacement behavior of heat exchanger piles at different temperatures is reported (Section 6.1.1). Finite Element analyses are presented (Section 6.1.2) to study the radial deformations and the magnitude of the stresses that develop around a pile subjected to a temperature increase. The results of the analyses provide insight on the magnitude of temperature-induced contact stresses on heat exchanger piles and the expected effect on shaft friction for a range of soil strength and stiffness values.

6.1.1 Thermo-hydro-mechanical soil-structure interaction in heat exchanger piles

Performance of heat exchanger piles can be affected by the temperature changes induced by heat exchange operations. Anisotropic thermal fields within the pile and the surrounding soil produce relative deformations between the soil and the pile, depending on the fixity conditions of the pile (Bourne-Webb et al., 2009). These deformations can induce slip at the pile-soil interface, which can modify the shear stress transfer between the pile and the soil. This can change the internal stresses along the pile in addition to those caused by the structural load. Seasonal elongation and shortening of the heat exchanger pile results in repeated cyclic shear straining, which can potentially affect the properties of the soil-pile interface, hinder the pile's frictional resistance, reduce its axial load capacity, and may cause unanticipated settlements or eventually result in failure. These types mechanisms were already noted in studies focusing on offshore pile behavior under cyclic loads (Poulos, 1989; Jardine and Standing, 2000).

Furthermore, soil behavior is affected by thermal history related to the heating/cooling cycles imposed by the heat exchanger pile. For instance, temperature changes can result in excess pore pressures (Campanella and Mitchell, 1968; Uchaipichat and Khalili, 2009), volume changes (Baldi et al., 1988; Cekerevac, 2003; Abuel-Naga et al., 2005) and can also affect the yield pressure of the soil (Eriksson, 1989; Moritz, 1995; Leroueil and Marques, 1996). These processes are likely to result in deformations and stresses that are not typically considered in foundation design. The

long-term effects of repeated thermal consolidation cycles are still not well understood. There is a potential risk that fatigue-like processes can considerably degrade the strength and stiffness of clayey soils along the shaft interface. Furthermore, moisture migration away from the heat exchanger pile can cause local desaturation of the soil, and reduce thermal conductivity - similar to the thermal instability observed in buried cables (Brandon et al., 1989). Such phenomena would significantly hinder the thermal performance of heat exchanger piles.

A variety of experimental and analytical studies has been performed to investigate the thermo-mechanical behavior of heat exchanger piles, including temperature effects on soil behavior and shear stress mobilization/relaxation at the pile-soil interface due to thermo-elastic pile deformations (Brandl, 2006; Laloui et al., 2006; Bourne-Webb et al., 2009; Knellwolf et al., 2011). Researchers at the University of Colorado at Boulder have performed load tests on semi-floating heat exchanger piles in a centrifuge using partially saturated Bonny silt compacted around the pile. The test pile was heated to different temperatures before applying the structural load (Rosenberg, 2010). This study demonstrates the effect of temperature on the load-displacement behavior of heat exchanger piles. Figure 6.1 and Table 6.1 show that the pile has an increased head stiffness and capacity at higher temperatures. This experimental study presents evidence about the temperature-induced changes in pile capacity, presumably as a result of the altered stress state around the test pile (McCartney and Rosenberg, 2011).

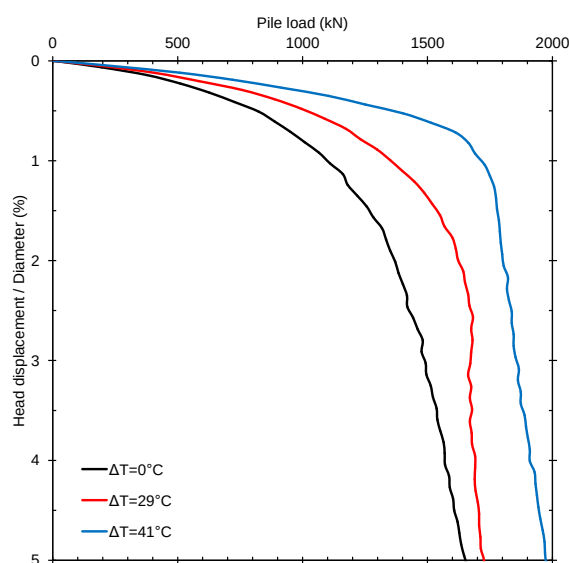


Figure 6.1 : Load-settlement curves in prototype scale for heat exchanger piles tested at different temperatures using reduced scale models in the centrifuge (adapted from McCartney and Rosenberg, 2011).

Table 6.1 : Pile load at various head displacement/diameter ratios for different loading cases.

Loading case	Head displacement/Diameter (%)	Pile load (kN)	Change in pile load ^a (%)
$\Delta T=0^{\circ}\text{C}$	1	1066	—
	2	1350	—
$\Delta T=29^{\circ}\text{C}$	1	1351	+27
	2	1622	+20
$\Delta T=41^{\circ}\text{C}$	1	1691	+59
	2	1799	+33

^a Change in pile load with respect to $\Delta T=0^{\circ}\text{C}$ loading case at the respective head displacement.

A summary of other experimental studies investigating temperature effects on the load-displacement behavior of heat exchanger piles is provided in Table 6.2. These studies utilized small-scale models tested under 1-g or at increased g-levels in the centrifuge to represent field-scale stresses. Wang et al. (2011; 2012) tested small-scale piles at different temperatures using loosely compacted dry and partially saturated N50 fine sand. They noted no change in shaft resistance with dry sand and a decrease in shaft resistance with the partially saturated sand at elevated temperatures. They also measured decreased shaft resistance at higher temperatures with piles tested in partially saturated 300WQ silica flour. Similarly, Kramer and Basu (2014) performed similar small-scale tests under 1-g using dry F50 Ottawa sand and observed a slight increase (~5%) in pile capacity at increased temperatures. Goode et al. (2014) performed experiments in the centrifuge with dry Nevada sand and measured no apparent change in pile response at different temperatures. As observed, the load-displacement measurements of the tested piles at increased temperatures in these studies show different trends. These tests represent different testing conditions, materials, and model preparation techniques, which are presumably reflected in the observed behavior through different mechanisms.

It is quite conceivable that the thermo-hydro-mechanical processes described above can play a role in affecting heat exchanger pile capacity at elevated temperatures. For instance, pore pressure development and thermal consolidation in soft saturated clays can alter the stress state and result in deformations around a heat exchanger pile. Similarly, thermally induced moisture migration followed by increased effective stress in unsaturated soils may play a role on pile behavior. Stewart and McCartney (2013) performed tests on end-bearing heat exchanger piles in the centrifuge using partially saturated compacted Bonny silt. They measured a reduction in moisture content near

the heated pile and attribute the increased pile resistance under similar conditions to increase in effective stresses and higher shaft resistance. It is also possible that lateral pressures induced by compaction around the test pile may have affected the overall behavior of the piles tested in this study as well as the tests reported in McCartney and Rosenberg (2011). However it is not clear as to how these stresses played a role in different shaft resistances at increased temperatures.

Table 6.2 : Summary of small-scale experimental studies investigating the temperature effects on load-settlement behavior of heat exchanger piles.

Study	Model	Soil type	Pile	ΔT (°C)	Remarks
McCartney and Rosenberg (2011)	Centrifuge (24g)	Bonny silt (compacted) w=13.2% fines=84% PI=4 $\phi'=32^\circ$	Concrete D=76.2mm (1.8m) H=381mm (9.1m)	29 / 41	40% increase in side shear resistance with heating
Wang et al. (2011)	Laboratory (1g)	N50 Fine sand (loosely compacted – 10 layers) $C_u=1.47$ $C_c=1.21$ w=0.5%	Steel tube D=25.4mm t=1.2mm	20	50% decrease in side shear resistance with heating
		300WQ Silica flour (loosely compacted – 10 layers) $C_u=4.8$ $C_c=2.13$ w=21.5%, 24%	<i>Pile surface is coated with a layer of N50 fine sand using epoxy resin</i>		10% to 50% decrease in side shear resistance with heating
Wang et al. (2012)	Laboratory (1g)	N50 Fine sand (loosely compacted – 10 layers) $C_u=1.47$ $C_c=1.21$ w=0%, 2%, 4%	Steel tube D=25.4mm t=1.2mm <i>Pile surface is coated with a layer of N50 fine sand using epoxy resin</i>	20 / 40	w=0% – No change in side shear resistance w=2%, 4% – Reduction in side shear resistance
Goode et al. (2014)	Centrifuge (24g)	Dry Nevada sand $D_r=60\%$ e=0.75 $D_{10}=0.09\text{mm}$ $D_{30}=0.11\text{mm}$ $D_{60}=0.16\text{mm}$ $\phi=35^\circ$ G=30MPa $\nu=0.3$	Concrete D=63.5mm (1.5m) H=342.9mm (8.2m)	7 / 12 / 18	No change in ultimate capacity with heating
Kramer and Basu (2014)	Laboratory (1g)	Dry F50 Ottawa sand (fine silica sand) (air pluviation) $e_{\max}=0.78$ $e_{\min}=0.48$ $D_{50}=0.28\text{mm}$ $C_u=1.8$ $G_s=2.65$	Concrete D=100mm H=1.22m	20	Slight increase in pile capacity (~5%) with heating Decrease in pile head stiffness with heating

The goal of this study is to investigate the magnitude of changes in contact pressure between the soil and the pile, as a result of the temperature-induced radial deformation of the heat exchanger pile. This phenomenon is postulated as one of the possible mechanisms of increased shaft resistance at higher temperatures (McCartney and Rosenberg, 2011; Stewart and McCartney, 2013; Goode et al., 2014). Temperature-induced expansion of the pile will result in increased lateral stresses, thereby affecting frictional resistance along the pile shaft. The mechanical behavior of the pile is expected to lie between free expansion under temperature increase (in the absence of soil, Figure 6.2a and constrained heating building up internal thermal stress (in the presence of a fixed support, Figure 6.2b. These represent lower and upper bounds for the expected radial stress development in actual field conditions: deformable soil partially restrains the free expansion of the pile, without fully preventing it. In the absence of development of soil plasticity, the problem of temperature-induced confining pressures may be viewed as a hyperstatic thermo-elasticity problem illustrated by the two springs in Figure 6.2c.

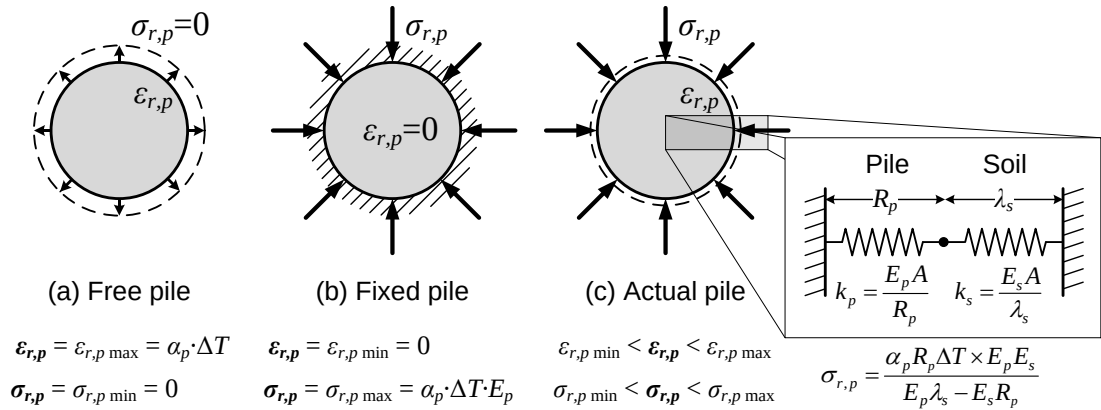


Figure 6.2 : Sketch showing the mechanical implications of heating a pile in: free, fixed and real conditions (embedded in elastic soil).

6.1.2 2D numerical analysis of soil contact pressure induced by pile heating

A Finite Element model has been developed with COMSOL Multiphysics™ software platform to study the influence of radial thermal expansion on the increase of contact pressure applied by the soil on the pile (COMSOL, 2013). The mesh and its dimensions are shown in Figure 6.3. The pile is modeled as a thermo-elastic medium, while the soil is assumed to follow a Tresca linearly elastic – perfectly plastic behavior. The soil and pile elements share the same nodes at the pile-soil interface. The initial temperature of the soil and the pile is 15°C and the external boundary of the model is

fixed. Material parameters used in the analyses are listed in Table 6.3. Undrained soil stiffness is assumed to be proportional to undrained shear strength as $E=1000 \times s_u$; this is a median value for the stiffness/strength ratio (Jardine et al., 1984). Undrained conditions were considered in the total stress analyses while the pore water pressure development and water flow was not considered.

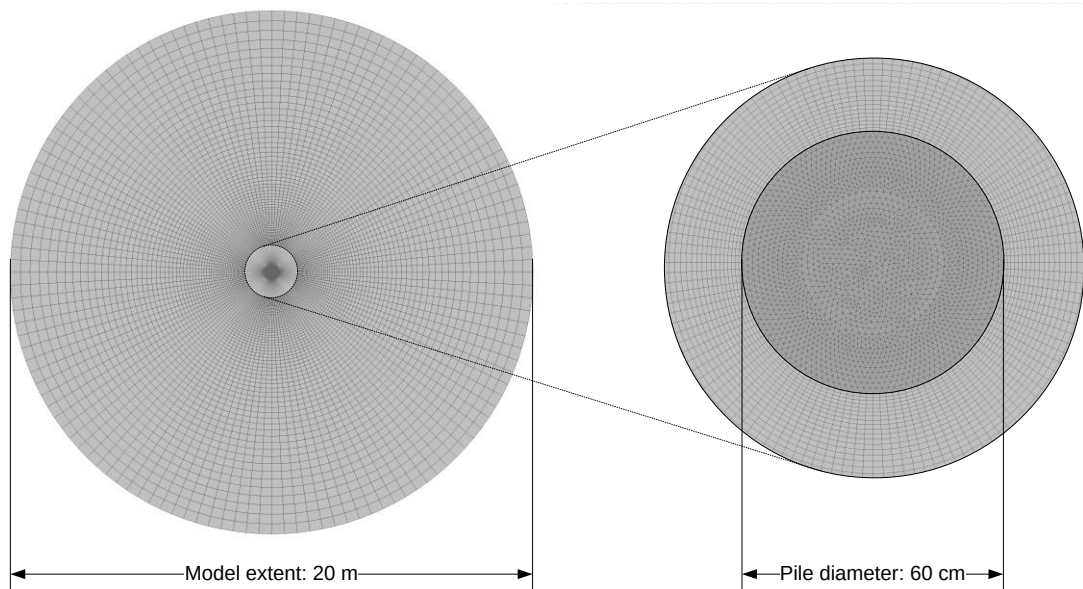


Figure 6.3 : Mesh adopted in the finite element analyses with a total of 17,338 elements.

Table 6.3 : Material parameters used in the finite element simulations.

Parameter / Material	Reinforced concrete	Soil
Mass density (kg/m^3)	2500	1500
Young's modulus (MPa)	30000 ^a	$1000 \times s_u$ ^b
Poisson's ratio	0.150	0.495
Thermal conductivity ($\text{W/m}\cdot\text{K}$)	1.5	2.0 ^c
Specific heat capacity ($\text{J/kg}\cdot\text{K}$)	1200	1500 ^d
Coefficient of thermal expansion ($\mu\epsilon/\text{K}$)	10 ^e	50 ^f
Undrained shear strength, s_u (kPa)	—	25 / 50 / 100 ^g

^a In several studies, Young's modulus of concrete has been reported as 29 GPa (Kramer and Basu, 2014), 29.2 GPa (Knellwolf et al., 2011) and 40 GPa (Bourne-Webb et al., 2009).

^b Ranjan and Rao (2000) expressed the correlation of undrained Young's modulus to undrained shear strength as $E=(750-1200) \times s_u$ for normally consolidated clays ($1500-2000 \times s_u$) for OC clay similar to the ranges given in Jardine et al. (1984).

^c Thermal conductivity of clays are in the range of 0.15-2.5 $\text{W}/(\text{m}\cdot\text{K})$ with saturated clays at the higher end of this range (Al-Khoury, 2012).

^d Specific heat capacity of clays are in the range of 920-2200 $\text{J}/(\text{kg}\cdot\text{K})$ (Al-Khoury, 2012, p.15)

^e Thermal expansion coefficient of reinforced concrete has been reported as 8.5 $\mu\epsilon/\text{K}$ by Bourne-Webb et al. (2009) and 10 $\mu\epsilon/\text{K}$ by Knellwolf et al. (2011) and Salciarini et al. (2013).

^f Thermal expansion coefficient of clays has been reported as 30 $\mu\epsilon/\text{K}$ (Salciarini et al., 2013) and 50 $\mu\epsilon/\text{K}$ (Mitchell and Soga, 2005).

^g Typical values of undrained shear strength selected to present detailed results for soft (12.5-25 kPa), medium (25-50 kPa) and stiff clay (50-100 kPa), respectively (Lambe and Whitman 1969, Terzaghi et al., 1996).

Stationary analyses were performed to establish the magnitude of soil resistance in response to thermal pile expansion. In these analyses, the soil temperature is maintained at its initial value (15°C), and the temperature of the pile is raised to 25°C. Even though these analyses do not represent field conditions during heat exchange operations where the soil temperature also increases, they provide baseline values for increased stresses that develop as a result of thermal pile expansion only. Young's modulus (E) of the medium around the pile is systematically varied between 10^2 kPa and 10^{12} kPa, to represent different levels of lateral restraint at the pile face. In this case, stiffness of the hypothetical medium around the pile greatly exceeds the range of typical values for soils, in order to represent full range of cases from free pile face to full fixity. Figure 6.4 shows that for $E < 10^6$ kPa, the total radial strain of the pile is equal to the thermal strain, which means that the mechanical strain induced by the reaction of the soil is negligible: the pile face deforms freely as a result of thermal expansion. As a matter of fact, the deformation of the soil is equal and opposite in sign to that of the pile $\epsilon_r = 10^{-4}$. For $E > 10^9$ kPa, the presence of the soil prevents the pile from expanding: radial strains in the soil and in the pile are close to zero. For $10^6 \text{ kPa} < E < 10^9 \text{ kPa}$, the soil acts as a spring, and applies a restraint to the pile in reaction to its thermal expansion. For realistic values of soil modulus (between 5×10^3 kPa and 10^5 kPa), a pile subjected to heating embedded in isothermal soil is expected to expand freely, as evidenced by total pile strains being near-equal to thermal strains and thus mechanical pile strains being very close to zero. This shows that, the soil provides almost no constraint to thermal pile expansion: the pile's response is close to that of the free pile shown in Figure 6.2a. A cavity expansion problem was also simulated for verification purposes, by applying the radial strains of the pile face on the cavity wall. The results, displayed in Figure 6.5, show that for $E > 10^9$ kPa, thermo-mechanical stresses around 3.5 MPa build up at the pile-soil interface, equivalent to those that would develop upon a pressurization of a fixed cylindrical cavity.

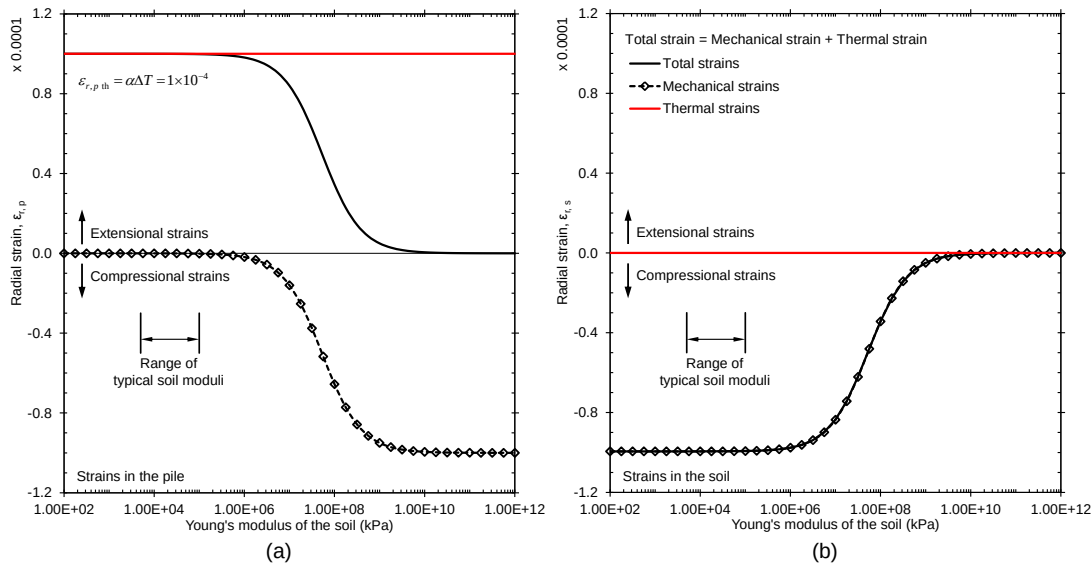


Figure 6.4 : Radial deformation of the pile and of the soil close to the interface (stationary plane stress analysis), assuming a wide range of elastic moduli for the host medium (realistic soil undrained Young's moduli are between 5×10^3 kPa and 10^5 kPa). Note: in all cases, the thermo-elastic strain is 0.0001, which corresponds to a purely thermo-elastic response in the absence of any soil restraint. Total strains and elastic strains at the soil side are identical because the thermo-elastic strains in the soil are equal to zero for the stationary analysis with $\Delta T = 0$ in the soil.

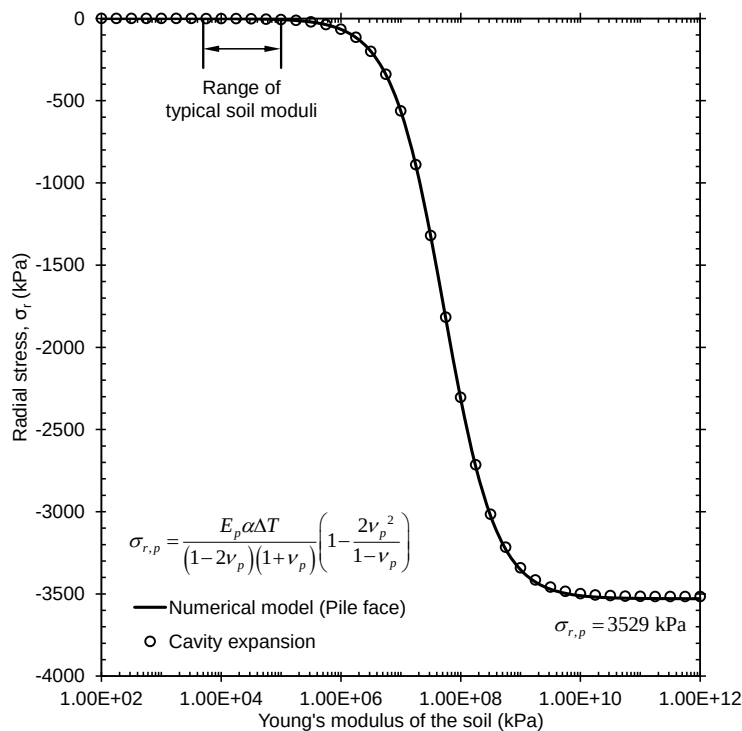


Figure 6.5 : Radial stress mobilized at the pile/soil interface in the stationary plane stress analysis, for a wide range of soil elastic moduli (realistic soil undrained Young's moduli are between 5×10^3 kPa and 10^5 kPa). Stresses computed numerically upon pile heating (solid lines) match the stresses computed using the cavity expansion theory (circles). Pile face displacements from the numerical analyses applied on the cavity wall. Radial stress acting on the pile at full fixity is 3529kPa.

Transient analyses were performed to check whether the thermal expansion of the soil has a considerable influence on the increase of contact stress at the pile face. Compared to the stationary analyses, transient analyses provide a more realistic representation of field conditions where the heated soil also expands during heat exchange operations and induces additional radial stresses on the pile: soil mechanical behavior is coupled with thermo-elasticity and heat transfer is governed by the diffusion equation. Undrained strength values of $s_u = 25, 50$ and 100 kPa were selected to represent soft, medium and stiff clay while the undrained soil stiffness is assumed to be proportional to undrained shear strength as $E=1000 \times s_u$. The pile was subjected to a linear temperature increase during 12 hours, from an initial temperature of 15°C up to 25°C (typical of a heat exchanger pile operating in building cooling mode). This temperature was maintained for 12 hours. Computed temperature variations within the soil are shown in Figure 6.6.

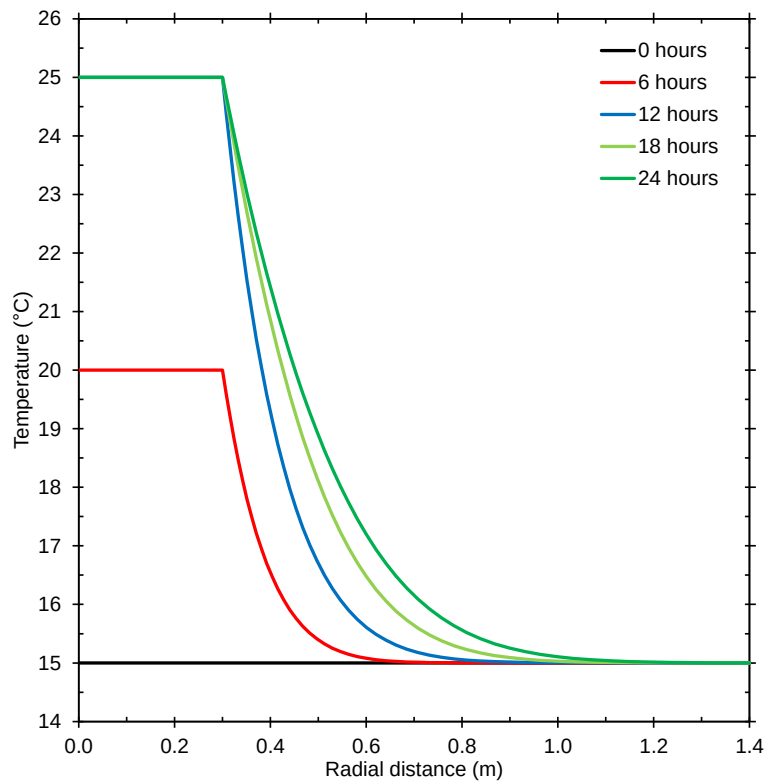


Figure 6.6 : Radial distribution of temperatures at different times of the transient analysis.

The 2D cross-sectional analyses considered both plane strain and plane stress conditions. In the plane stress analysis, the cross-section is free to deform in the out-of-plane direction and in the plane-strain analysis out-of-plane deformation is fully fixed. Total strains in the pile and the soil in response to heating are shown in Figure

6.7 for plane stress and plane strain conditions. Stationary and transient analyses results are provided together for comparison. Strain plots are the same for all the soils analysed, which confirms the thermo-elastic response of the pile and the soil. The radial strain of the pile is equal to the strain of a pile in free thermal expansion, which confirms the negligible resistance from the soil similar to the results presented in Figure 6.4. In plane stress (Figure 6.7a), the radial deformation of the pile is purely thermal. In plane strain (Figure 6.7b), the radial deformation of the pile occurs as a result of thermal expansion and Poisson's effects due to the constraint along the out-of-plane direction.

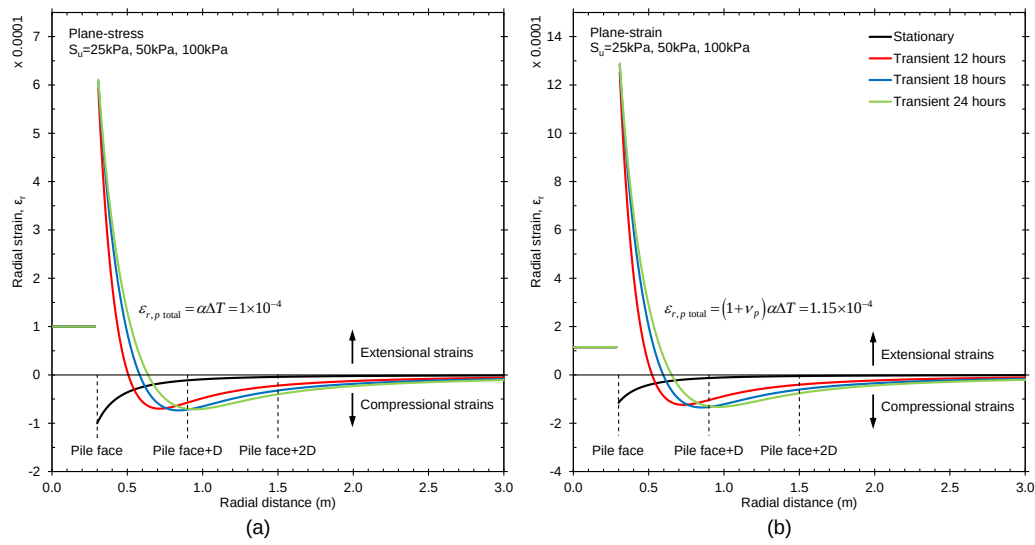


Figure 6.7 : Strain vs. radial distance for the stationary and the transient models: (a) Plane-strain model, $s_u = 25, 50, 100\text{kPa}$; (b) Plane-stress model, $25, 50, 100\text{kPa}$.

Note: the same results were obtained for the three soil strengths studied.

In the stationary analyses, radial strains in the soil, which occur as a result pile expansion, are purely compressional. These strains are negligible beyond 1.2 m away from the pile face. In the transient case, the radial strains in the soil occur as a result of the thermal expansion of both the pile and the soil. The soil surrounding the pile exhibits extensional radial strains within a distance equal to no more than half the pile diameter. Presumably, the compression induced by the expanding pile is overcome by the thermal expansion of the soil, which has five times the thermal expansion coefficient of the pile. Thermal expansion of the heated soil zone is also imposing compressional radial strains from about a half pile diameter to about 2.7 m away from the pile face, a distance about four and a half times the pile diameter. The magnitudes of these compressional strains are larger and expand beyond those induced only by thermal expansion of the pile as seen from stationary analysis.

Computed radial stresses within the pile and the soil are shown in Figure 6.8 for all the considered cases. Radial stresses in the soil are compressive both for stationary and transient cases for soils with different strength values. In the stationary case, the maximum stress increase occurs at the pile-soil contact and reduces with distance to negligible values at about four pile diameters away from the pile face. In the transient case, compressional stresses reach a maximum magnitude at 10-15 cm from the pile face. This indicates that compressional stresses maximize as a combination of thermal expansions of the pile and heated soil zone. The progressive increase of temperature and consequent thermal expansion of the soil results in larger compressional stresses. Furthermore, the radial stress increase in plane strain (Figure 6.8b, d, f) is higher than the stress increase in plane stress (Figure 6.8a, c, e). The compressional stresses in plane strain are increased as a result of the Poisson effects and constrained deformations along the out-of-plane direction.

For the transient case, it may seem rather contradictory that compressional stresses develop around a zone where total strains are extensional within the soil. We interpret that compressional stresses develop as a result of the blocked strains similar to compressional stresses that would develop within a heated solid bar with imposed fixities. In other words, the observed extensional strains are smaller compared to what they would have been, had they not been constrained due to the kinematics of the pile-soil interaction.

Figure 6.9 presents the temperatures induced radial stresses on the pile face for a range of soil moduli. In plane stress, the pile can expand freely in the out-of-plane direction (longitudinal pile deformations), therefore the additional degree of fixity induced by soil thermal expansion does not increase the radial stress in the pile (Figure 6.9, solid lines). In plane strain, the resulting contact stresses that develop at the pile face are higher than those in plane stress due to the Poisson effects, as a result of the constrained deformations along the out-of-plane direction. Furthermore, the increase in contact pressure is higher in the transient analysis due to the thermal expansion of the soil (Figure 6.9, dashed lines). For typical soil modulus values, the additional lateral stresses that develop at the soil-pile contact do not exceed 15 kPa at the end of 12 hours of heating. These results indicate that the heating-induced radial stresses applied by the soil on the pile remain at values that would not have a significant effect on shaft resistance.

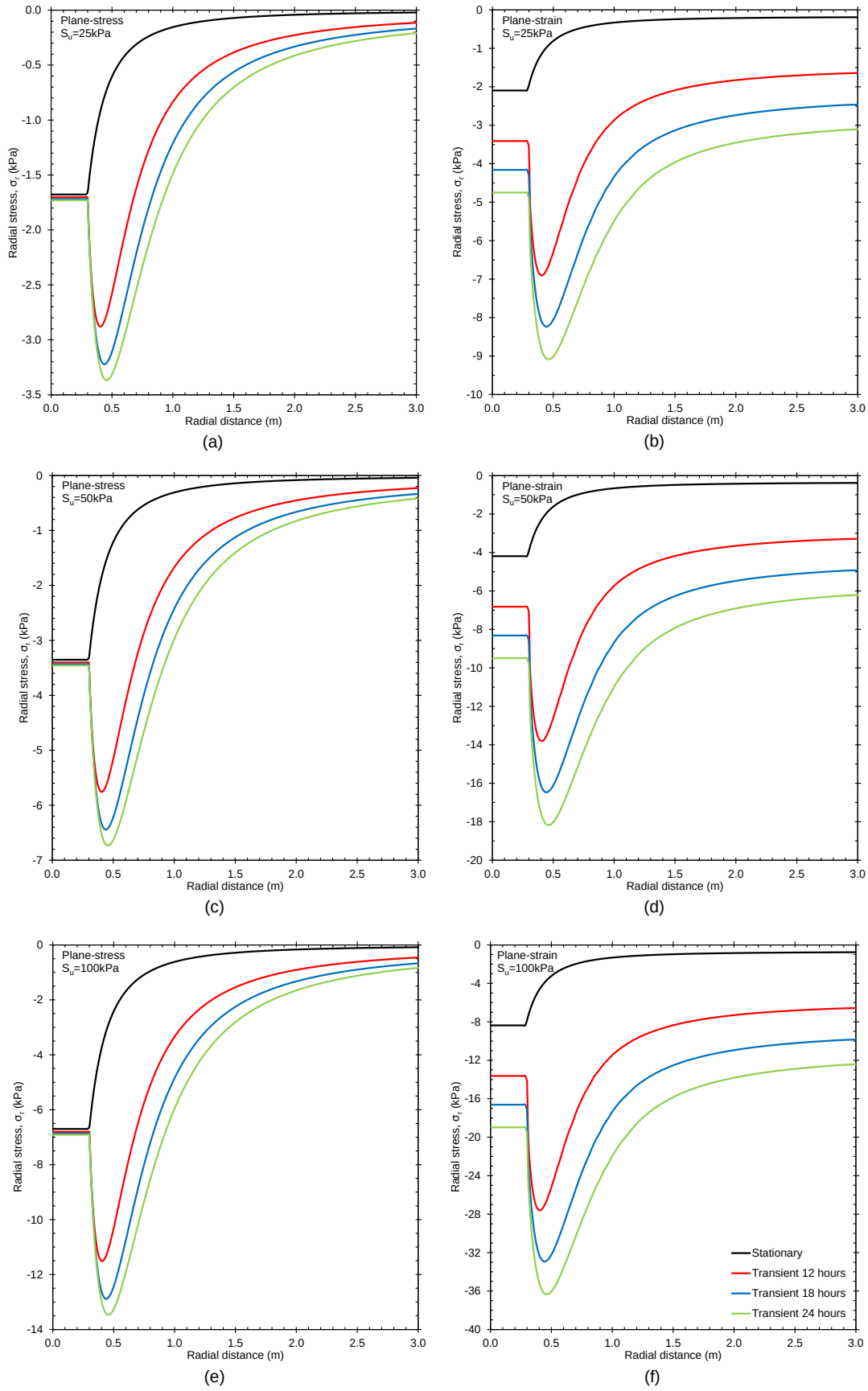


Figure 6.8 : Stress vs. radial distance for the stationary and the transient models: (a) (c) (e) Plane-strain model, $s_u = 25, 50, 100$ kPa, respectively; (b) (d) (f) Plane-stress model, $25, 50, 100$ kPa, respectively.

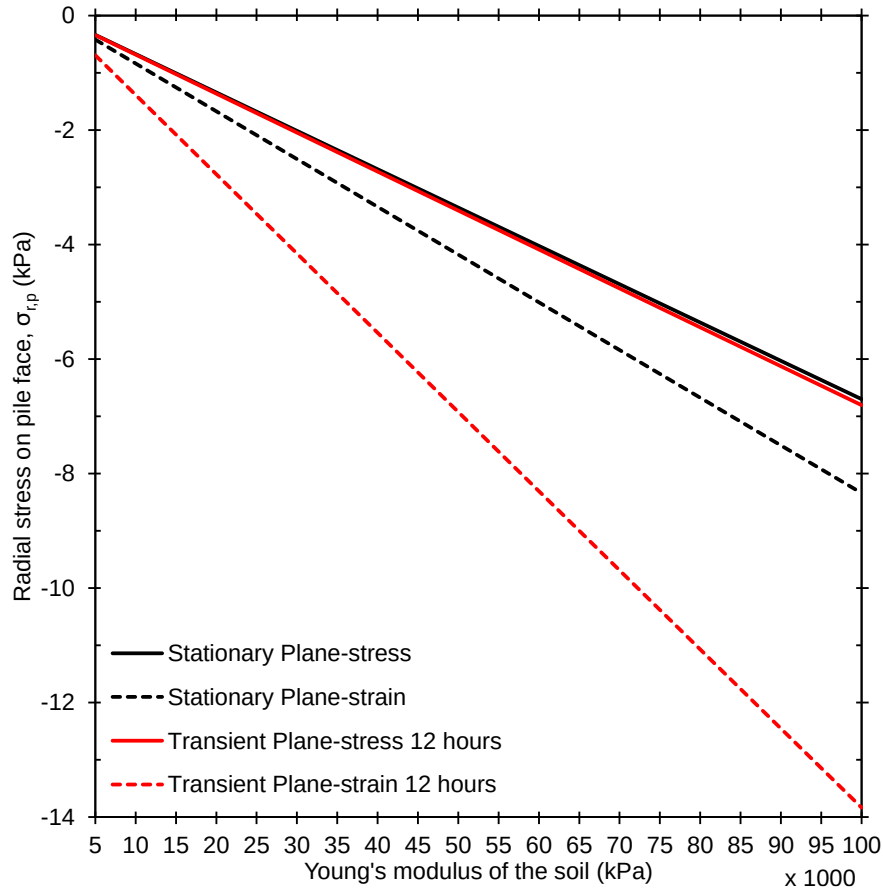


Figure 6.9 : Radial stresses at the pile face for plane stress and plane strain cases from the stationary and transient analyses ($T_{\text{initial}} = 15^{\circ}\text{C}$ and $T_{\text{final}} = 25^{\circ}\text{C}$).

Consequently, the effect of increased radial stresses applied to the pile due to thermal dilation would not have a significant effect on load-displacement behavior, should the pile be floating (plane stress) or fixed at the top and bottom (plane strain). The higher increases in contact pressure noted in the Finite Element Analysis correspond to stiffer and stronger soils (Figure 6.9). For typical values of friction coefficients at pile-soil interface, these increases in contact pressure would only slightly increase the pile-soil interface strength by 2 kPa or less for soft soils, and by 5 kPa or less for medium to stiff soils. In proportion, the contribution of heating to shaft resistance would be relatively small. We can conclude that thermal expansion of a heat exchanger pile induces small strains and this results in small magnitude stresses with minimal soil resistance. These findings from numerical analyses are in agreement with pressuremeter experiments, which show that much higher strains are necessary to develop large stresses around an expanding cylindrical element (Briaud, 1992).

6.2 Analysis of Friction Induced Thermo-Mechanical Stresses on a Heat Exchanger Pile in Isothermal Soil

Temperature-controlled triaxial tests performed on saturated kaolin samples evidenced that temperature increase causes normally consolidated soils to harden (i.e., modulus increase) (Cekerevac and Laloui, 2004). Normally consolidated clays tend to contract with heating, while overconsolidated clays dilate during heating. In addition, it was shown that the slope of the critical state line is independent of temperature, the hydrostatic yield limit (preconsolidation pressure) decreases with heating, and shear strength tends to increase with temperature (Leroueil and Marques, 1996). A thermo-mechanical elasto-plastic model was later proposed for soils subjected to cyclic temperature changes (Laloui and Cekerevac, 2008). Thermal shear hardening was also observed by comparing cyclic loading tests performed at ambient and elevated temperatures (Cekerevac and Laloui, 2010).

Thermo-hydro-mechanical soil and concrete constitutive models were implemented in several finite difference and finite element codes. In most cases, the pile is modeled as a thermo-elastic solid, and the ground is assumed to follow a thermo-poro-elastic behavior (Laloui et al., 2006; Bourne-Webb et al., 2009; Li and Zheng, 2009; Li et al., 2010; Raymond et al., 2011; Diersch et al., 2011; Amatya et al., 2012). A recent study by the authors showed that the increase in contact pressure induced by radial thermal expansion of the pile is small in magnitude and therefore would not result in significant increases in shaft resistance (Olgun et al., 2014). Free body diagrams based on one-dimensional thermo-elasticity can explain pile longitudinal contraction in heat extraction mode and pile longitudinal extension in heat injection mode (Bourne-Webb et al. 2009). The resulting vertical deformation of the pile causes differential displacements at the interface between the pile and the soil, and thus, develops friction at the soil-pile interface. More sophisticated numerical models used to study the interaction between heat exchanger piles and the surrounding soil mass focus on material properties and fluid flow in the tubes. For instance, Abdelaziz et al. (2011) studied the influence of ground thermal conductivity, grout characteristics, flow pipe diameter, flow rate and heating power on the temperature distribution around a heat exchanger pile. Water content of the soil around the pile can also impact heat exchanger pile operation as the thermal conductivity is strongly influenced by the degree of saturation. Fan et al. (2007) showed that the degree of soil saturation and

ground water flow influence the heat transfer between a heat exchanger pile and the surrounding soil.

Interface elements combining non-linear stiffness components with sliders and dilatant components were employed to mimic the behavior of granular media in simple shear at the interface with the pile (Comodromos and Bareka, 2005). Thin elements were employed to model shaft friction and the effects of localized shear around cast-in-place piles (Lee and Long, 2008; Liu et al., 2012). Contact mechanics was also used in large deformation to simulate pile installation (Fischer et al., 2007), and quantitative experimental data was recently obtained to characterize the shear behavior of soil/steel interfaces subjected to large displacements (Ho et al., 2011). However, despite experimental evidence brought on the development of shear stresses at the interface between heat exchanger piles and the ground (McCartney and Rosenberg, 2011; Suryatriyastuti et al., 2012), no model was proposed to account for the evolution of friction resistance with temperature variation in a heat exchanger pile. This implies that the displacements of the soil in contact with the pile follow the slip displacements of the outer surface of the pile, and therefore, the thermal strain incompatibilities between the soil and the pile can be neglected. The impact of slip displacements along the pile on soil shear strength has never been quantitatively assessed so far.

In this section, a mechanical interface model is utilized to account for the discontinuities and slip displacements that may occur between the soil and the heat exchanger pile (Ozudogru et al., 2015b). A model problem of heat exchanger pile is studied with the finite element method, in stationary axisymmetric conditions. Contrary to the current models available to predict internal thermo-mechanical stresses in a heat exchanger pile, an interface element is used to interpret slip displacements and shear stresses along the pile/soil contact surface of a free-top floating pile subject to prescribed elevated temperatures (Section 6.2.1). In order to understand the impact of the degree of fixity induced by the constraint of the soil in contact with the pile, a parametric study on soil strength is presented in Section 6.2.2. Interface effects are also studied for a fixed-top floating pile, and the impact of pile temperature increase on the performance of the foundation is discussed.

6.2.1 Thermo-mechanical stresses induced by friction on a heated pile floating in an isothermal soil mass

6.2.1.1 Numerical model

The goal of this numerical study is to determine whether the reaction of the soil mass plays a significant role on the thermo-mechanical performance of heat exchanger piles. Emphasis is placed on the thermo-mechanical stresses developed in the pile upon temperature increase. In practice, the steady state distribution of temperatures within a cross section of a heat exchanger pile equipped with a U-tubing system is indeed uniform, and the maximum temperature difference between the top and bottom of the pile does not exceed a few degrees (Bourne-Webb et al., 2009; Laloui et al., 2006). Former studies also indicate that the gradients of temperature expected in the soil mass upon heating and cooling cycles amount to a few degrees only (Brandl, 2006; Katzenbach et al., 2008). Accordingly, it is assumed that temperature within the pile is uniformly distributed. Since the present study focuses on the transmission of shear stress between the soil and the pile, heat diffusion within the soil around the pile is neglected: the soil is modeled as a reservoir that remains at a constant temperature. Even though this assumption seems limited compared to the actual processes in the field, it allows the uncoupling of soil-pile interaction from soil dilation and helps us study soil-pile shear transfer in the context of temperature-induced pile elongation.

The finite element model used in the analyses was built with COMSOL Multiphysics™ software (COMSOL, 2013), in stationary and axisymmetric conditions. The dimensions of the numerical models are indicated in Figure 6.10.

The top of the pile is either free (Section 6.2.1 and Section 6.2.2.1) or fixed (Section 6.2.2.2). The soil is free of stress at the top of the domain and fully fixed at the bottom of the model. In the far field, geostatic stress conditions are assumed with roller boundary condition on the right hand side boundary. The left hand side boundary represents the axis of symmetry of the domain. The heat exchanger pile is modeled as a thermo-elastic concrete cylinder embedded in either a perfectly elastic or an elastoplastic soil mass governed by Mohr-Coulomb yield criterion. Spring elements are used at the pile toe to model the normal reaction of the soil. Interface elements are used in order to model the tangential reaction of the soil on the pile shaft. The normal contact law is linear elastic, and the tangential contact law is elastic-perfectly plastic, as

illustrated in Figure 6.11. The interface shear strength is assumed to be equal to that of the soil.

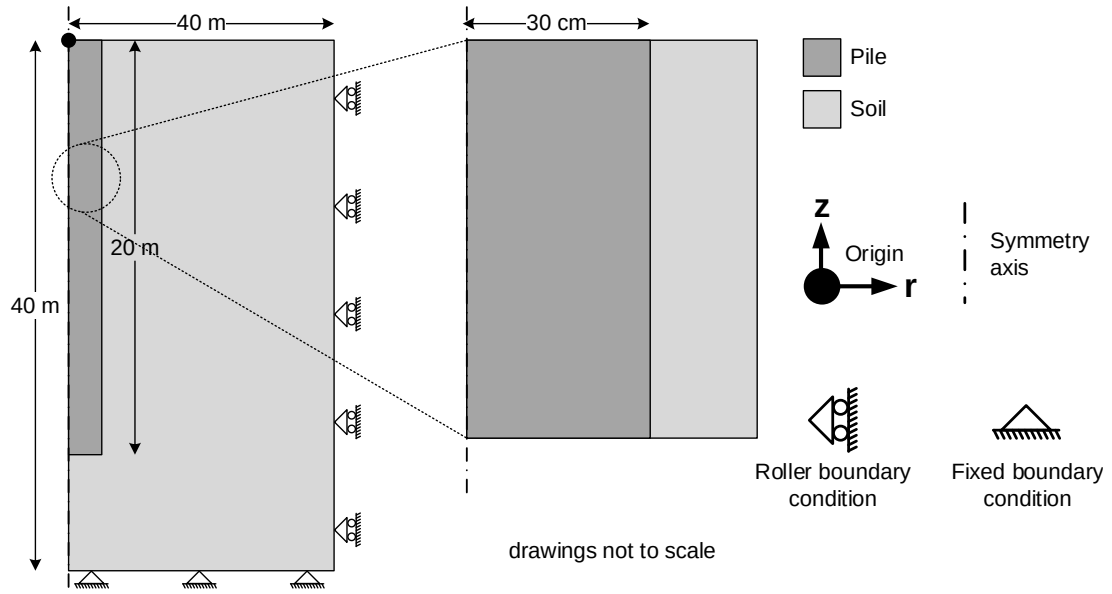


Figure 6.10 : Finite Element Model geometry and dimensions.

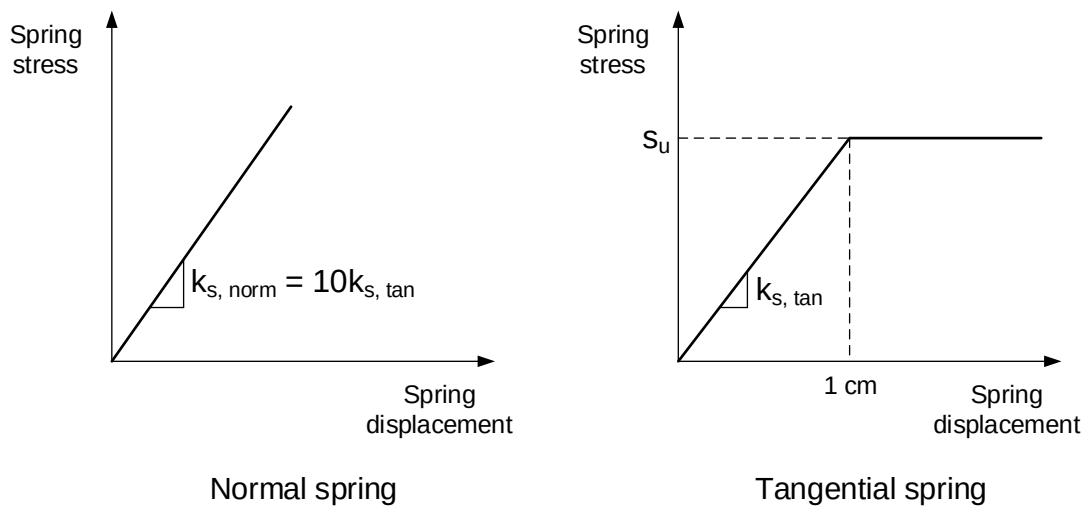


Figure 6.11 : Stress-displacement behavior of the interface element.

Table 6.4 summarizes the main constitutive equations used in the numerical model. Material parameters assigned to the pile, the soil and the interface are listed in Table 6.5 and Table 6.6.

The mesh shown in Figure 6.12 consists of 41,318 finite elements. Initially, both the soil and the pile are set at a temperature of 15°C. The temperature of the pile is uniformly increased by increments of 1°C, up to a temperature of 40°C.

Table 6.4 : Constitutive equations used in the numerical model for soil elements, concrete elements (in the heat exchanger pile) and the interface elements.

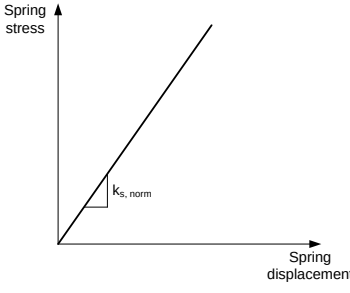
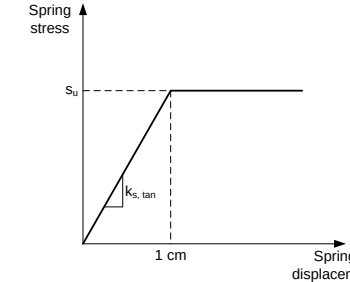
Soil	Mohr-Coulomb elastic-perfectly plastic model	$ \tau + \sigma \tan \phi - c = 0$
Concrete	Thermo-elastic model	$\sigma = \frac{E}{(1+\nu)} \varepsilon + \frac{\nu E}{(1+\nu)(1-2\nu)} Tr(\varepsilon) \delta$ $+ \frac{E}{3(1-2\nu)} \alpha \Delta T \delta$
Interface		$\sigma \cdot \mathbf{n} = -\mathbf{k}_s (\mathbf{u}_{pile} - \mathbf{u}_{soil})$
	Linear elastic normal contact law	
	Linear elastic-perfectly plastic tangential contact law	

Table 6.5 : Material properties.

Parameter / Material	Concrete	Soil
Mass density (kg/m ³)	2500	2000
Young's modulus (MPa)	17000	500 × s _u
Poisson's ratio	0.150	0.495
Thermal conductivity (W/(m·K))	1.5	2.0
Heat capacity (J/(kg·K))	1200	1500
Coefficient of thermal expansion (1/K)	1e-5	—
Cohesion (kPa)	—	25 / 50 / 100 / elastic
Angle of internal friction (°)	—	0

Table 6.6 : Soil types and strength parameters.

Parameter / Soil type	1	2	3	4
Cohesion (kPa)	25	50	100	—
Friction angle (°)	0	0	0	—
Young's modulus (MPa)	12.5	25.0	50.0	50.0
Poisson's ratio	0.495	0.495	0.495	0.495
Pile/soil tangential interface stiffness (MPa/m)	2.5	5.0	10.0	10.0
Pile/soil normal interface stiffness (MPa/m)	25.0	50.0	100.0	100.0

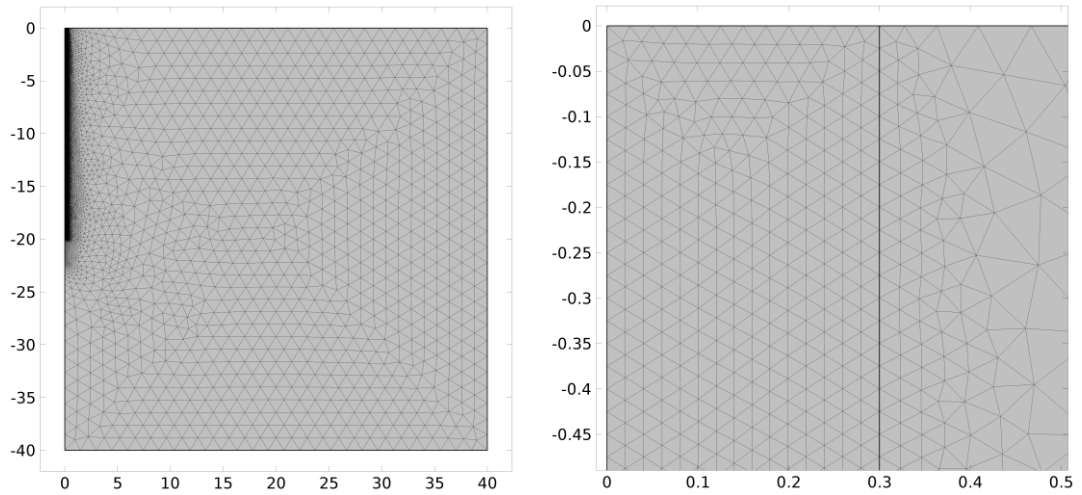


Figure 6.12 : Mesh adopted in the finite element model.

6.2.1.2 Internal stresses in a free-top heated pile embedded in elastic soil

First, the system is analyzed when the soil is perfectly elastic with a Young's modulus $E = 50 \text{ MPa}$. The pile is subjected to the normal reaction of the soil at the toe, and to normal and tangential reaction on the shaft. As a result, the vertical strain of the pile is less than that of a free pile (Figure 6.13), which would be equal to the product of the thermal expansion coefficient ($\alpha = 10 \mu\epsilon/^\circ\text{C}$) and the temperature change ($\Delta T = 25^\circ\text{C}$). The pile at mid-depth would undergo a strain of $250 \mu\epsilon$ if it was free to expand as indicated by the thermal strain in the pile as shown in Figure 6.13. The mechanical strain in the pile ($\epsilon \sim -32 \mu\epsilon$) is indicative of the compressional strains as a result of the constraint from the shear resistance of the soil at the interface in reaction to the temperature-induced pile elongation. As a result, the total strain in the pile (total strain = thermal strain + mechanical strain) is equal to $\sim 218 \mu\epsilon$ because of the induced compressional mechanical strain in reaction to the soil resistance.

It is verified that vertical stress in the soil does not change upon pile temperature increase (Figure 6.14), which is consistent with the assumptions made in the numerical model: the soil does not undergo any thermo-mechanical stress changes since soil temperature is maintained constant. Therefore, the vertical heave deformations noted in the soil at the vicinity of the heated pile are induced by a purely mechanical effect: the soil is dragged by friction against the pile outer surface as the pile undergoes temperature-induced elongation. This can be verified by comparing the vertical displacement of the soil and the pile at soil-pile contact (Figure 6.15) with the slip displacement along the interface at that point (Figure 6.16). For example, at mid-depth

of the pile ($z = 10$ m), the slip displacement due to drag slightly exceeds 0.095 mm for a 25°C temperature increase, where the vertical pile displacement is about 0.179 mm at that depth compared with the soil displacement of 0.084 mm. These results also demonstrate the importance of assuming constant soil temperature as the soil-pile interaction due to pile elongation is isolated and these effects can be observed in the absence of temperature-induced soil deformations.

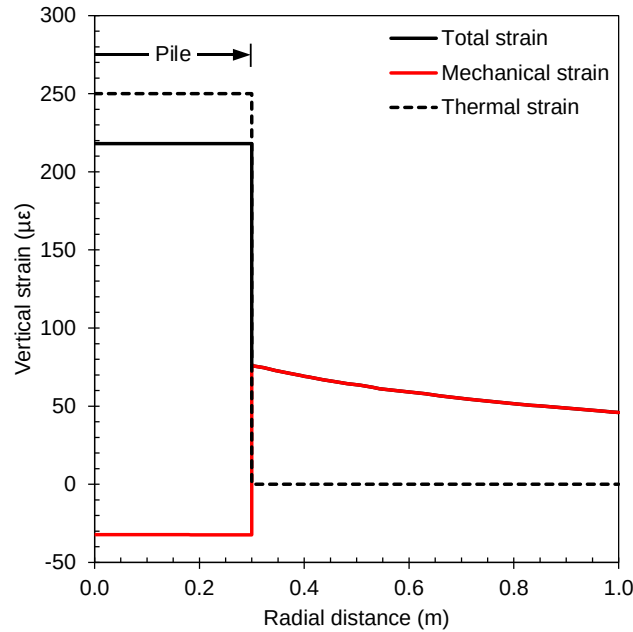


Figure 6.13 : Components of vertical strains vs. radial distance at mid-pile depth ($z=10$ m) ($\Delta T=25^\circ\text{C}$) relative to $\Delta T=0^\circ\text{C}$, for an elastic soil.

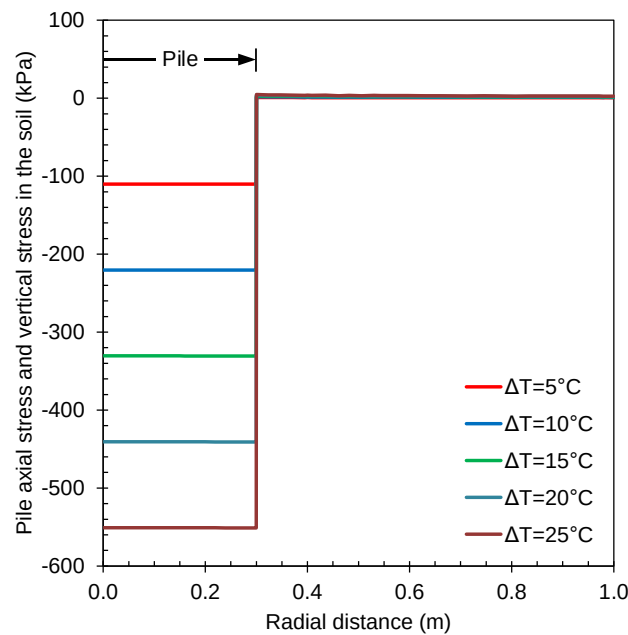


Figure 6.14 : Axial stress in the pile and vertical stress in the soil vs. radial distance at mid-pile depth ($z=10$ m) relative to $\Delta T=0^\circ\text{C}$, for an elastic soil.

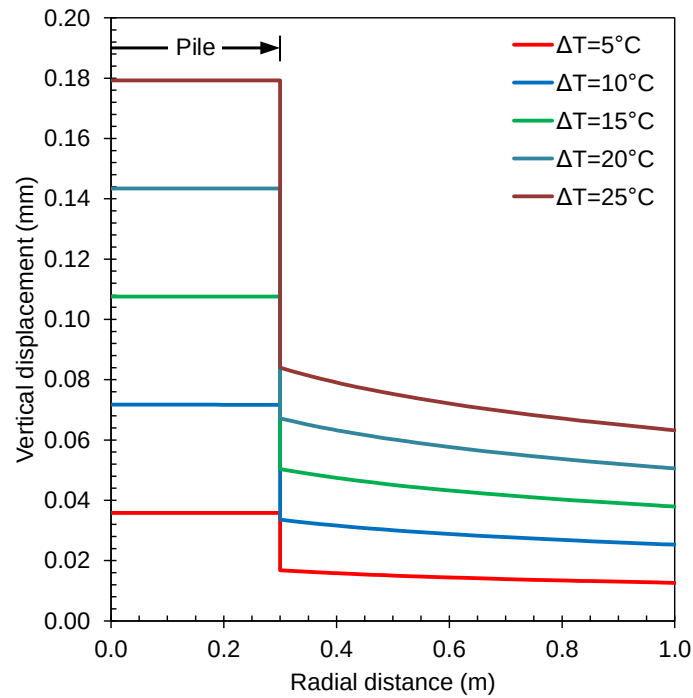


Figure 6.15 : Vertical displacement vs. radial distance at mid-pile depth ($z=10\text{m}$) relative to $\Delta T=0^\circ\text{C}$, for an elastic soil.

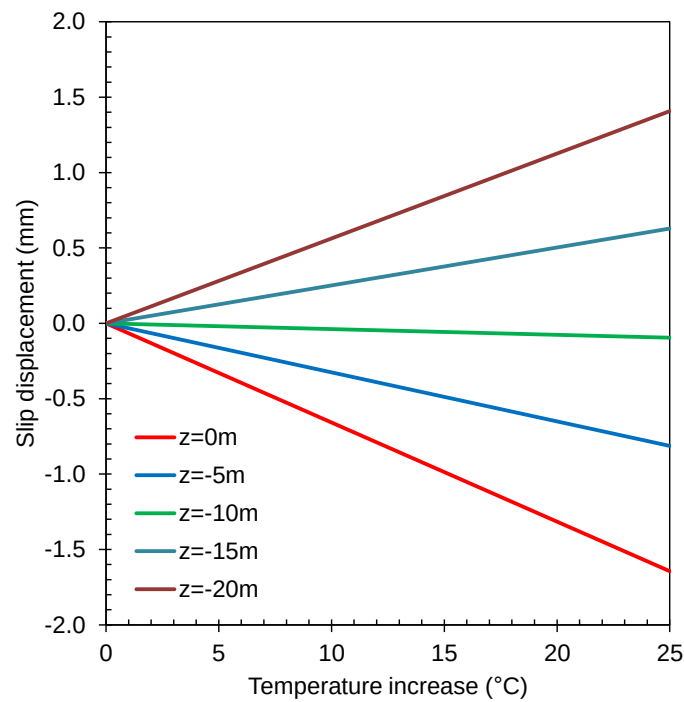


Figure 6.16 : Slip displacement at the pile-soil interface vs. temperature change at mid-pile depth ($z=10\text{m}$) relative to $\Delta T=0^\circ\text{C}$, for an elastic soil.

Figure 6.17 shows that finite fixity at the pile toe not only counteracts thermal elongation of the pile, but also narrows the zone of the pile subjected to downward displacement ($z = 10.7\text{ m} > 10\text{ m}$). The top part of the pile ($z < 10.7\text{ m}$) tends to expand

upwards upon temperature increase, while the lower segment of the pile expands downward (Figure 6.17). The neutral point at around where the pile elongates upwards and downwards would be at mid-length of the pile if the pile was expanding freely in the absence of any soil. However in this particular example, the point of inversion where the temperature-induced pile deformations transition from upward to downward (i.e. the neutral point) is located at a depth of $z = 10.7$ m as seen in Figure 8. Soil reaction at the pile toe in addition to the shear resistance along the pile shaft causes the neutral point to propagate to levels deeper than mid-length of the pile. The neutral point also represents where the opposing soil resistances along the upward and downward segments of the pile are balanced. Consequently, the compressional stress due to pile elongation and the associated soil resistance is maximized at the neutral point where thermo-mechanical shear stresses acting on the pile reverse from downward to upward along the pile length. The thermo-mechanical stress distribution would be symmetrical if the pile was free at both ends for an elastic soil with uniform stiffness throughout. Stress distributions for different levels of temperature increase in the analyses are shown in Figure 6.18 and it is seen that the compressional stress along the pile achieves a maximum value of 552.9 kPa at $z = 10.7$ m depth.

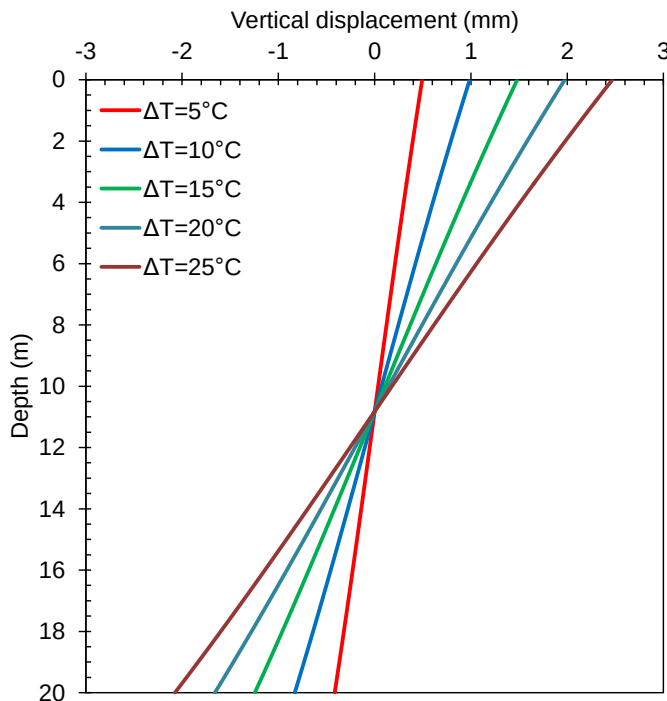


Figure 6.17 : Vertical displacement along the pile center axis relative to $\Delta T=0^{\circ}\text{C}$, for an elastic soil.

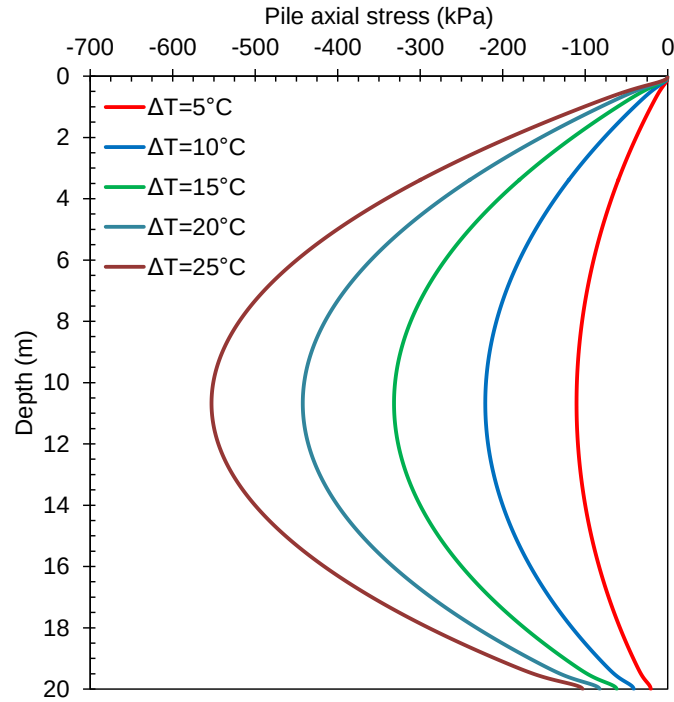


Figure 6.18 : Axial stress along the pile relative to $\Delta T=0^{\circ}\text{C}$, for an elastic soil.

6.2.1.3 Interface behavior - elastic soil

Aside from the pile ends, the total normal stress at the interface equals the radial thermo-mechanical stress developed in the pile. For the elastic soil case, the interface contact is linear elastic in both directions; normal stiffness and tangential stiffness being 100 MPa/m and 10 MPa/m, respectively. Normal stress increase at the soil-pile interface is small and remains in the order of a few kilopascals (Figure 6.19). This is an important observation suggesting that the normal stress changes at the pile-soil interface are likely to be small and therefore temperature changes will not have a significant influence on pile capacity (Olgun et al., 2014). Slip displacement along the interface remains in the order of a few millimeters (Figure 6.20), consistent with the small magnitude of deformation difference between the pile and the soil at mid-length of the pile (Figure 6.15). The linear relationship between slip displacement and interface shear stress is reflected in Figure 6.20 and Figure 6.21. The distribution of shear stress along the interface follows a trend opposite to that of slip. Moreover, it can be verified that the amount of shear stress mobilized along the soil-pile interface is equal to the product of the tangential interface stiffness and the slip displacement (in agreement with the model), which amounts to a maximum of less than 20 kPa for a temperature increase of $\Delta T = 25^{\circ}\text{C}$. The comparison of Figure 6.17 and Figure 6.21 allows the verification of the balance of axial forces on any portion of the pile:

$$\pi r^2 \sigma_a(z) = \pi r^2 \sigma_a(z=0) + 2\pi r \int_0^z \tau(z') dz' \quad (6.1)$$

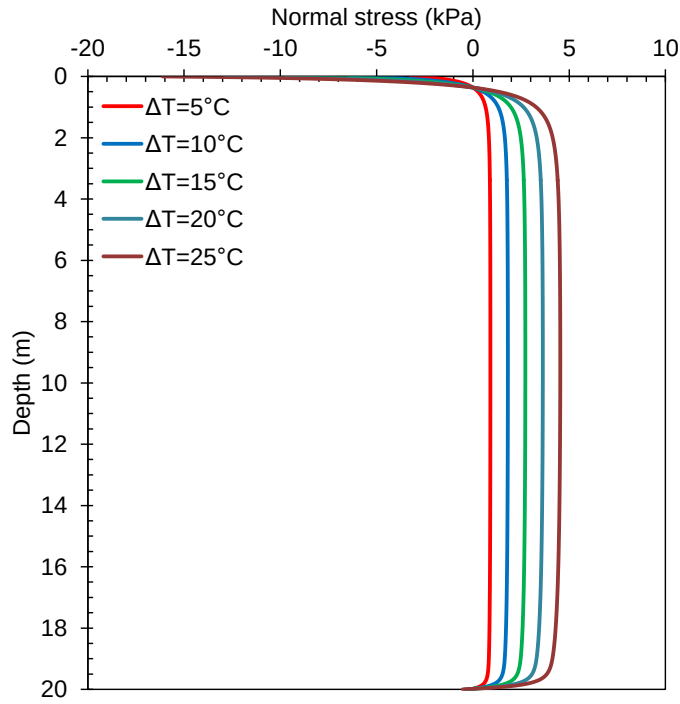


Figure 6.19 : Normal stresses along the pile-soil interface relative to $\Delta T=0^\circ\text{C}$, for an elastic soil.

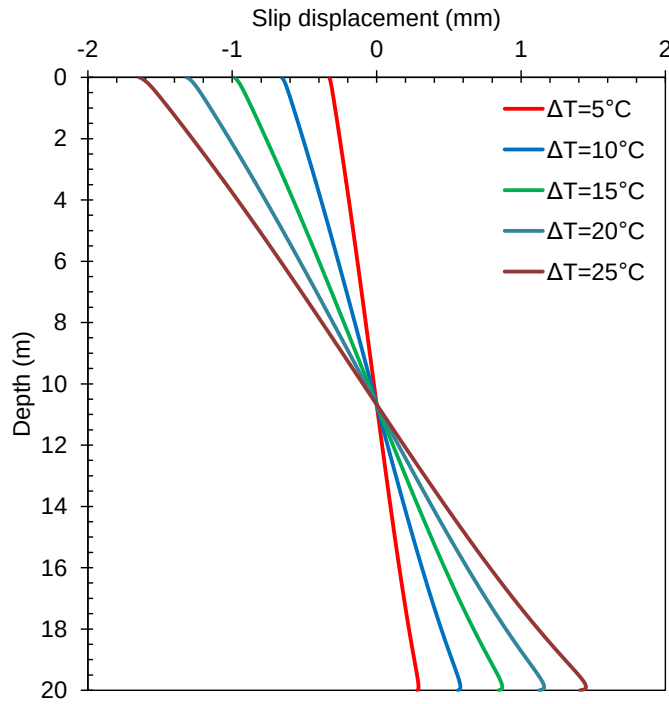


Figure 6.20 : Slip displacements along the pile-soil interface relative to $\Delta T=0^\circ\text{C}$, for an elastic soil.

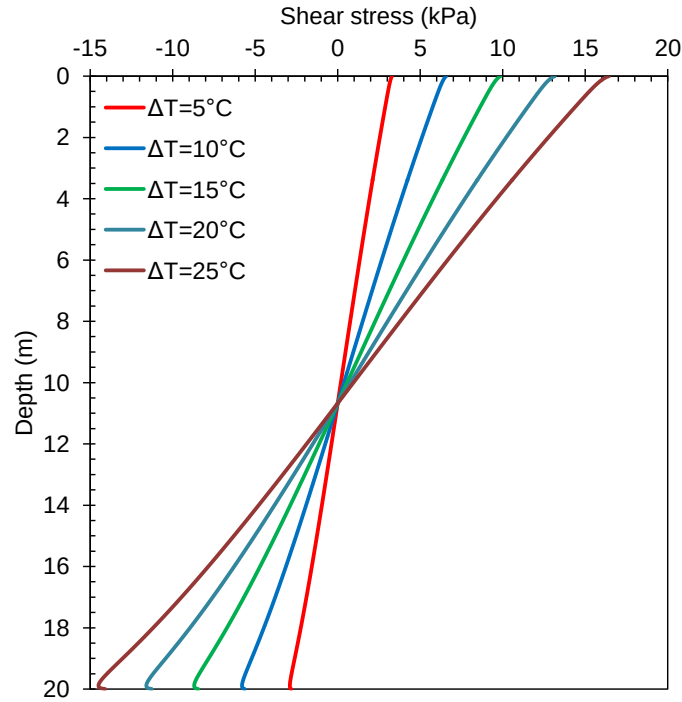


Figure 6.21 : Shear stresses along the pile-soil interface relative to $\Delta T=0^{\circ}\text{C}$, for an elastic soil.

The equation above shows that the shear stress mobilized at the interface is balanced by the axial stress developed within the pile. Recommendations made in previous thermo-mechanical analyses (Laloui et al., 2006) indicate that floating heat exchanger piles typically need to be reinforced near the top of the pile, in order to support the additional stresses induced by temperature increase. This is particularly important for piles constrained at the top in the presence of a structure as opposed to the pile analyzed above. Additional analyses are presented below where fixity of the pile top is investigated.

6.2.1.4 Validation of the numerical model by a field test

The finite element model presented above was validated using the measurements from the field test presented by Amatya et al. (2012). This field test was located at Lambeth College in South London. The test setup included a mechanically loaded heat exchanger pile and a heat sink pile. The main test pile and the heat sink pile were subjected to reverse thermal operations, such that when the main test pile was cooled down, the heat sink pile was heated with the heat extracted from the main pile.

Data collected for the heat sink pile was used for model validation. The pile had a diameter of 55 cm and a length of 30 m. The head of the heat sink pile was free and had no mechanical load or constraint. The drilling log indicated that the first 4 m of

the soil profile consisted of granular fill and sand and gravel layer, underlain by a stiff, fissured silty clay layer. The clay layer extended below the toe of the pile. The strength parameters were evaluated solely from standard penetration test (SPT) data and it was observed that the undrained soil strength increased with depth (Figure 6.22a).

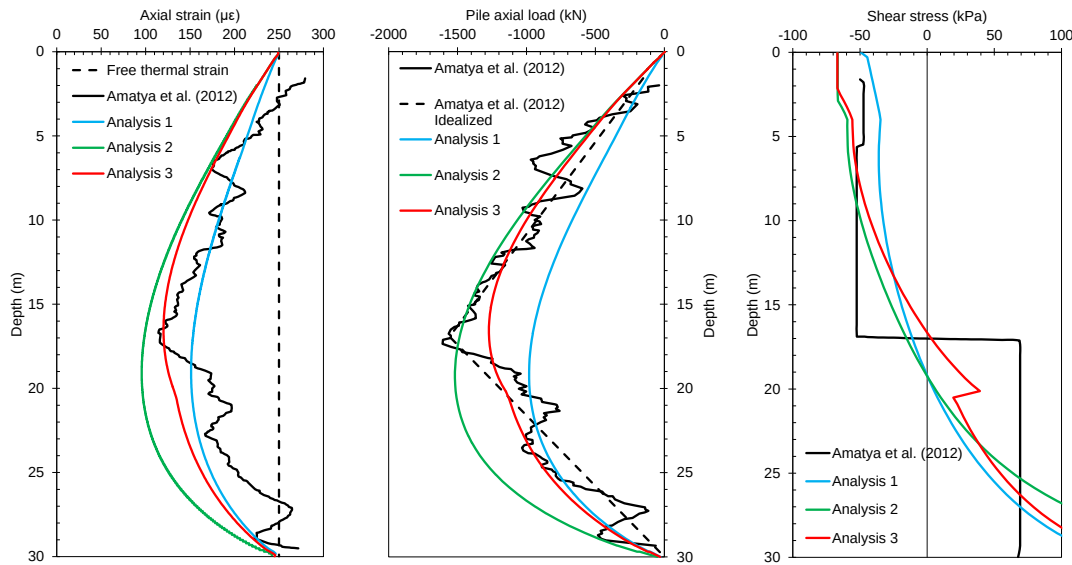


Figure 6.22 : Soil and interface parameters along the depth: (a) Undrained shear strength, (b) Young's modulus; (c) Tangential spring constant along the pile-soil interface.

The Young's modulus of the concrete was 40 GPa and the coefficient of thermal expansion was $8.5 \mu\epsilon/^\circ\text{C}$. During the field test, strain and temperature data were continuously measured with optical fiber sensors (OFS) in addition to strain gauges at discrete elevations within the pile. The heat sink pile was subjected to a temperature increase of 29.4°C and it was observed that the temperature change was uniform along the entire length of the pile. The strain data was used to evaluate an idealized axial load profile (Figure 6.23b) and the shear stress along the pile-soil interface was interpolated using this idealized profile (Figure 6.23c).

For validation purposes, three finite element analyses were performed with different Young's modulus to shear strength ratios (E/S_u) and for different slip displacements at fully mobilized shaft resistance. Details are given in Table 6.7 and shown in Figure 6.22. The geometry and the mesh were similar to those presented in the preceding sections. Pile temperature was gradually and uniformly increased to achieve a temperature increase of 29.4°C in reference to the initial pile and soil temperature.

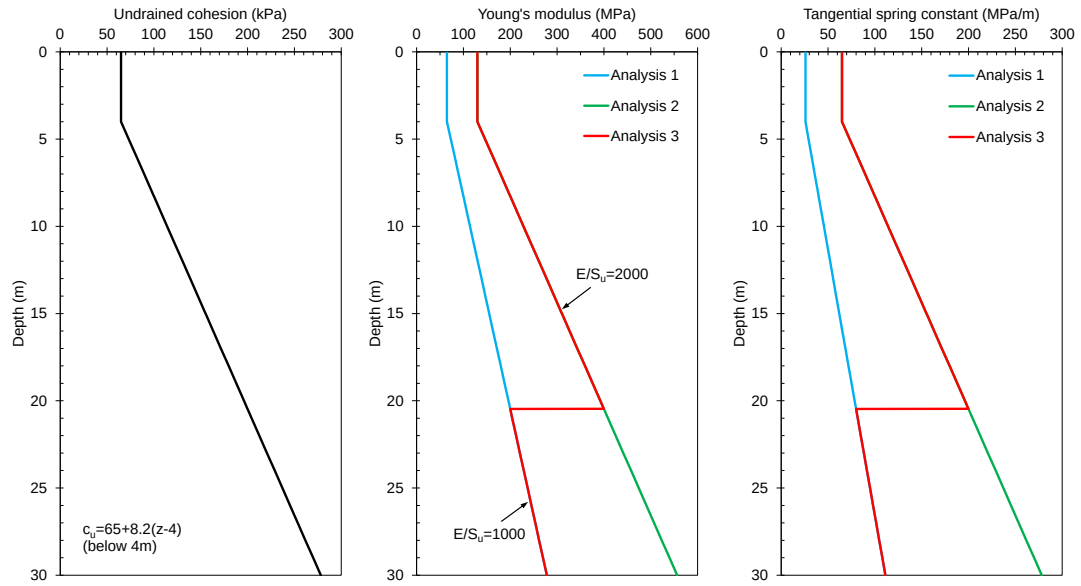


Figure 6.23 : Comparison of field test measurements and finite element analyses: (a) Axial strain along the length of the pile; (b) Pile axial load along the depth; (c) Shear stress along the pile-soil interface.

Table 6.7 : Input parameters used in the analyses for model validation.

Input parameter / Analysis #	Analysis 1	Analysis 2	Analysis 3
E/S_u	1000	2000	2000, $S_u \leq 200$ kPa 1000, $S_u > 200$ kPa
Slip displacement at fully mobilized shaft resistance (mm)	2.5	1.0	1.0, $S_u \leq 200$ kPa 2.5, $S_u > 200$ kPa

The results of the numerical analyses are presented in Figure 6.23, along with the field measurements. It can be seen that, Analysis 3 captures the peak axial strain while Analysis 1 has a better fit of the average Figure 6.23a). The field data shows that the strain values measured near the ends of the pile (e.g., the first and the last 4 m) are slightly higher than the free thermal strain, which is theoretically impossible. Figure 14b shows the evaluated pile axial load with respect to depth. It can be inferred that the value of the peak axial load is approximately predicted by Analysis 2, but the general behavior of the pile axial load is best captured by Analysis 3. It is not practical to compare the shear stress distribution along the pile-soil interface since it was interpolated using the idealized axial load profile. On the other hand, the neutral point provided by Analysis 3 is the only one which agrees with the field measurements (Figure 6.23c). Overall, it can be concluded that the finite element modeling approach presented in this study works well in simulating thermo-mechanical behavior of a heat exchanger pile under temperature changes. These results also highlight the importance of soil and interface parameters on the results.

6.2.2 Parametric analyses

6.2.2.1 Influence of soil strength on the thermo-mechanical stress induced by friction in a heated pile floating in isothermal soil

The previous analysis shows that slip displacements between a heated pile and the surrounding soil are small in magnitude and the maximum value of the shear stress mobilized at the interface is in the order of 16 kPa near the pile top for a temperature increase of 25°C. In order to study the impact of potential soil yielding on the mechanical performance of the heat exchanger pile, simulations were repeated for an elasto-plastic soil, for the following shear strengths: $S_u = 25\text{ kPa}$, 50 kPa and 100 kPa , instead of the elastic soil discussed in Section 6.2.1. Young's modulus of the soil in different sets of parametric analyses was varied to maintain a modulus to shear strength ratio $E/S_u = 500$. The stiffness and the yield limit of the tangential spring in the interface elements were varied accordingly (Table 6.4). The other material parameters, the mesh, the initial conditions and the boundary conditions are kept the same as in Section 6.2.1.

The difference between the tangential displacement of the pile and that of the soil along the interface are shown in Figure 6.24. The resulting shear stress distributions for the three soil strength cases and the elastic case are shown in Figure 6.25. The magnitude of the slip displacement at the soil/pile interface (i.e. difference in tangential displacement between soil and the pile) is very similar for the different soil models. In fact, the distribution of slip displacement for the case with $S_u = 100\text{ kPa}$ is almost identical to the case with elastic soil. Moreover, the shear stress distributions for different cases vary in proportion to soil modulus for different cases. Figure 6.26 shows that the variation of axial stress in the pile is almost proportional to the stiffness of the tangential spring in the interface. For instance, the values of peak axial stress at $z = 10.7\text{ m}$ are 150.6 kPa, 292.5 kPa and 552.9 kPa for the soils with $S_u = 25$, 50 and 100 kPa, respectively. The ratios of the maximum axial stresses with respect to the $S_u = 100\text{ kPa}$ soil are 0.27/0.53/1.00 for $S_u = 25$, 50 and 100 kPa, respectively. It is seen that the magnitude of shear stresses and axial pile stresses are almost proportional to the stiffness of the soil-pile interface, which is directly correlated to soil strength and stiffness in this model. Moreover, the axial stress in the pile for $S_u = 100\text{ kPa}$ case is almost identical to the case with pile in elastic soil. The amount of slip at the interface is always below 1 cm, which implies that the interface remains elastic for all

the soil conditions investigated. Similar conclusions can be drawn from the variations of strains within the pile (not shown herein for the sake of brevity). While thermal strains remain at the same value for the four cases for a given temperature increase, mechanical strains increase almost linearly with soil shear strength. This is related to the same phenomenon described above. Identical piles in soils with different strengths undergo similar thermal strains in response to a given temperature increase. However, the mechanical strains are manifested in proportion to the interface strength in the corresponding soil mass with a given soil strength and stiffness.

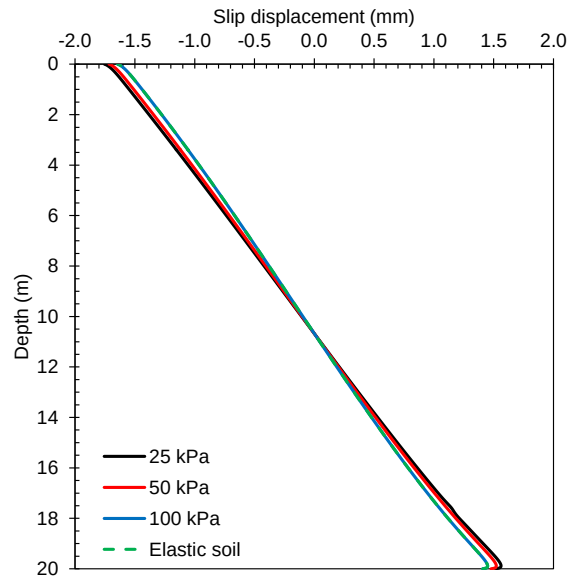


Figure 6.24 : Slip displacements along the pile-soil interface relative to $\Delta T=0^{\circ}\text{C}$, for $\Delta T=25^{\circ}\text{C}$, in the four soils considered in this study.

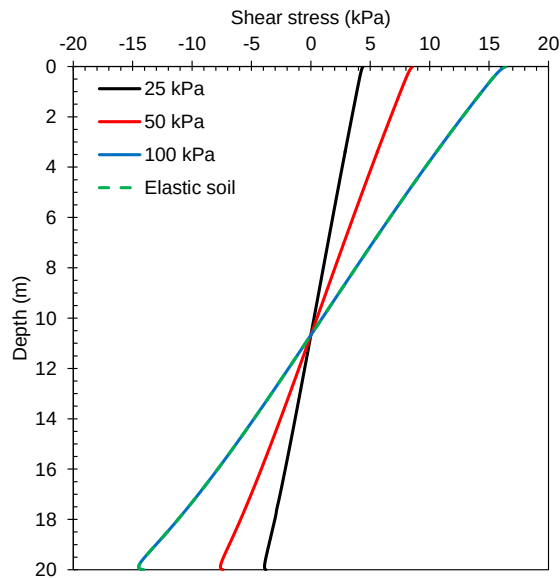


Figure 6.25 : Shear stresses along the pile-soil interface relative to $\Delta T=0^{\circ}\text{C}$, for $\Delta T=25^{\circ}\text{C}$, in the four soils considered in this study.

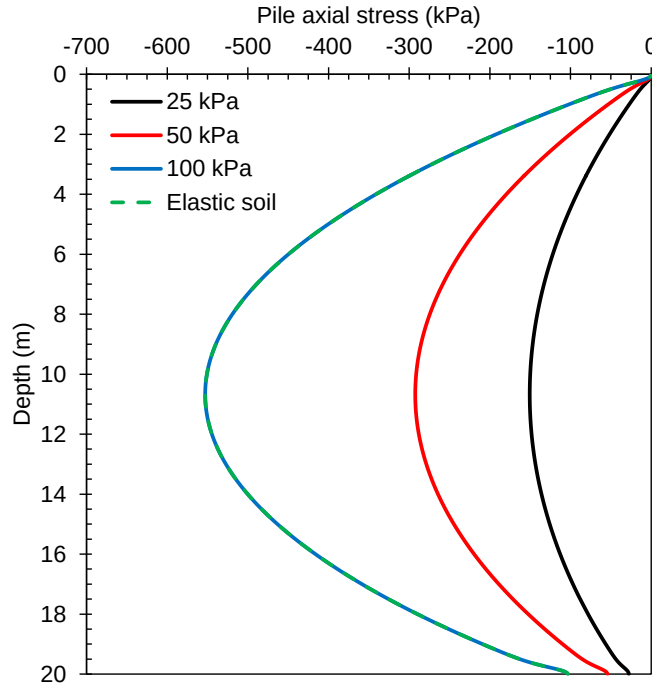


Figure 6.26 : Axial stresses along the pile relative to $\Delta T=0^{\circ}\text{C}$, for $\Delta T=25^{\circ}\text{C}$, in the four soils considered in this study.

6.2.2.2 Influence of the pile fixity on the thermo-mechanical stresses induced by friction in a heated pile floating in isothermal soil

The four simulations above were repeated for an identical pile fixed at the top in order to determine the effect of a building on the temperature-induced stress increases within a heat exchanger pile. It is quite conceivable that the constraint from a building would impose a different displacement field along the pile, which will directly influence the development of slip displacements and shear stresses along the pile length. In these analyses, displacements at the top of the pile were fully fixed, which represents an end-case in comparison to a building with a foundation that will provide some degree of constraint but will still have some finite stiffness. All the other properties of the numerical model were kept the same as in Section 6.2.1 and Section 6.2.2.1.

The main effect of heating a pile with a fixed top is the concentration of compressional stresses near the top of the pile, due to the mechanical boundary conditions imposed on the pile. In this case, soil friction plays a minor role compared to the constrained vertical elongation of the pile due to the fixity at pile top. The maximum values of compressional stress increase near the top of the pile due to heating are 1157.5 kPa, 1582.2 kPa and 2228.4 kPa for the soils with $S_u = 25, 50$ and 100 kPa, respectively

(Figure 6.27). Similarly, the pile in elastic soil behaves almost identical compared to the pile within the soil with $S_u = 100$ kPa.

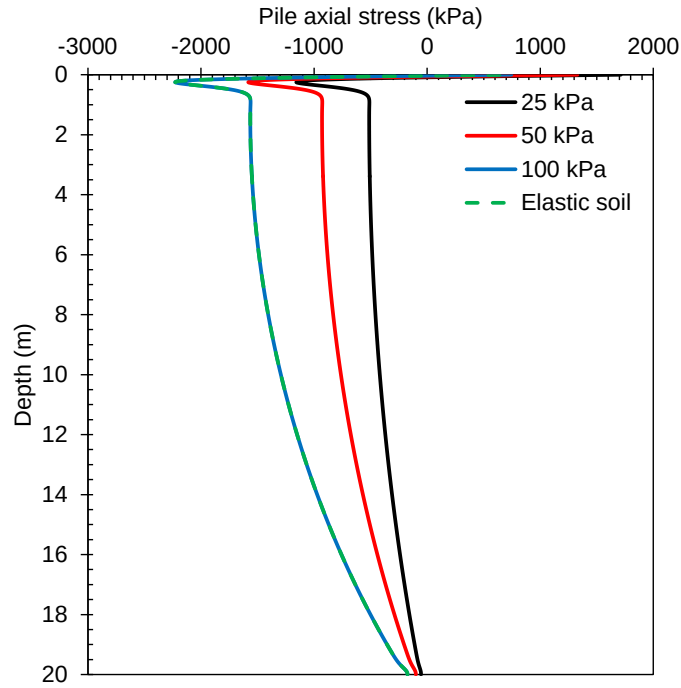


Figure 6.27 : Axial stresses along the pile relative to $\Delta T=0^{\circ}\text{C}$, for $\Delta T=25^{\circ}\text{C}$ (fixed top).

It is also seen that some level of tensile stresses develops near the top of the pile, presumably as a result of the soil heave from the upward deformations of the soil at the lower levels. Upward slip of the soils near the ground surface relative to the fixed pile top and the associated shear stresses result in tensile stresses near the top of the pile. Shear stresses along the interface are negative and act upward within a very narrow zone, up to about 1.5 m depth (Figure 6.28). This is demonstrated by the slip displacements which are negative within the top 1.5 meters and positive at deeper levels along the pile (Figure 6.29). This indicates that the soil around the pile tends to displace upwards compared to the pile along the soil-pile interface at these upper levels. These results show that heave of the soils near the top and the associated shear stresses along this segment of the pile causes the development of tensile stresses. The displacement field within the soil around the pile indicates that the upward displacement of the soils at the lower levels pushes the soils near the top upwards. This secondary displacement field causes a relative upwards slip displacement of the soil at the upper levels with respect to the pile with fixed top causing tensile stresses along the pile. This underscores the importance of modelling the full displacement field

which would only be possible with an approach similar to the one presented in this study. Other simpler approaches which employ springs to model soil-pile interface would not be able to capture such a phenomenon related to the full displacement field and coupling of soil deformations at different levels (Knelwolf et al., 2011).

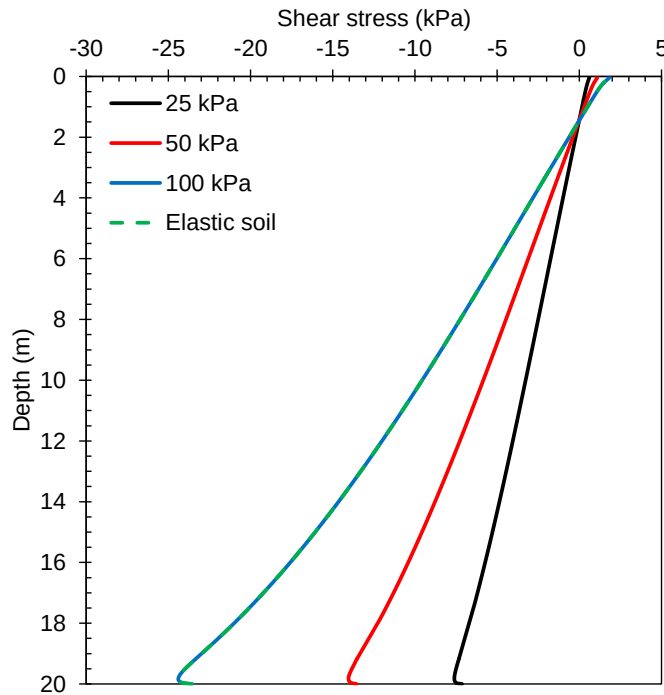


Figure 6.28 : Shear stresses along the pile-soil interface relative to $\Delta T=0^{\circ}\text{C}$, for $\Delta T=25^{\circ}\text{C}$ (fixed top).

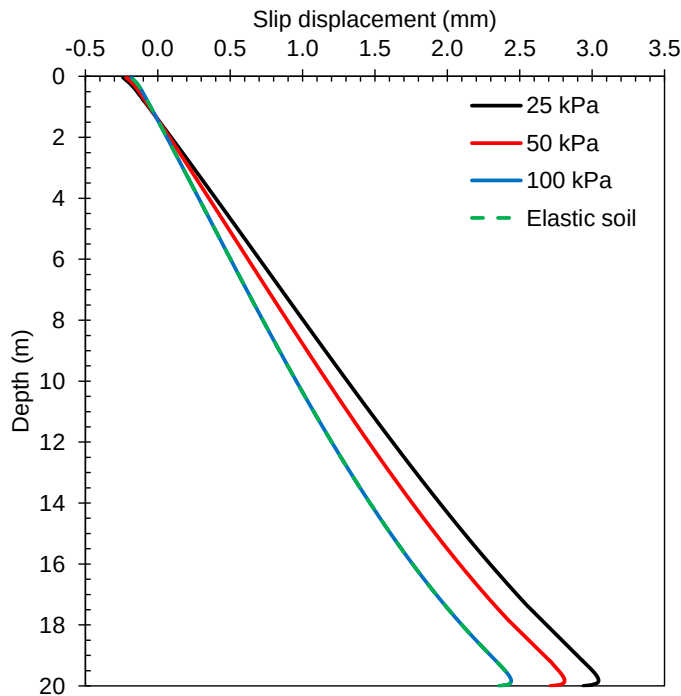


Figure 6.29 : Slip displacements along the pile-soil interface relative to $\Delta T=0^{\circ}\text{C}$, for $\Delta T=25^{\circ}\text{C}$ (fixed top).

7. LONG-TERM PERFORMANCE OF HEAT EXCHANGER PILES

Sustainable operation of heat exchanger piles or any other thermo-active foundation element relies on maintaining the constant temperature of the ground. Ground temperature changes can directly affect the heat exchange capacity and efficiency of operations. It is critical to evaluate how the heat exchanger pile operation over the life cycle of the structure will change the temperature gradients around the pile. This is particularly important at regions where the respective heating and cooling energy demands are not balanced.

In this chapter, numerical analyses have been performed to investigate the long-term performance of heat exchanger piles (Olgun et al., 2014a). Energy demand analyses were performed to derive the base energy load for a typical commercial building at three different climatic environments. These loads were applied to a heat exchange pile to simulate heat exchange operations using a numerical model. Thermo-mechanical response and geotechnical performance of heat exchanger piles were also investigated. The analyses considered thirty years of heat exchange operations and resulted in significant findings for long-term performance of heat exchanger piles under different climatic conditions. The numerical analysis approach and the fundamental results are summarized. The main emphasis is given to practical implications and the effect of energy demand is investigated for different climatic conditions.

7.1 Long-Term Performance of Heat Exchanger Piles

Heat exchanger piles are dual functioning deep foundation elements for structural support which are used conjunctively as heating/cooling elements, as illustrated in Figure 7.1. As shown, the piles contain circulation tubes and act as heat exchangers, as heat energy from the superstructure is circulated through the tubing with water/antifreeze mix. Heat energy is fed into and withdrawn from the ground for cooling in the summer and heating in the winter respectively. Heat exchange is performed with a heat pump. Of particular importance, energy cost savings in typical

buildings could be as much as 80% (Hamada et al., 2001). The added cost of geothermal loops for heat exchange is relatively small because heat exchanger piles are used where deep foundation elements are already planned for structural support.

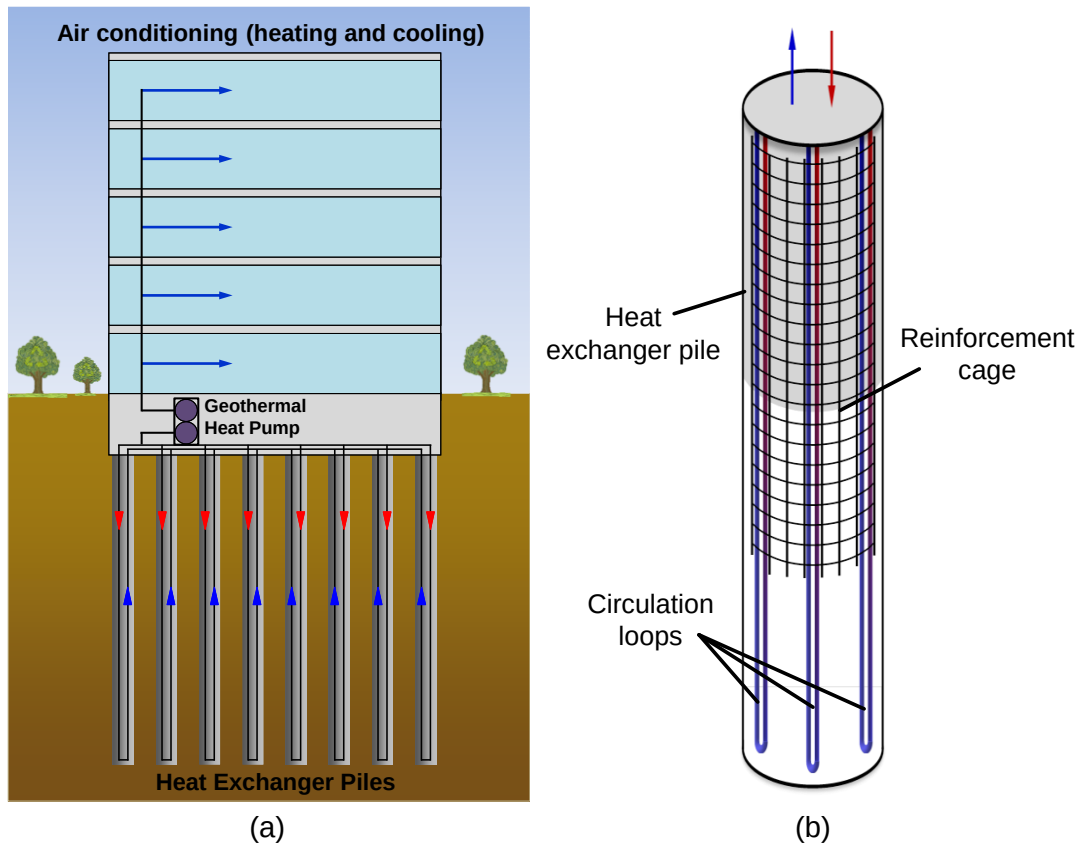


Figure 7.1 : (a) Schematic showing a building supported by heat exchanger piles (b) Typical configuration of a heat exchanger pile with integrated circulation loops.

Ground-source heat pump systems are advantageous over air-source heat pump systems because they exploit the ground with a more favorable baseline temperature for heating and cooling of buildings. For example, as the ambient temperature in Blacksburg, VA is -2°C (28°F) during the winter, the ground with an undisturbed temperature of 13°C (55°F) presents a heat source for raising the room temperature to the comfort zone levels. A similar temperature differential between the ambient temperature and the constant ground temperature also provides the potential for building cooling in the summer. Therefore it is critical to maintain the stable temperature of the ground over seasons for long-term sustainability of heat exchange operations.

The balance of heating and cooling between consecutive episodes ensures the sustainable utilization of the ground as a renewable energy source. A balanced thermal

exploitation helps offset the thermal effects of each respective cooling and heating season. Under certain circumstances heat exchange operations can generate anisotropic thermal fields in the ground as shown in Figure 7.2. This is particularly pronounced if the heat energy extracted in the winter for space heating (i.e. ground cooling) is not balanced in comparison to the heat energy injected into the ground in the summer for cooling purposes (i.e. ground heating).

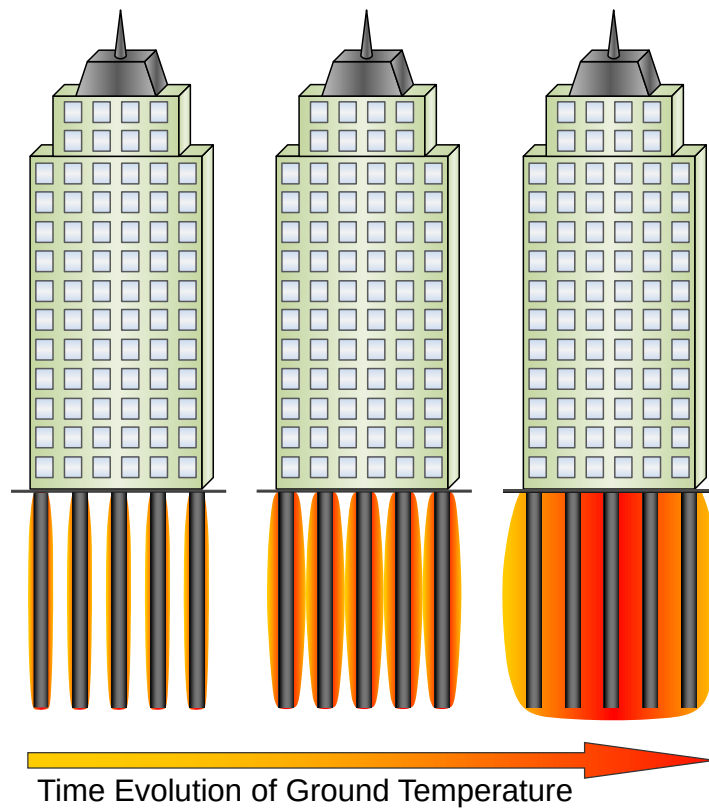


Figure 7.2 : Long-term temperature change around heat exchanger piles and evolution of temperatures in the ground.

It is necessary to evaluate how the heat exchange operations will change the temperature gradients around the pile over the life cycle of the structure. It is quite possible that the heat exchange operations over seasons can alter the temperature fields around the heat exchanger piles especially when energy demand is non-symmetrical over seasons. Long-term progression of temperatures around a heat exchanger pile has implications in terms of the efficiency of heat exchange operations. Ground can progressively cool when the energy demand is heating dominated in colder climates and on the contrary it can progressively get warmer in climates where energy demand is cooling dominated in warmer climates. It is also possible that the load carrying capacity of the pile can be compromised due the potential temperature induced effects

on soil compressibility around the pile. Similarly, if imbalanced thermal loads are induced over long time periods, and/or if key soil parameters are altered by these loadings, the foundation system might undergo differential movements as well.

Thermo-mechanical response and geotechnical performance of heat exchanger piles were examined experimentally and numerically by several researchers. Wang et al. (2011) studied the effect of temperature on the pile shaft resistance by using a laboratory scale model. The tests were performed at different vertical stress levels, temperature gradients and heat exchanger modes. Similarly, McCartney and Rosenberg (2011) carried out centrifuge tests to study the impact of different temperature increase levels on the shear stress distribution along the pile. Load-settlement curves were analyzed for various mechanical loads in isothermal conditions. Salciarini et al. (2013) developed a numerical model to investigate the soil-structure interaction of a piled raft induced by temperature changes. A foundation system of an ideal water tank tower embedded in a realistic soil profile was considered and a series of coupled thermo-mechanical analyses were performed on a number of heat exchanger pile layouts.

Laloui et al. (2006) and Bourne-Webb et al. (2009) performed full scale instrumented thermo-mechanical field tests to study the change in pile loads due to thermal effects. Both studies suggested that significant changes in the axial load distribution along the pile axis can occur as a consequence of temperature variations. On the other hand, the response of the heat exchanger pile is thermo-elastic, such that the pile expands and contracts elastically during thermal loading cycles. Bourne-Webb et al. (2009) clarified the mechanisms of heat exchanger pile behavior during heating and cooling and presented a conceptual model to explain the pile response to thermal loadings, as shown in Figure 7.3.

The conceptual model proposed by Bourne-Webb et al. (2009) considers a frictional heat exchanger pile. Figure 7.3a shows that as the pile is mechanically loaded pile head displaces and resistance is mobilized at the pile-soil interface. The pile contracts under cooling loads and the surrounding soil resists the thermal contraction (Figure 7.3b). During the cooling phase, the axial loads in the pile are reduced and tensile stresses may be observed in the lower part of the pile in extreme cases (Figure 7.3c). The reverse applies when the heat exchanger pile is in heating mode (Figure 7.3d and Figure 7.3e).

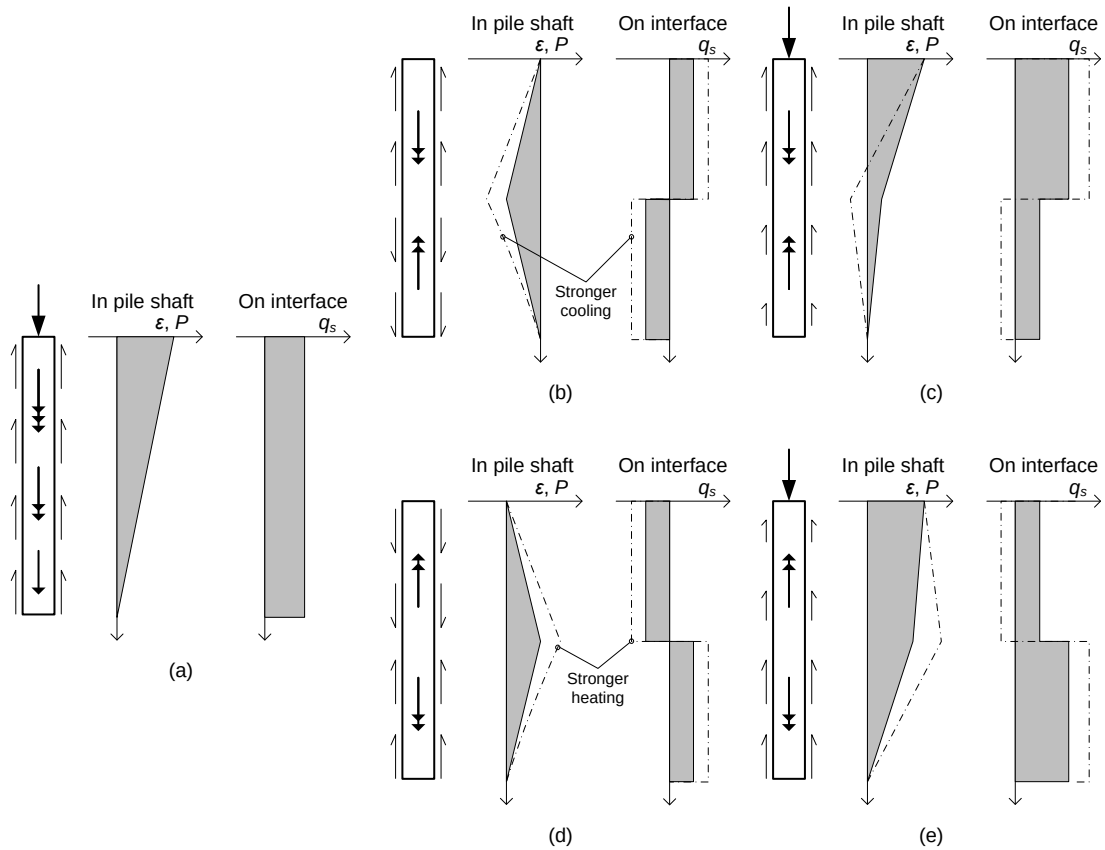


Figure 7.3 : Mechanisms for response of pile to thermal loading: (a) mechanical load only; (b) cooling only; (c) combined load and cooling; (d) heating only; (e) combined load and heating. (after Bourne-Webb et al., 2009).

7.2 Energy Demand Analysis

Energy demand and temperature progression analyses were performed in an effort to investigate the sustainable utilization of the ground as a renewable thermal resource. An office building with four rooms is utilized to estimate the seasonal energy demand for heating and cooling. Layout of the office building is shown in Figure 7.4. Energy demand and the estimated ground thermal loads were further used to model long-term heating and cooling of the ground around the heat exchanger piles.

Building energy demand was estimated for three locations of general interest for long-term performance of ground coupled heat pump systems; 1) A location where heating and cooling loads are reasonably balanced (Charlotte, NC), 2) Energy demand is heating dominant (Chicago, IL), and 3) Energy demand is cooling dominant (Austin, TX). Energy demand calculations utilized the seasonal climate at each specific location as well as the building properties, heat energy utilization patterns of the end-users (Abdelaziz, 2013).

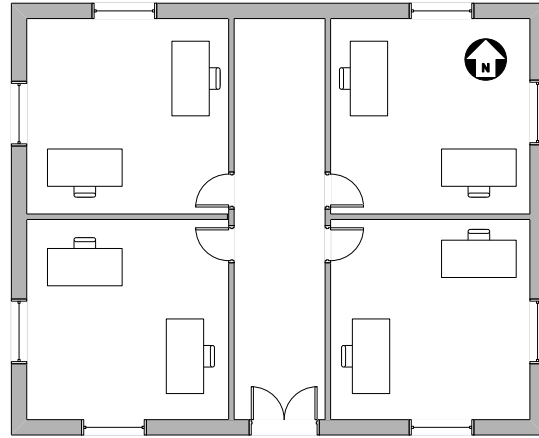


Figure 7.4 : Layout of the office building used in the energy demand analyses.

Annual temperatures at these locations are shown in Figure 7.5. As seen, the mean annual temperature in Charlotte, NC is 15.5°C with a moderate to warm climate. We anticipate the energy demand in Charlotte to be seasonally balanced. On the other hand the mean annual temperature in Chicago, IL is 10.8°C with a colder climate. Mean annual temperature in Austin, TX is 20.8°C with a much warmer climate. Chicago is selected such that the colder climate at this location corresponds to a case where the heating demand is more dominant and the ground will be used predominantly for heating purposes in the winter, whereas the cooling in the summer is less pronounced. On the contrary the warm weather in Austin will be representative for cases where the energy demand is cooling dominant and the ground will potentially be heated progressively.

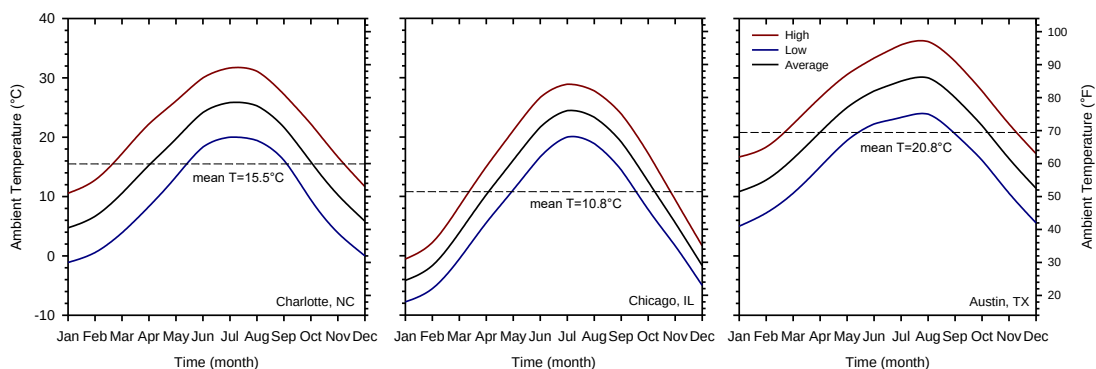


Figure 7.5 : Monthly average ambient temperatures at three selected locations and the corresponding estimated annual mean temperatures.

Energy demand analyses were performed for this office building for the three locations. These analyses considered the ambient temperature variations at these locations as well as energy consumption patterns for space heating and cooling. Realistic parameters were used to characterize the building. Therefore, even though

this is a generic analysis, it captures the important trends that govern the explored concepts of seasonal energy use at different climates. Ground thermal loads were calculated by distributing the total thermal load over the total geothermal loop length such that the maximum ground thermal load was about 50 W/m. Details of the energy load calculations can be found in Abdelaziz (2013) and the results are presented in Figure 7.6 for these three cities.

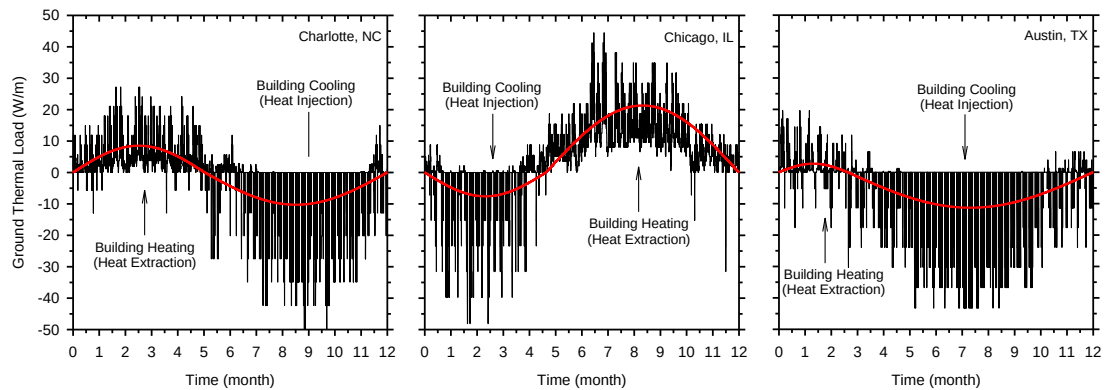


Figure 7.6 : Estimated seasonal energy demand curves at three selected locations and the representative sine wave approximations.

It is seen that the duration and amplitudes of heating and cooling episodes in winter and summer are similar for Charlotte. A slightly higher cooling load is apparent from the reported ground thermal loads. Even though the energy demand is fairly symmetrical between winter and summer in Charlotte, the operational energy of a geothermal heat pump induces additional energy to the system and results in the development of a heat injection bias for the ground thermal load.

The colder climate in Chicago results in an energy demand dominated by heating in the winter. As seen, the duration of the heating episode in the colder season spans over 7 months. This imbalance between heating and cooling loads presents a potential for progressive cooling of the ground. On the contrary, the energy demand is cooling dominant in Austin, representing the other side of the energy imbalance spectrum. The cooling dominated nature of the energy demand in Austin is exemplified by the higher amplitude and longer duration of the cooling episode compared to short and low amplitude annual heating cycle. Heat exchange operations in Austin are likely to result in the progressive heating of the ground. Collectively these cases represent different energy demand patterns and will help us investigate thermal gradients around a heat exchanger pile for these climatic conditions.

7.3 Numerical Simulation of Long-Term Thermal Operations

A two-dimensional numerical model was developed for performing the long-term energy demand analyses by utilizing the readily available modules of COMSOL Multiphysics™, a finite element simulation environment (COMSOL, 2013). The model uses a horizontal cross-section of the heat exchanger pile–soil system and it consists of several components. These include the fluid, circulation pipes, concrete, and the soil surrounding the pile. The heat exchanger pile used in the analyses is 60 cm in diameter, within a homogeneous, isotropic soil medium. The pile was assumed to have double 1 1/4" HDPE tubes placed symmetrically with a cross alignment. Figure 7.7 shows the cross section of the pile used in the analyses. The center-to-center distance between the upward and the downward legs of the tubes is 35 cm.

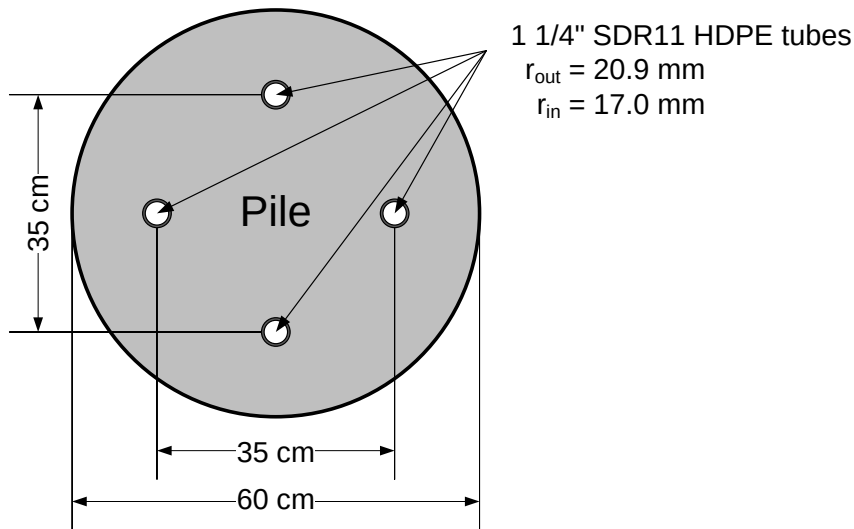


Figure 7.7 : Cross section of the heat exchanger pile used in the finite element analyses.

The variation of temperature distribution along the vertical direction will be small as the predominant heat exchange takes place radially. Therefore any vertical thermal process is assumed to have negligible influence on the long-term thermal pile operation and the 3D heat transfer problem is reduced to a 2D time-dependent heat conduction problem which is governed by the following equation:

$$\rho c_p \frac{\partial T}{\partial t} + \nabla \cdot (-\mathbf{k} \nabla T) = q \quad (7.1)$$

Equation (7.1) is solved for temperature, T , making use of the assigned boundary conditions. No heat flux (insulation – Neumann) boundary condition was specified for the extents of the model. The extent of the model was selected to be relatively large

(>100 m) in order to eliminate potential external boundary effects during the long-term thermal operation. It was ensured that there is no temperature change at these boundaries throughout the runtime of the model. The Neumann boundary condition with no heat flux can be expressed as:

$$q'' = (-\mathbf{k}\nabla T) = 0 \quad (7.2)$$

Fluid flow inside the pipes and the associated convective heat transfer is simulated by an equivalent solid, which has the same heat capacity (i.e. mass density and specific heat capacity) as the circulation fluid. On the other hand, the thermal conductivity of the equivalent solid is selected to be very high (1000 W/m·K). With this approach, the fluid domain can be utilized to inject a specific temperature or to inject/extract heat per unit length of the pile, at selected time periods during the analysis. High thermal conductivity also leads to a practically uniform temperature distribution within the cross-section of each pipe leg. The seasonal ground thermal loading was simulated by applying the sine wave approximated energy demand curves as a time dependent heat source on the fluid domains.

In this modeling approach, the thermal resistance due to the convective heat transfer between the water and the pipe inner wall is neglected. On the contrary, the convective thermal resistance is taken into account as a conductive resistance in the pipe wall, by considering an effective thermal conductivity of the pipe material $k_{p \text{ eff}}$ in the analyses given by:

$$k_{p \text{ eff}} = \frac{\ln(d_{po}/d_{pi})}{\frac{2}{d_{pi} h_{int}} + \frac{\ln(d_{po}/d_{pi})}{k_p}} \quad (7.3)$$

where, h_{int} is the internal convective heat transfer coefficient, k_p is the actual thermal conductivity of the pipe material, d_{po} and d_{pi} are the outside and inside diameters of the pipe, respectively. The internal convective heat transfer coefficient depends on the flow characteristics and the thermal properties of the water and it is evaluated using the correlations developed by Gnielinski (1976) and Churchill (1977). The modeling technique involving the equivalent solid approach explained above is adopted from a study by Lazzari et al. (2010). It should also be noted that, since the axial direction is not considered in the model, the heat rate per unit length is equally distributed in all pipes without making any distinction between upcoming and downgoing pipes. Thus,

the temperature of the equivalent solid corresponds to the mean temperature of the fluid in the pipes. Water was used as the heat exchange fluid in Charlotte and Austin, while antifreeze (20/80 propylene glycol and water mixture) was used in Chicago due to the colder climate. The thermal and rheological properties of the water-antifreeze solution are taken from the ASHRAE Handbook Fundamentals (ASHRAE, 2009). The thermal properties of the ground and the heat exchanger material are selected by referring to the studies by Farouki (1981) and Salomone et al. (1989). The input parameters used in the numerical analyses are listed in Table 7.1.

Table 7.1 : Parameters used in the thermal analyses.

Parameter	Value	Unit
Water		
Freezing point	0	°C
Flow rate	20	dm ³ min ⁻¹
Dynamic viscosity	1.0	mPa s
Thermal conductivity	0.59	W m ⁻¹ K ⁻¹
Effective thermal conductivity	1000	W m ⁻¹ K ⁻¹
Specific heat capacity	4187	J kg ⁻¹ K ⁻¹
Density	1000	kg m ⁻³
Antifreeze (20/80)		
Freezing point	-7.5	°C
Flow rate	20	dm ³ min ⁻¹
Dynamic viscosity	4.1	mPa s
Thermal conductivity	0.46	W m ⁻¹ K ⁻¹
Effective thermal conductivity	1000	W m ⁻¹ K ⁻¹
Specific heat capacity	3929	J kg ⁻¹ K ⁻¹
Density	1026	kg m ⁻³
Pipe		
Thermal conductivity	0.39	W m ⁻¹ K ⁻¹
Effective thermal conductivity, water	0.37	W m ⁻¹ K ⁻¹
Effective thermal conductivity, antifreeze	0.32	W m ⁻¹ K ⁻¹
Specific heat capacity	2300	J kg ⁻¹ K ⁻¹
Density	960	kg m ⁻³
Pile		
Thermal conductivity	1.50	W m ⁻¹ K ⁻¹
Specific heat capacity	1200	J kg ⁻¹ K ⁻¹
Density	2500	kg m ⁻³
Ground		
Thermal conductivity	2.00	W m ⁻¹ K ⁻¹
Specific heat capacity	1500	J kg ⁻¹ K ⁻¹
Density	1600	kg m ⁻³

The finite element mesh discretization of the models is performed by using triangular elements. Quarter symmetry is used in all of the models to reduce the computational effort, since the geometry and the physical processes in all models are symmetrical.

The mesh of the single pile model consists of 7,966 triangular domain elements and the mesh of the 5x5 pile group model with 6 diameters center-to-center spacing has 34,357 triangular domain elements. The meshed model domain and the quarter symmetry approach are shown in Figure 7.8.

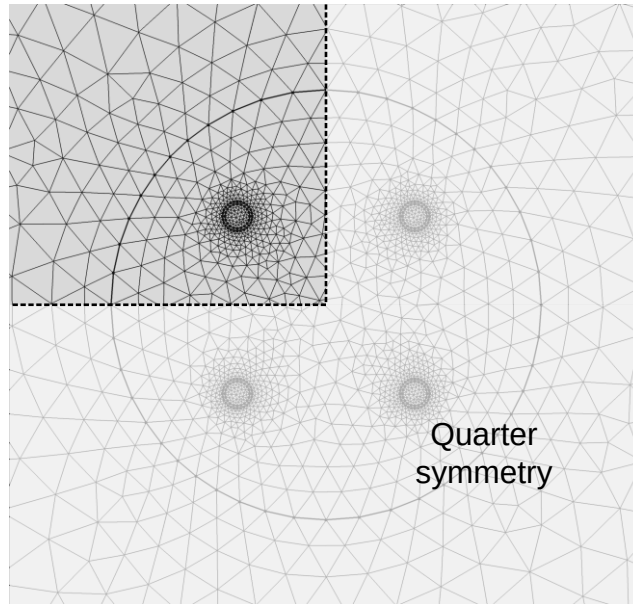


Figure 7.8 : Finite element mesh of the model and the use of quarter symmetry.

A series of numerical simulations were performed for several pile arrangements ranging from single pile to numerous pile groups with a selection of 2x2, 3x3, 4x4 and 5x5 rectangular grids. 3, 4, 5 and 6 diameters center-to-center spacings were assumed for the pile groups, conforming to the guidelines set forth in Federal Highway Administration Drilled Shafts Design Manual (FHWA, 2010).

It is computationally expensive to use the actual thermal loads to perform the long-term analyses over 30 years of heat exchange operations. Therefore, an equivalent sine wave approach was developed to represent the ground thermal load for long-term analyses and applied the annual sine wave consecutively for a duration of 30 years to simulate long-term operations.

Overlays of the sine wave approximations with respect to the seasonal energy demands at the selected cities are shown in Figure 7.6. The equivalent sine wave provides a smoother thermal load curve that can be analyzed with a much larger time step compared to the 30-minute time step of the analysis with the actual thermal load. The equivalent sine wave is selected such that it results in ground temperatures that match those from the actual ground thermal load profile particularly at the end of the annual

cycle. Details of the sine wave approximation method are presented in Abdelaziz (2013).

Validity of the sine wave approximation was confirmed by comparison of finite element analyses using the 1-year simulations that utilized the thermal loads based on the actual energy demands and the representative sine waves. The results are shown in Figure 7.9, where the temperatures at the pile surface are plotted for the analyses that used the actual ground thermal loads and the equivalent sine wave loads. The temperatures computed using the equivalent sine wave are in very good agreement with the results from the model with the actual thermal loads. This confirms the validity of the equivalent sine wave approach as the actual thermal loads can be represented with equivalent sine waves. As mentioned above, the equivalent sine wave approach allows a much more efficient computational effort by making it possible to perform long-term analyses over an extended duration.

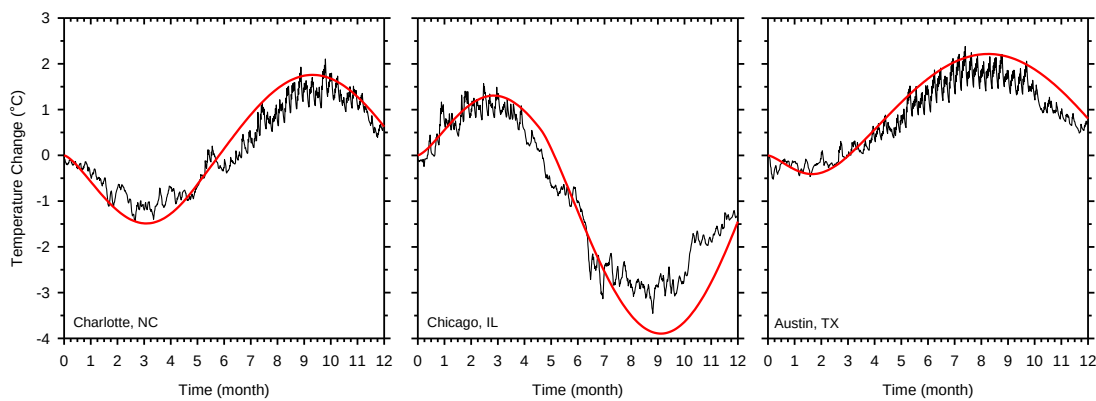


Figure 7.9 : Computed pile surface temperatures for 1-year of heat exchange operations at the three locations using the actual ground thermal loads and the representative sine waves.

The equivalent sine waves are further used to perform the finite element analyses over 30 years to simulate long-term performance for the three selected locations. The results of the long-term analyses for the single pile are shown in Figure 7.10, Figure 7.11 and Figure 7.12, where long-term progressions of temperatures for Charlotte, Chicago and Austin are presented, respectively. It can be seen that the temperatures at the pile center and the pile face stay relatively unchanged in Charlotte at the end of 30 years of heat exchange operations. The temperature at the pile face starts from the initial ground temperature of 15.5°C and fluctuates with seasonal episodes of heating and cooling. Similarly ground temperatures at 0.5 diameter and 1 diameter from the pile face are

also shown and these results demonstrate that the temperature fluctuations occur at smaller amplitudes at increasing distances from the pile.

It is seen that there is a slight increase of ground temperatures at the end of 30 years due to the above mentioned effect of the additional heating induced by the geothermal heat pump operation. This additional heat plays a role to imbalance towards a net increase of building cooling demand. Nevertheless Charlotte represents a reasonably well balance between heating and cooling demands. The temperature fluctuations at this location are due to the ground heating in the summer and ground cooling in the winter which apparently offset each other after each seasonal cycle. This shows that, relative and opposite effects of heating and cooling episodes are critical for offsetting the temperature induced changes for a sustainable heat exchange operation.

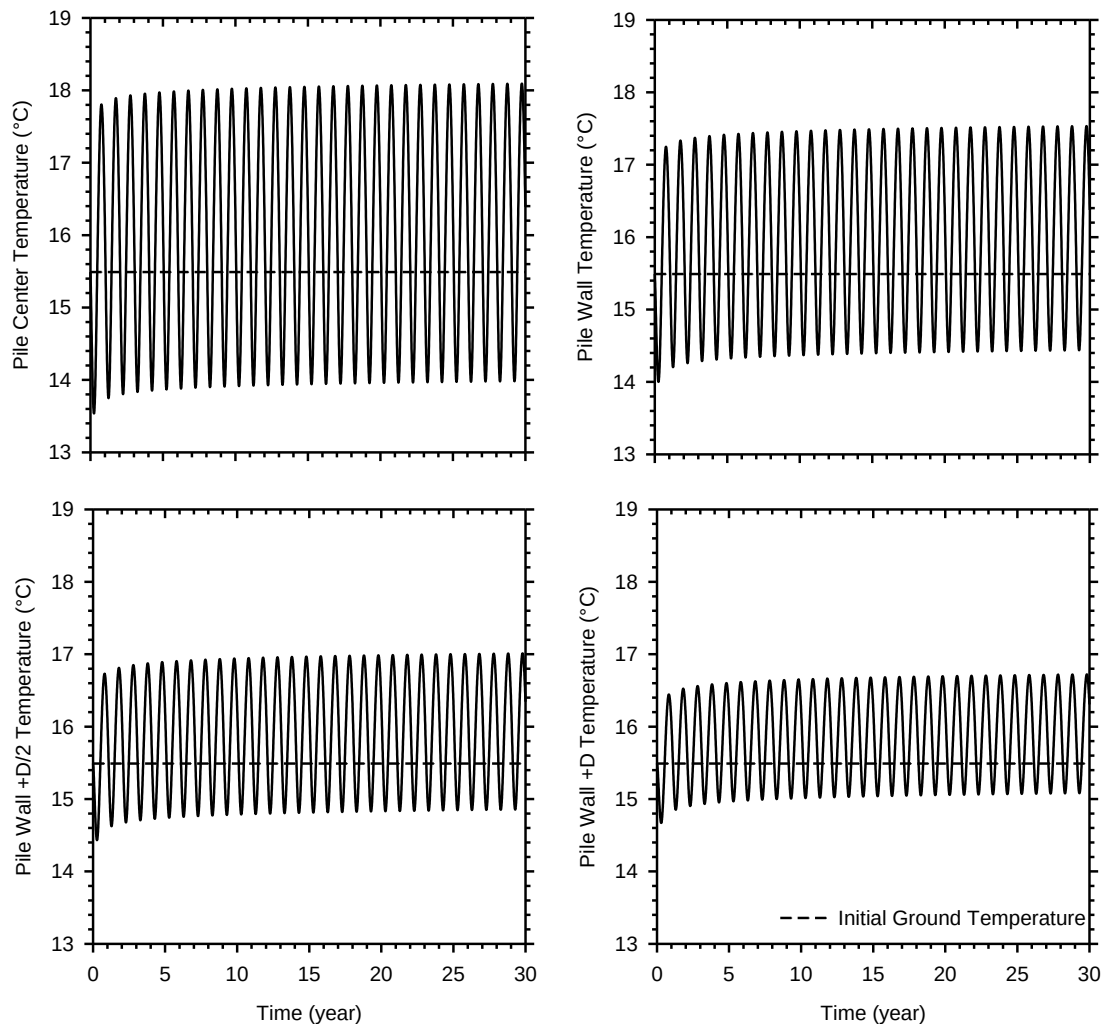


Figure 7.10 : Progression of temperatures in and around the single pile for long-term operations in Charlotte, NC.

Pile and ground temperatures as a result of long-term heat exchange operations for the single pile in Chicago are shown in Figure 7.11. Ground progressively gets cooler in Chicago where the energy demand is heating dominated and an unbalanced amount of heat energy is extracted from the ground over seasons. On the contrary the ground gets progressively warmer at Austin, as the energy demand is primarily cooling dominated as shown in Figure 7.12. These two cases where energy demand is imbalanced represent two opposite end of the energy demand scenarios. Induced temperature changes at these two cities show similar but opposite trends, each reflecting the nature of the energy imbalance for the corresponding climate.

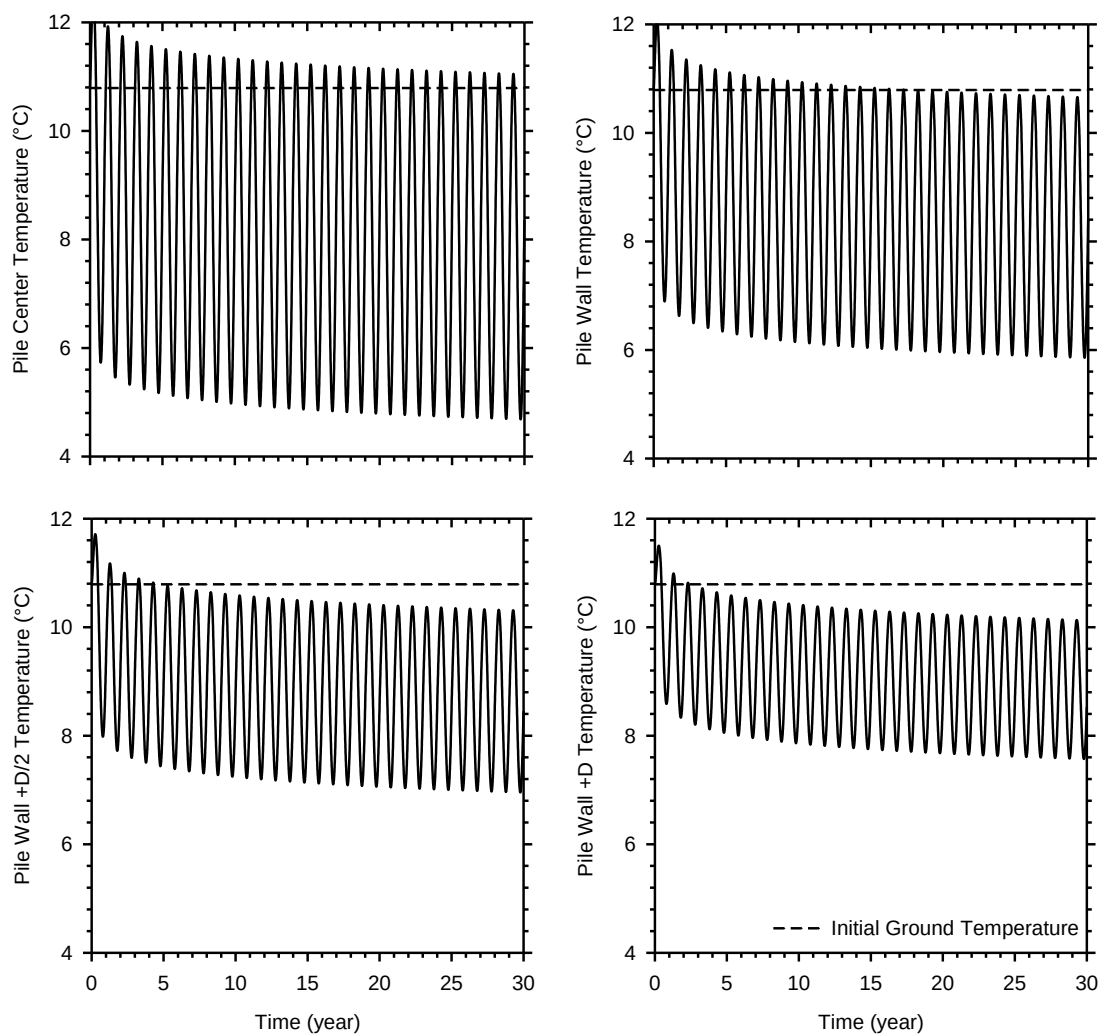


Figure 7.11 : Progression of temperatures in and around the single pile for long-term operations in Chicago, IL.

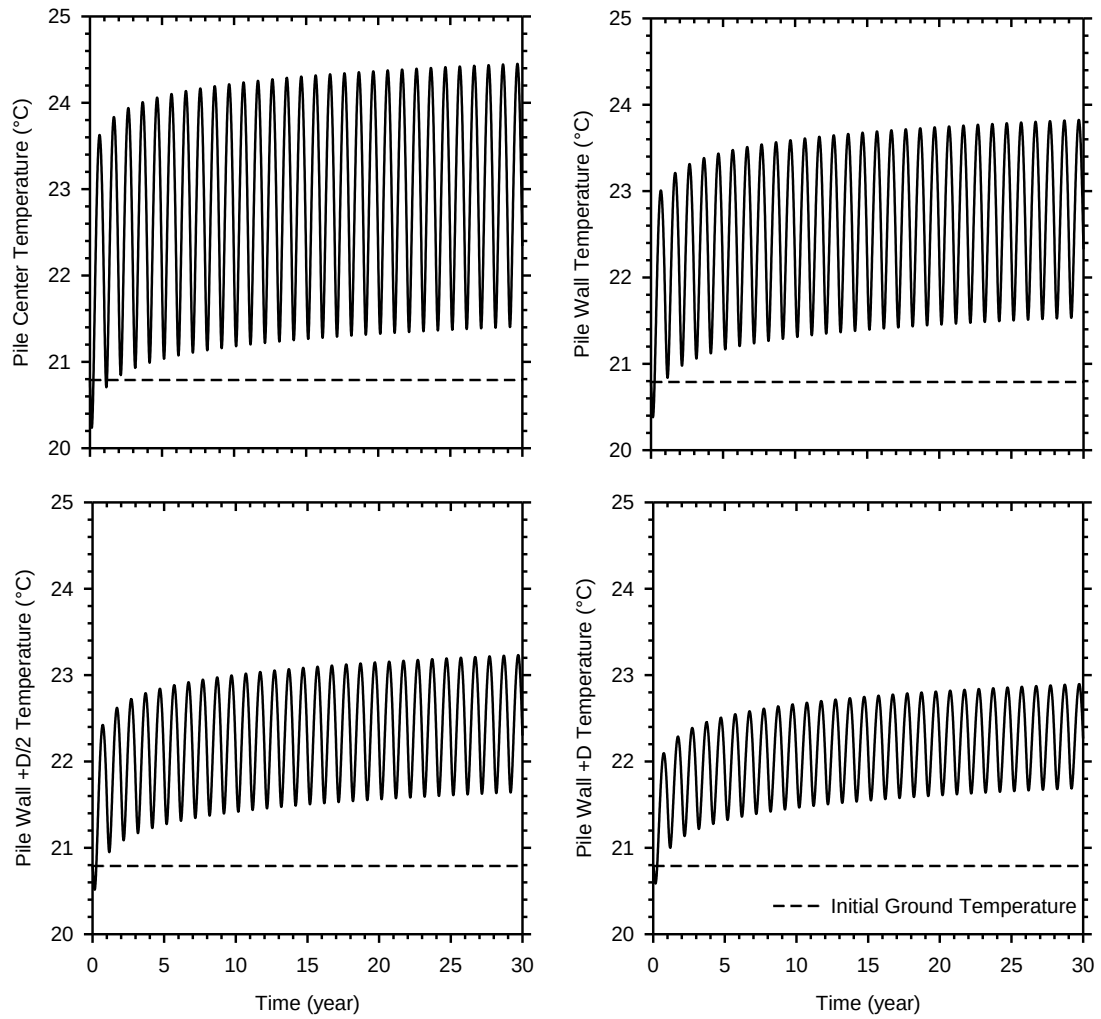


Figure 7.12 : Progression of temperatures in and around the single pile for long-term operations in Austin, TX.

The temperature distributions with radial distance over 30 years are shown in Figure 7.13. It is inferred that the volume influenced by thermal operations extends to a distance of 60 m in each location at the end of 30 years of operation.

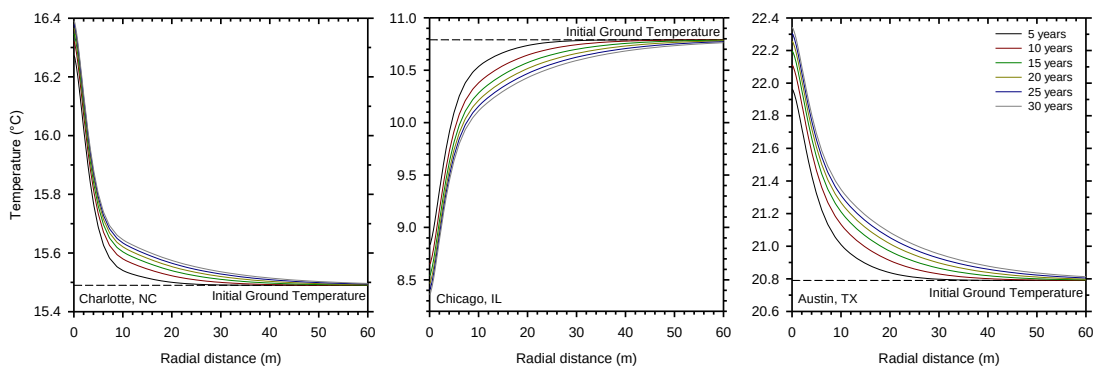


Figure 7.13 : Temperature distributions with radial distance at selected locations during 30 years of operation.

In addition to single pile analyses, numerical simulations were performed for several pile groups in each location. These analyses were implemented in order to investigate the heat exchange behavior and the temperature progressions in the ground using multiple piles with various center-to-center spacings. The results shown here are for the critical pile in each group. The critical pile of a group is specified as the pile that has the greatest temperature change during the heat operation and it is located either at the center of the rectangular grid or it is one of the closest piles to the center of the group.

The progression of the pile center temperatures in each location is given in Figure 7.14. These plots show the temperatures for the single pile and the pile groups where center-to-center spacing is 4 diameters. Each data point indicates the pile temperature at the end of each year, up to 30 years of operation. From this figure, it can be inferred that temperature change is becoming more dominant as the number of piles in the group are increased. For the 5x5 group, after 30 years of operation the temperature of the critical pile rises to 25°C in Charlotte and as high as 45°C in Austin. In Chicago, it is seen that the temperatures drop below zero after 3 years of operation in 3x3 group, and after the first year in 4x4 and 5x5 groups. This makes pile groups larger than 2x2 unfeasible for that location due to freezing of the piles.

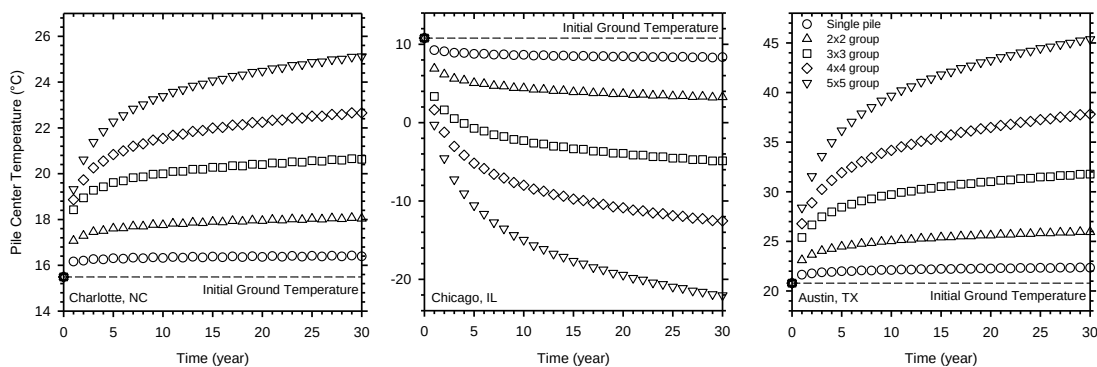


Figure 7.14 : Progression of pile center temperatures at the end of each year for the single pile and pile groups with 4 diameters center-to-center spacing.

The efficiency and sustainability of a heat pump operating in the critical locations (Chicago and Austin) was assessed by investigating the coefficient of performance (COP) and the energy efficiency ratio (EER) over 30 years. COP of a heat pump denotes the ratio of the heat output to the amount of consumed electrical energy. On the other hand, EER is a measure of the cooling efficiency of a heat pump. Higher COP and EER equate to lower operating costs. For the efficiency calculations, a

standard geothermal heat pump (WFI 5 Series 500A11 ND038 with 1050 cfm air flow capacity) was selected and the specifications were provided by the manufacturer (WFI, 2013). Figure 7.15 shows the yearly averaged estimated COP and EER during the operation period for Chicago and Austin, respectively. It is seen that the efficiency of the heat pump declines dramatically for larger pile grids. Moreover, in both locations, the heat pump loses efficiency and becomes inoperable after 11 years for the 4x4 group and after 4 years for the 5x5 group.

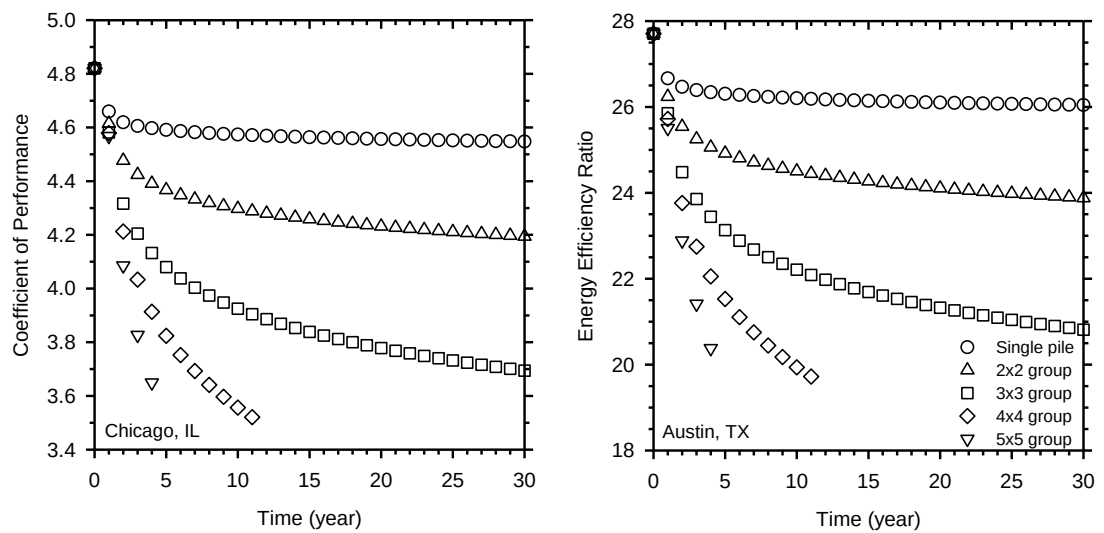


Figure 7.15 : Yearly averaged coefficient of performance (COP) and energy efficiency ratio (EER) for the single pile and pile groups with 4 diameters center-to-center spacing.

Figure 7.16 summarizes the temperature change at the pile center for all cases and locations at the end of 30 years of heat exchange operation. The trends indicate that the effect of pile center-to-center spacing is greater as the pile group is getting larger and it becomes more predominant for 4x4 and larger pile groups.

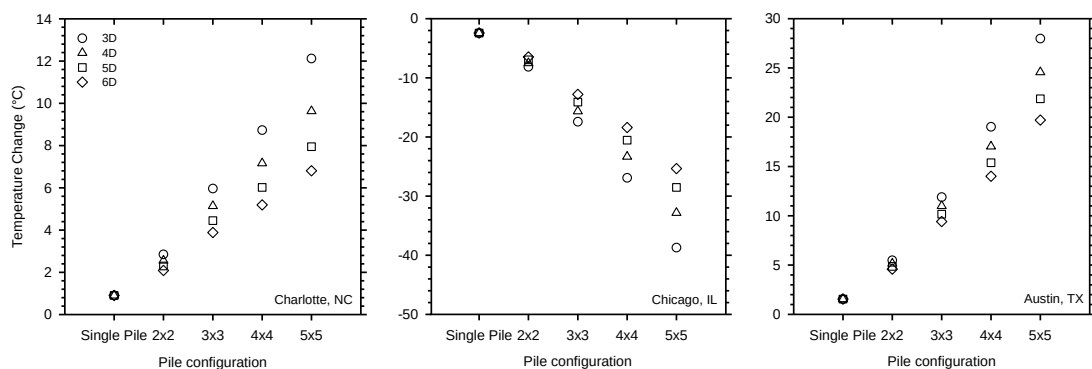


Figure 7.16 : Temperature change for all pile configurations at the end of 30 years of heat exchange operation.

The normalized temperatures with respect to 6 diameters center-to-center pile spacing for each pile group are shown in Figure 7.17. The results suggest that the temperature ratio tends to increase as the piles in a given group are more closely spaced. It can be inferred that for the same pile spacing, this ratio is greater in the largest group. It should be also noted that the trends for Chicago and Austin are similar even though they represent two opposite imbalanced energy demand scenarios. On the other hand, higher temperature ratios are observed in Charlotte, where the energy demand is well balanced. This behavior is a result of having lower temperature changes in such cases (Figure 7.16) and these yield higher ratios as they are normalized.

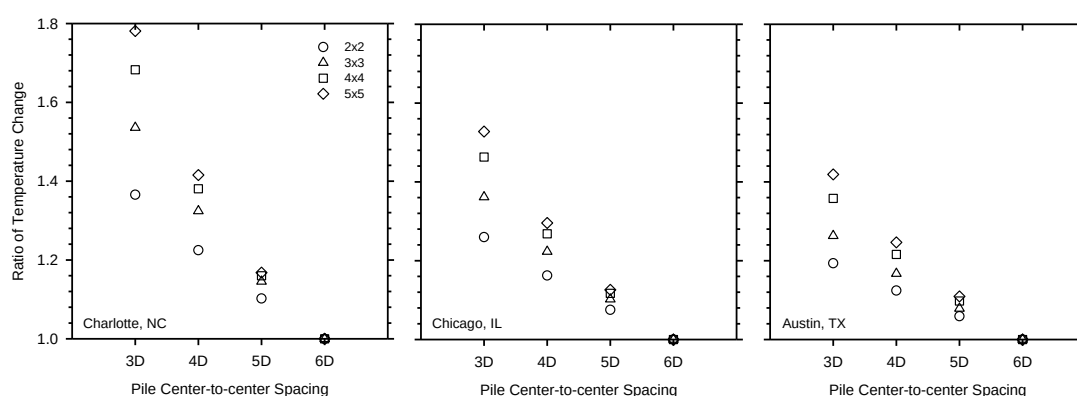


Figure 7.17 : Ratio of temperature change with respect to 6 diameters center-to-center spacing for all pile configurations at the end of 30 years of heat exchange operation.

A final set of analyses were performed in Austin scenario only to illustrate the effect of ground thermal conductivity on the pile temperatures. The base parameter for the ground thermal conductivity was selected to be $2.0 \text{ W/(m}\cdot\text{K)}$ as given in Table 7.1. Analyses were performed for two more cases where the ground thermal conductivity is $1.5 \text{ W/(m}\cdot\text{K)}$ and $2.5 \text{ W/(m}\cdot\text{K)}$. The results for the single pile and the pile groups with 4 diameters of center-to-center spacing is presented in Figure 7.18. It is seen that the effect of ground thermal conductivity is minimal for the field arrangement with single pile as the difference between two extreme cases are less than 1°C . On the contrary, the effect is more predominant for the pile groups. In 5x5 group, there is a 10°C difference between the pile temperatures for the minimum and maximum ground thermal conductivity cases.

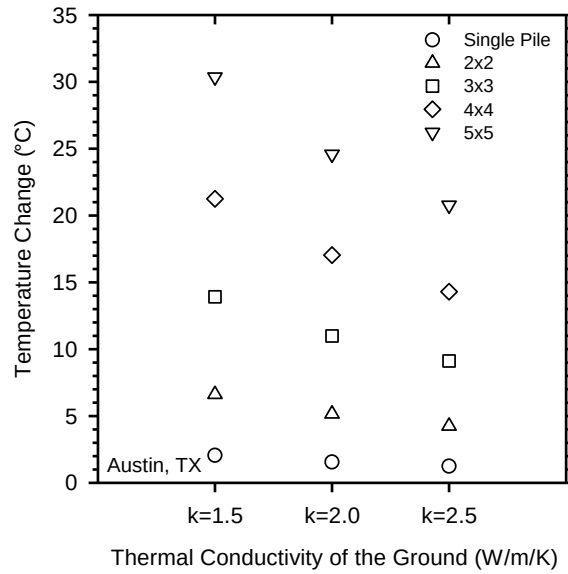


Figure 7.18 : The effect of ground thermal conductivity on the pile temperatures for the single pile and pile groups with 4 diameters center-to-center spacing at the end of 30 years of heat exchange operation in Austin, TX.

These results indicate that the seasonal balance of energy demand during heat exchange operations can be critical for progression of temperatures around an heat exchanger pile. Any significant changes to the initially stable ground temperatures can result in the system to lose efficiency after several years of operation. It is also quite noteworthy to indicate that soil compressibility and strength can also be affected with temperature changes and therefore can affect the capacity of the heat exchanger pile and/or result in unanticipated pile head movements. This study underlines the potential for such temperature changes as a result of long-term operations. However any changes to soil strength/compressibility due to these temperature changes are beyond the scope of this study.

7.4 Numerical Simulation of Long-Term Thermo-Mechanical Behavior

A series of coupled thermo-mechanical numerical analyses were performed for each site in which only the single pile case was considered. Thermo-mechanical model was built using COMSOL Multiphysics™, by utilizing the readily available modules in time dependent and axisymmetric conditions. The dimensions of the model are shown in Figure 7.19. The soil is free of stress at the top of the domain and fully fixed at the bottom. In the far field, geostatic stress conditions are assumed with a roller boundary condition on the right hand side boundary. The left hand side boundary represents the

symmetry axis of the domain. The heat exchanger pile is modeled as a thermo-elastic concrete cylinder embedded in an elasto-plastic soil mass governed by Mohr-Coulomb yield criterion. Interface elements are used at the pile-soil boundaries: along the side of the pile shaft and at the base of the pile. The normal and tangential contact law at the interface is linear elastic. The governing equations of the thermo-elastic model, Mohr-Coulomb elasto-plastic model and the interface contact law are given in Equation (7.4), Equation (7.5) and Equation (7.6), respectively.

$$\begin{bmatrix} \sigma_r \\ \sigma_\phi \\ \sigma_z \\ \tau_{rz} \end{bmatrix} = \frac{E}{(1-2\nu)(1+\nu)} \begin{bmatrix} 1-\nu & \nu & \nu & 0 \\ \nu & 1-\nu & \nu & 0 \\ \nu & \nu & 1-\nu & 0 \\ 0 & 0 & 0 & G \end{bmatrix} \begin{bmatrix} \varepsilon_r \\ \varepsilon_\phi \\ \varepsilon_z \\ \gamma_{rz} \end{bmatrix} - \frac{E}{(1-2\nu)} \alpha \Delta T \begin{bmatrix} 1 \\ 1 \\ 1 \\ 0 \end{bmatrix} \quad (7.4)$$

$$|\tau| + \sigma \tan \phi - c = 0 \quad (7.5)$$

$$\sigma \cdot \mathbf{n} = -\mathbf{k}_s (\mathbf{u}_{pile} - \mathbf{u}_{soil}) \quad (7.6)$$

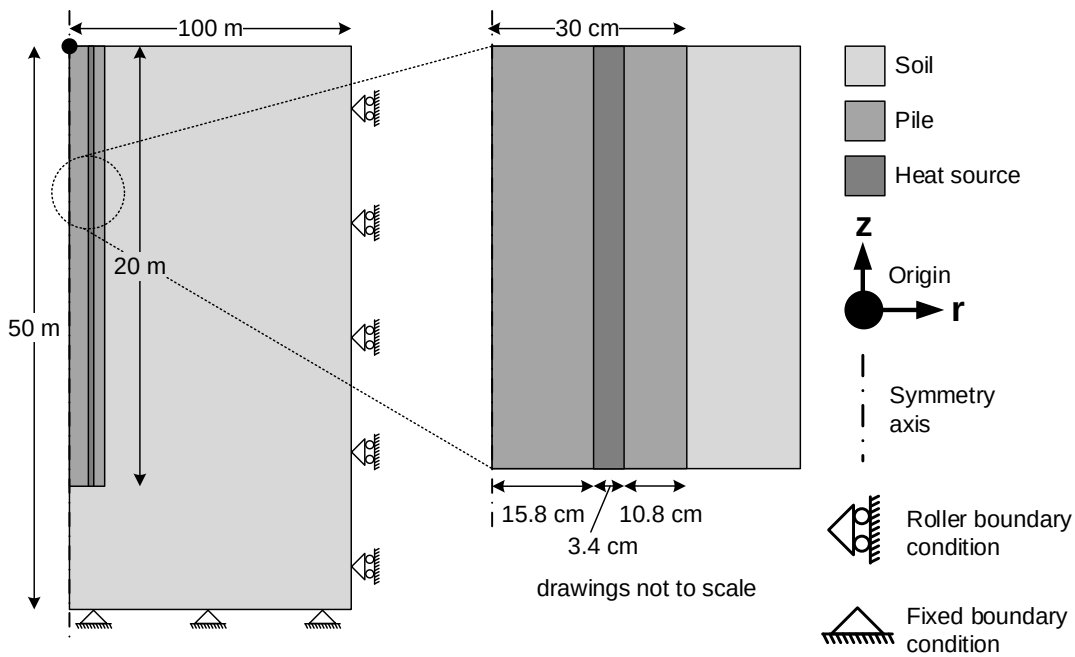


Figure 7.19 : Thermo-mechanical model geometry and dimensions.

The soil profile was selected to be isotropic and homogenous. Even though it is not a realistic case, this assumption was made for the sake of simplicity and in order to match the thermal analyses. The circulating tubes embedded in the heat exchanger pile were approximated by an internal domain, which has a width equal to the diameter of the circulating tube. The thermal loading was simulated by applying the sine wave

approximated energy demand curve as a time dependent heat source on this domain (Figure 7.19). The mesh shown in Figure 7.20 consists of 14,318 triangular domain elements and 1,823 boundary elements. Thermo-mechanical model shares the same thermal properties for the pile and the ground previously given in Table 7.1. Additional input parameters related to the mechanical processes are listed in Table 7.2.

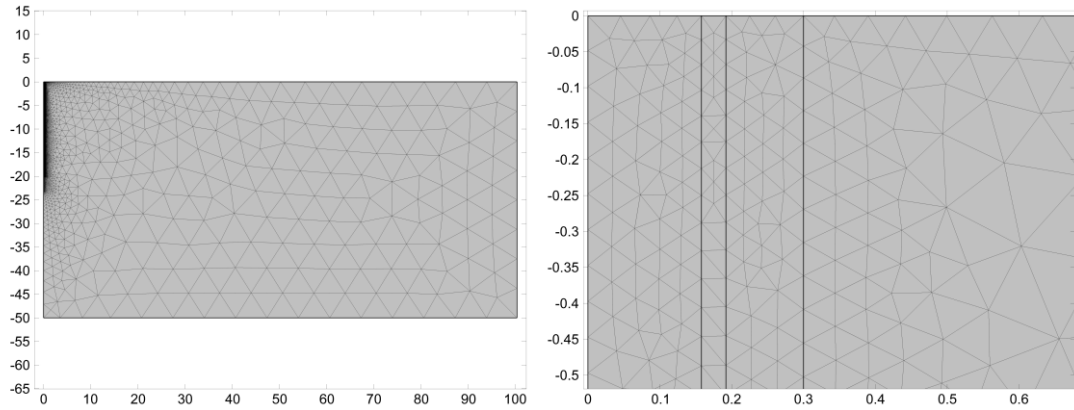


Figure 7.20 : Mesh adopted in the thermo-mechanical finite element model.

Table 7.2 : Additional parameters used in the thermo-mechanical analyses.

Parameter	Value	Unit
Pile		
Young's modulus	17,000	MPa
Poisson's ratio	0.15	—
Coefficient of thermal expansion	1e-5	K ⁻¹
Ground		
Young's modulus	12.5	MPa
Poisson's ratio	0.495	—
Coefficient of thermal expansion	5e-5	K ⁻¹
Cohesion	25	kPa
Friction angle	0	°
Interface at the pile shaft		
Tangential stiffness	2.5	MPa
Normal stiffness	25	MPa
Interface at the pile base		
Tangential stiffness	25	MPa
Normal stiffness	25	MPa

The pile was mechanically loaded to failure in a separate analysis. The ultimate bearing capacity of the pile is found to be 1030 kN corresponding to a head displacement/diameter ratio of 4% as indicated in Figure 7.21. For the thermo-mechanical analyses, prior to applying the seasonal thermal loads, the heat exchanger pile was mechanically loaded to 515 kN (factor of safety=2) using a ramp function. The mechanical loading phase starts at 6 h and reaches the final load at 12 h. Thermal

loading phase begins at the end of the first day. The load-time histories of the mechanical and the thermal loads are given in Figure 7.22. The results of the long-term thermo-mechanical analyses for the single pile are shown in Figure 7.23, Figure 7.24 and Figure 7.25.

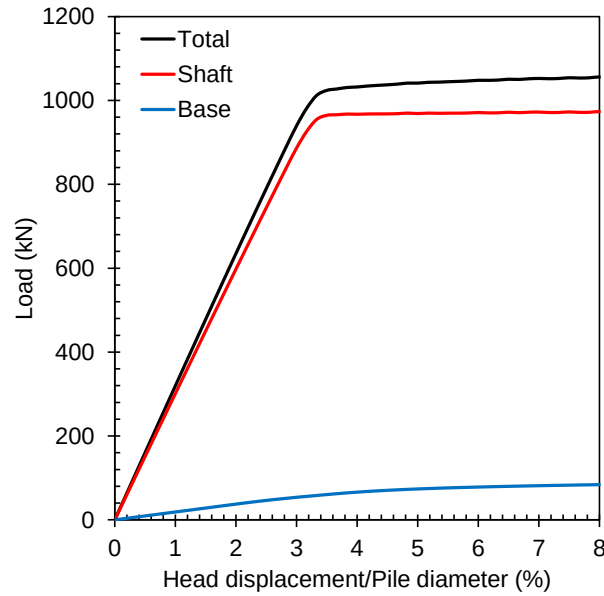


Figure 7.21 : Results of the pile load test.

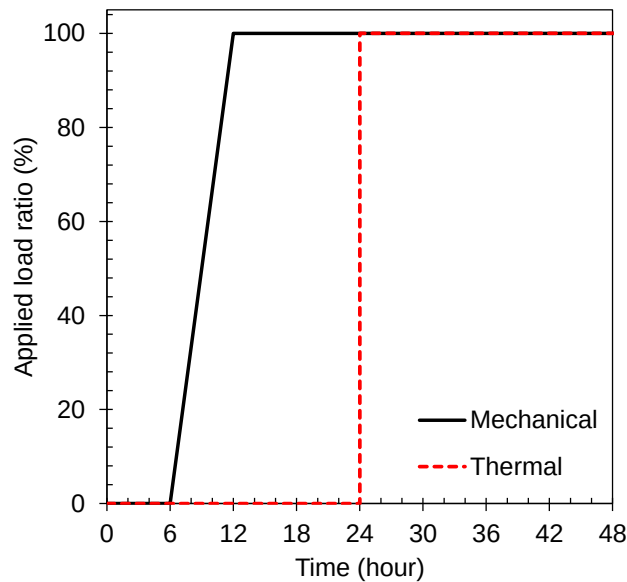


Figure 7.22 : Load-time histories of mechanical and thermal loads.

The progression of pile head displacement for the three selected locations is presented in Figure 7.23. The initial settlement of the pile head is 9.7 mm which is obtained after loading the pile to the service load with a factor of safety of 2. It can be inferred that the displacements follow the progression of pile temperatures (Figure 7.10, Figure 7.11

and Figure 7.12) in each location. Pure elastic elongation of the pile is observed over 30 years of operation and there is an accumulation of displacements due to the thermo-mechanical effect at the end of 30 years. In Charlotte and Austin, at the end of 30 years of operation, the pile dilates with relative head displacements of 0.3 mm and 0.9 mm, respectively. On the other hand, due to progressively cooling of the ground, the pile in Chicago shows contractive behavior and the final relative head displacement is 1.3 mm.

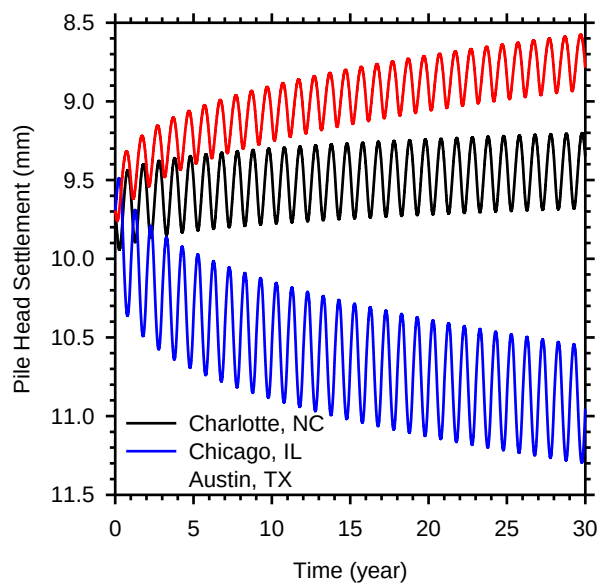


Figure 7.23 : Progression of the pile head settlement at selected locations during 30 years of operation.

Figure 7.24 shows the axial stresses at the mid-depth of the pile ($z=10\text{m}$) over 30 years of operation for each site. It is inferred that the stresses in the pile increase as the pile is heated and decrease as the pile is cooled during seasonal operation. This behavior agrees with the temperature induced pile response concept presented by Bourne-Webb et al. (2009) (Figure 7.3). On the contrary, the accumulation of axial stresses over the long-term suggests the opposite behavior. For example, in Chicago even the pile is progressively cooled, the axial stresses tend to increase over the long-term operation. Similarly, decrease in axial stresses is observed in Austin where the thermal loads are heating dominated. This phenomenon can be explained by the difference in the thermal expansion coefficients of the pile and the soil. The envelopes of axial stress (minimum and maximum stress levels) along the pile over 30 years of operation are presented in Figure 7.25. The black solid line represents the stresses induced by mechanical loading

only. It is seen that the maximum change in pile axial stress over 30 years amounts to 75 kPa in Chicago, while in Charlotte and Austin a change of 40 kPa is observed.

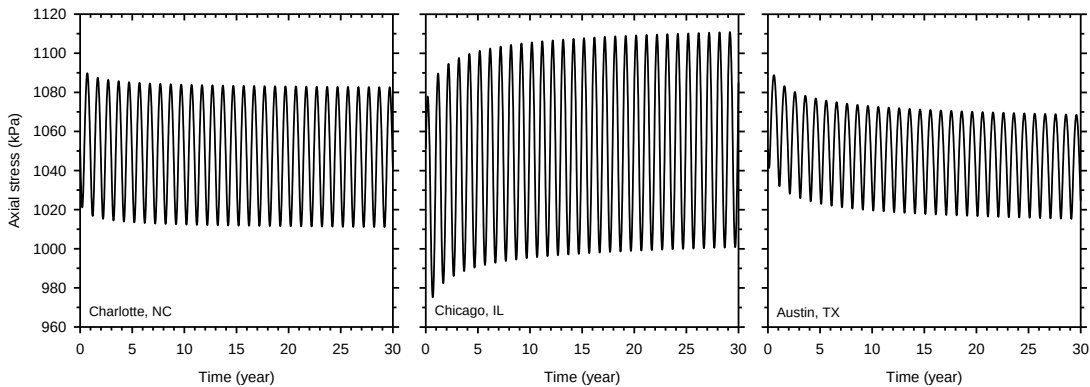


Figure 7.24 : Progression of the pile axial stress at mid-depth at selected locations during 30 years of operation.

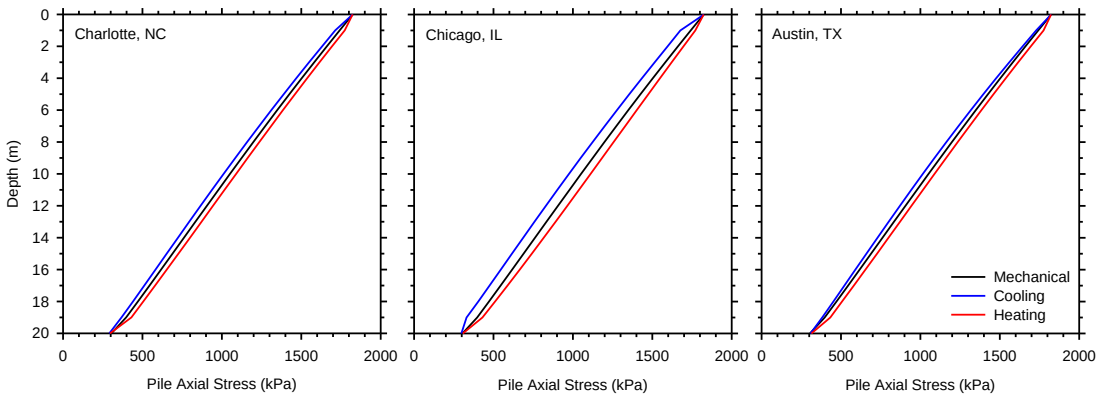


Figure 7.25 : Axial stress envelopes along the pile (mechanical, cooling and heating) at selected locations.

8. CONCLUSIONS

There is a developing trend around the world to explore alternative energy sources. The main driving forces are growing global energy demand, depleting natural resources and the adverse effects of greenhouse gas emissions from fossil fuel consumption. From the beginning of the 21st century, energy consumption has been increasing due to an increasing demand for improving the quality of life. Countries, governments and individuals need to find ways of reducing the carbon footprint and one such option is the utilization of alternative, renewable energy sources. Increased use of renewable energy is required in the coming decades to contribute to a reduction in global energy use and also a reduction in carbon dioxide emissions.

Energy for heating and cooling purposes accounts for 30% of a country's energy consumption (Moon and Choi, 2015). Therefore, there is the need for technologies to produce highly efficient and environmental friendly heating and cooling systems (Miyara et al., 2012; Xu et al., 2012). Geothermal energy is one of the promising renewable sources that can be utilized to offset such trends. Yet, the use of traditional geothermal energy systems require interaction with kilometre-deep strata of rock, where thermal energy is much greater and can produce hot fluids to be used either for district heating and/or electricity generation. More recently, encouraging developments are being achieved in the field of shallow geothermal energy systems with the use of geothermal boreholes. These systems show great potential, comparative to the traditional systems, in terms of long-term sustainability, access, flexibility and economics. They exemplify a highly efficient renewable energy technology for space heating and cooling (Sanner et al., 2003; Omer, 2008; Lund et al., 2011). Shallow geothermal energy is based on the principle that the subsoil can be employed as a thermal energy source by using its natural potential and thermal storage capabilities (Michopoulos et al., 2007). The key parameters in the design and dimensioning of the ground-source heating and cooling systems are the thermal characteristics and the heat exchange capacity of the in-situ soil and rock formations. The magnitude and rate of thermal energy that can be harvested with a GHE depends mainly on the thermal

properties of the ground, in particular thermal conductivity (Wagner and Clauser, 2005). Proper design and sizing of GHE systems for space heating/cooling requires a good estimate of the ground thermal properties.

Over the past 20 years, the ground coupling concept has been expanded from geothermal borehole systems to the use of building foundation elements as heat exchangers. Heat exchanger pile (also mentioned as energy pile) in particular is an innovative technology that combines geothermal heat exchange and structural foundation support. In this hybrid system, geothermal circulation loops are integrated into the deep foundation elements, such as piles, piers, or drilled shafts that are already designed to provide structural support. The heat energy is fed into the ground for cooling in the summer and withdrawn from the ground for heating in the winter. The temperature differential between the ground and the outside temperature acts as an energy pathway to harvest stored ground energy for bringing the building temperature to comfort zone levels. As an added benefit, the additional cost of geothermal borehole drilling for loop placement is offset by this combined use and installation costs of geothermal heat exchangers are significantly reduced.

Heat exchanger piles possess a twofold technological character that has drawn a dual related scientific interest in their behavior. In fact, the energy performance of the energy piles can markedly vary for different (1) site layouts, (2) foundation geometries, (3) pipe configurations, and (4) soil and foundation material properties. In addition, the geotechnical behavior of heat exchanger piles can strongly vary for different (5) restraint conditions and (6) applied thermal loads. Consequently, these two fundamental aspects of the energy piles are interconnected and coupled through the thermal and mechanical responses of these foundations. Because of this twofold character, energy piles are repeatedly subjected to thermo-mechanical loads that involve fundamental changes of the foundation and soil behavior, as well as of their interaction.

Currently, the above mentioned phenomena have not been studied exhaustively and there is a lack of understanding in the long-term thermo-mechanical behavior of heat exchanger piles and thermal operation of heat exchanger pile groups. Therefore, the application of such foundations for the support of new structures and infrastructures is still restrained. Looking at such challenge, the main objectives of this dissertation were (1) to estimate the thermal properties of the ground accurately and efficiently, (2) to

assess the thermo-mechanical behavior of heat exchanger piles and (3) to predict the long-term performance of heat exchanger piles and pile groups. For this purpose, several numerical modeling tools were developed and utilized for the design and analysis of geothermal heat exchangers. The contributions and findings of the study are presented below:

- Three types of finite element numerical modeling techniques with different complexities (2D, 3D, and 1D-3D Hybrid) were developed in order to simulate the thermal operations of geothermal heat exchangers. The models were successfully validated against field test data and finite line source analytical method. The analysis results of the numerical and analytical models were compared by the average fluid temperatures, temperature profiles at several radial distance-time combinations, and temperature versus radial distance at mid-depth of the heat exchanger at various times. The numerical model is verified and the results suggest that it can successfully simulate the operation of vertical geothermal heat exchangers.
- Further modifications to the 3D and the 1D-3D Hybrid models were provided in order to incorporate initial ground temperature as a geothermal gradient and ambient air temperature variations as a boundary condition at the ground surface. The 1D-3D Hybrid Model was utilized in several case studies confirming that it can be used as a robust design tool for heat exchanger piles.
- A 2D axisymmetric finite element numerical model was developed to analyze the thermo-mechanical behavior of heat exchanger piles. The model utilizes interface elements at the pile-soil boundary to model the soil-pile interaction. The finite element model was validated using the measurements from a field test performed at Lambeth College in South London (Amatya et al., 2012). It was inferred that the finite element modeling approach works well in simulating thermo-mechanical behavior of a heat exchanger pile under temperature changes. A parametric study was performed in stationary conditions with respect to various soil/interface stiffnesses and mechanical constraints (e.g. free pile head, fixed pile head and/or fixed pile toe). The soil-pile interaction is directly influenced by the interplay between the thermal strains in the pile and the associated shear stresses that develop along the soil-pile interface. The constrained vertical elongation is the most detrimental factor

for the performance of the pile foundation. Pure interface effects induce internal stresses about ten times smaller than the stresses observed at pile ends for the case in which mechanical constraint is present. It is also worth noticing that while mechanical constraints require oversizing heat exchanger piles at the ends, interface effects require additional reinforcement for the pile at mid-depth. Interface deformations are relatively small indicating that these are likely to remain elastic and reversible in response to heating and cooling cycles. It was concluded that it may not be critical to account for pile friction in dimensioning heat exchanger piles as structural constraint can play a more significant role.

- Using the above mentioned thermo-mechanical model, long-term (30 years) thermo-mechanical analyses were performed using actual energy demands on a mechanically loaded heat exchanger pile in locations with extreme climatic conditions, i.e., hot, moderate and cold. The pile showed pure elastic elongation over the operation period and the considered long-term energy operations do not have much influence on the geotechnical performance. The findings have implications for long-term heat exchange efficiency of heat exchanger piles and can also have potential effects on soil behavior around the pile.
- A 2D horizontal cross-section finite element model was developed to quantitatively assess the radial stresses and strains undergone by a heated pile embedded in deformable soil. A parametric study with respect to various soil stiffnesses, which consists of plane stress and plane strain analyses were performed to assess the effect of thermally induced lateral stress increase as a possible mechanism that can affect shaft resistance and load-displacement behavior of heat exchanger piles. The increase of contact pressure induced by the radial thermal expansion of the pile is small in magnitude (less than 15 kPa) and therefore would not result in significant increase in shaft resistance. It is rather unlikely that temperature induced radial expansion of the pile would increase pile capacity significantly, due to the minimal change in contact pressure at the pile-soil interface.
- Finally, a series of energy demand analyses coupled with finite element analyses were performed to address the long-term performance of heat exchanger piles within a sustainable framework. A parametric study was

performed to assess the long-term performance of heat exchanger pile groups of several pile arrangements ranging from single pile to numerous pile groups with a selection of 2x2, 3x3, 4x4 and 5x5 rectangular grids and their efficiency for areas where the energy demand is nonsymmetrical. It was attained that the nature and degree of temperature progression around a heat exchanger pile is directly related to the seasonal energy demand and sustainable heat exchange operations are linked to the seasonal energy demand at different environments. This study underlines the fact that the heat exchange operations can lose efficiency over time unless preventive measures are taken. These could include a recharging approach to seasonally balance the ground temperatures or the development of sustainable approaches to balance the energy demands of different structures within an urban setting.

The numerical modeling tools developed in this dissertation can be further improved by conducting further studies. Recommendations for future research are presented below:

- The application of the proposed numerical models can be further expanded to simulate underground thermal energy storage. This study would combine solar thermal technology with geothermal energy. For example, during the hot season, heat is collected by solar thermal collectors and injected into the ground using geothermal heat exchangers. The stored energy can later be utilized for applications such as bridge deck deicing, when needed.
- The proposed numerical models can be modified in order to take into account the effect of groundwater flow on the performance of geothermal heat exchangers. This requires developing thermo-hydro-mechanical (THM) coupled models, which will have increased complexity. On the other hand, such models would provide estimation of changes in soil strength and compressibility due to temperature changes, allowing to simulate applications such as thermally induced consolidation.

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APPENDICES

APPENDIX A: Thermal conductivity testing systems around the world

APPENDIX B: Analysis results of the field thermal conductivity tests at Berkel Site

APPENDIX A: Thermal conductivity testing systems around the world



Figure A.1 : Swedish thermal response test rig (TED). Photo: Peter Olsson (Eklof and Gehlin, 1998).



Figure A.2 : Canadian thermal response test rig. Photo: Environment Canada.



Figure A.3 : German thermal response test rig (UBeG). Photo: UBeG GbR, Wetzlar.



Figure A.4 : German thermal response test rig (Landtechnik Weihestephane). Photo: Landtechnik Weihestephane.



Figure A.5 : Dutch thermal response test rig with heating and cooling mode. Photo: Groenholland.



Figure A.6 : United Kingdom thermal response test rig. Photo: Geoscience.



Figure A.7 : Oklahoma State University thermal response test rig. Photo: Jeffrey Spitler.



Figure A.8 : Suitcase thermal response test set-up of Ewbanks, USA. Photo: Signhild Gehlin.



Figure A.9 : Suitcase thermal response test unit of TriSun Construction, Oklahoma.
Photo: Jeffrey Spitler.



Figure A.10 : Thermal response test unit of Precision Geothermal, USA. Photo:
Precision Geothermal.



Figure A.11 : Thermal response test unit of EPFL, Switzerland. Photo: EPFL.



Figure A.12 : Thermal response test unit used in Bulgaria (Georgiev et al., 2009).

APPENDIX B: Analysis results of the field thermal conductivity tests at Berkel Site

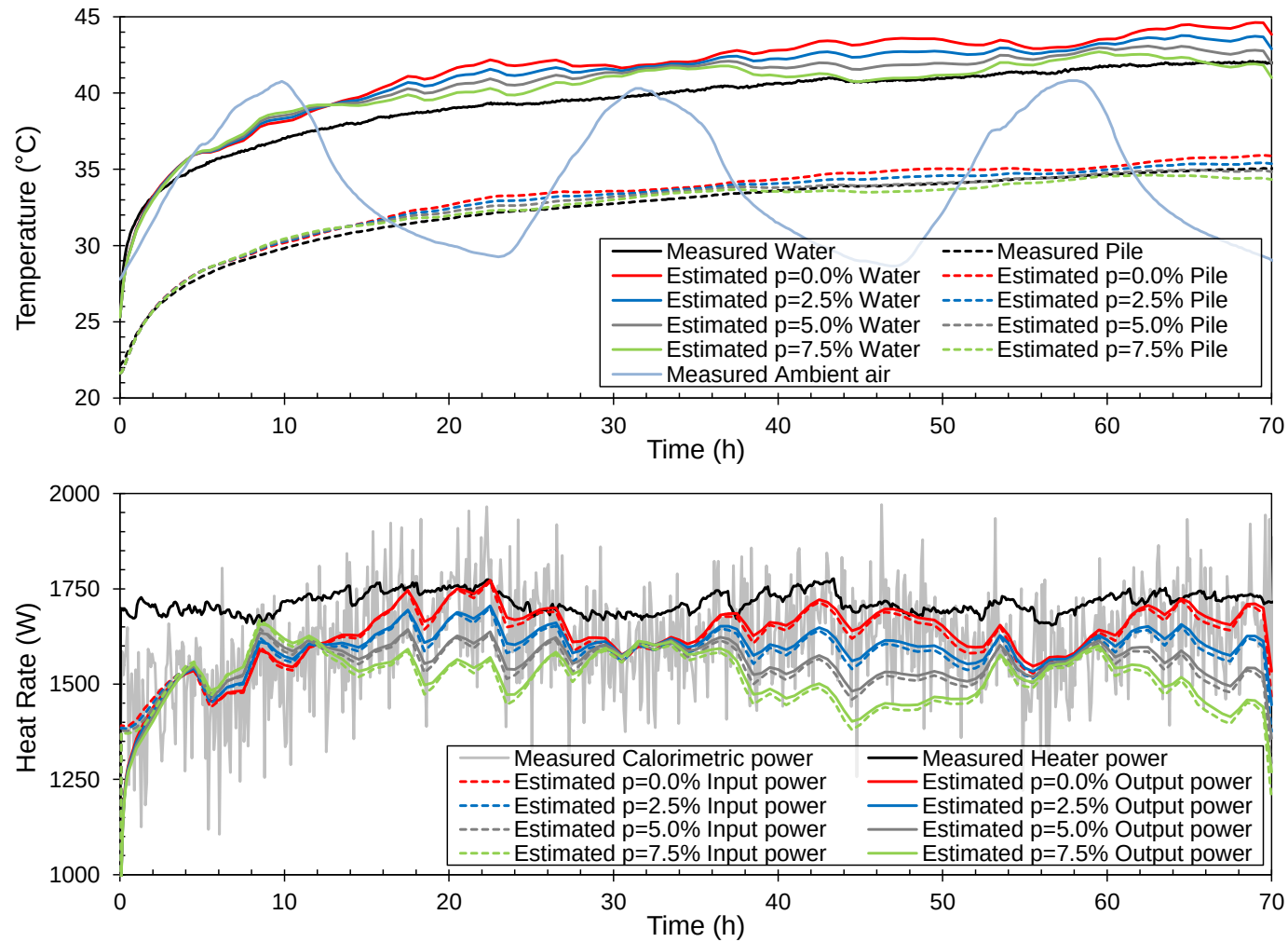


Figure B.1 : Average temperature and heat rate plots of all models for Test #1.

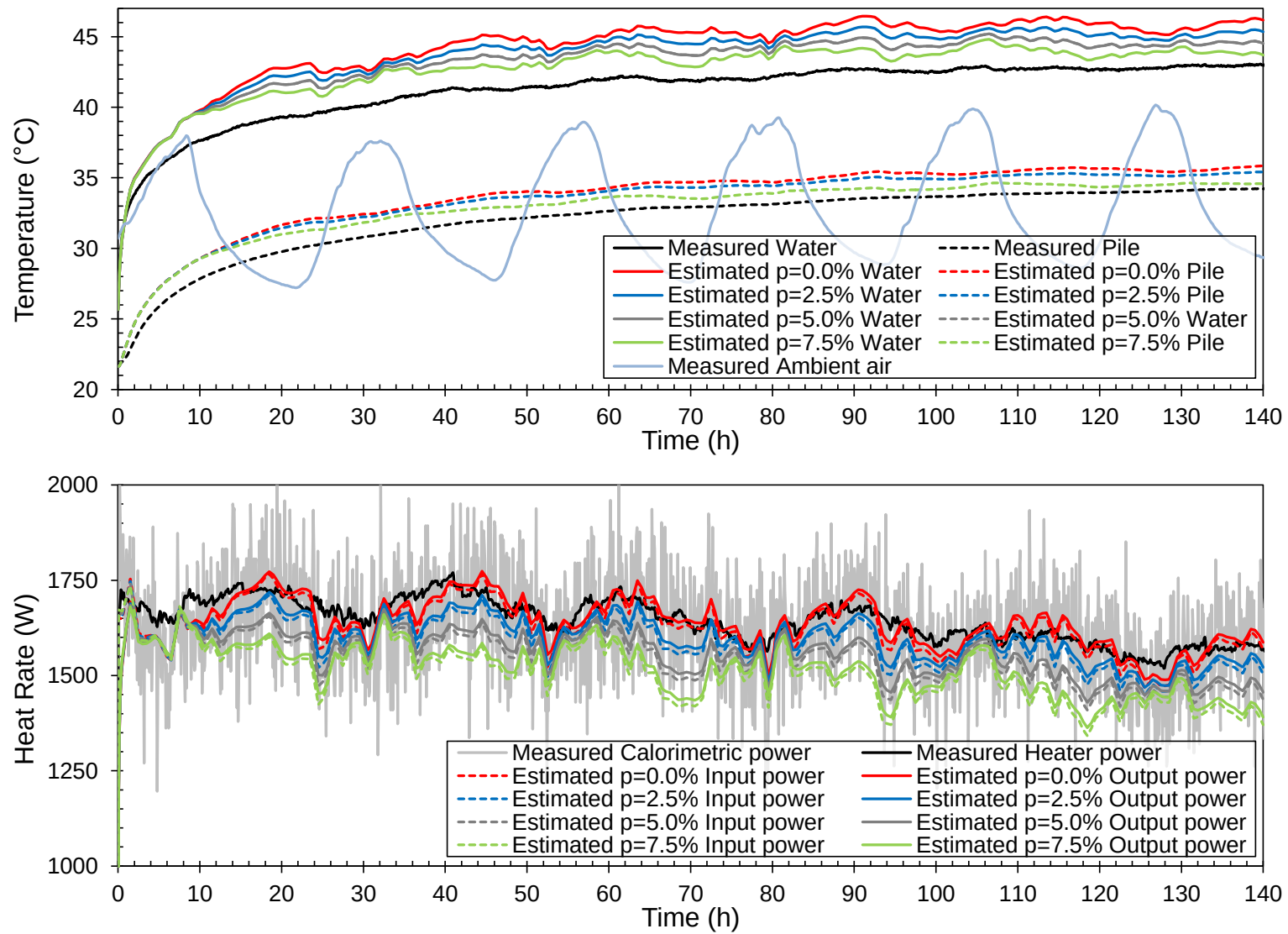


Figure B.2 : Average temperature and heat rate plots of all models for Test #2.

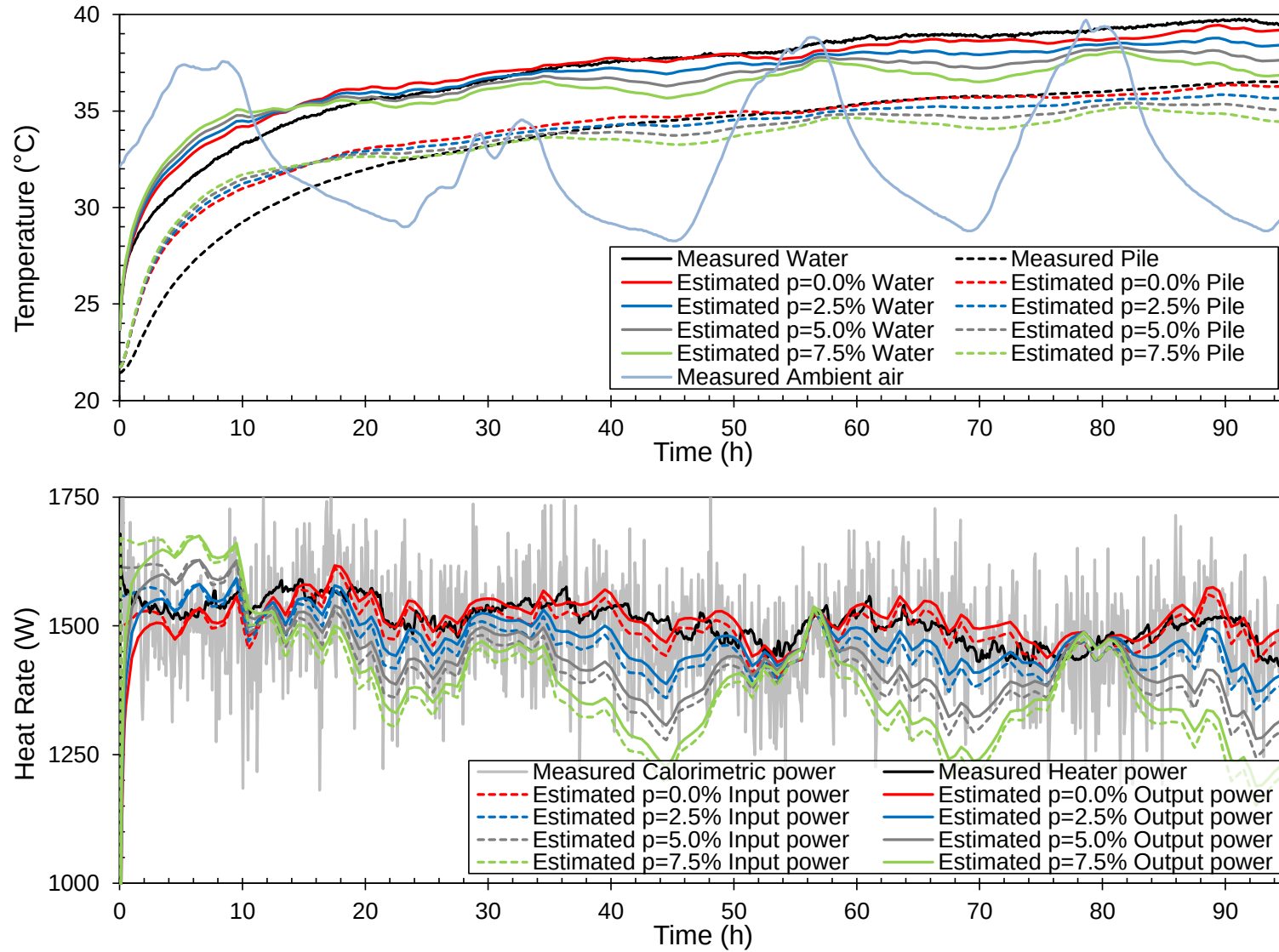


Figure B.3 : Average temperature and heat rate plots of all models for Test #3.

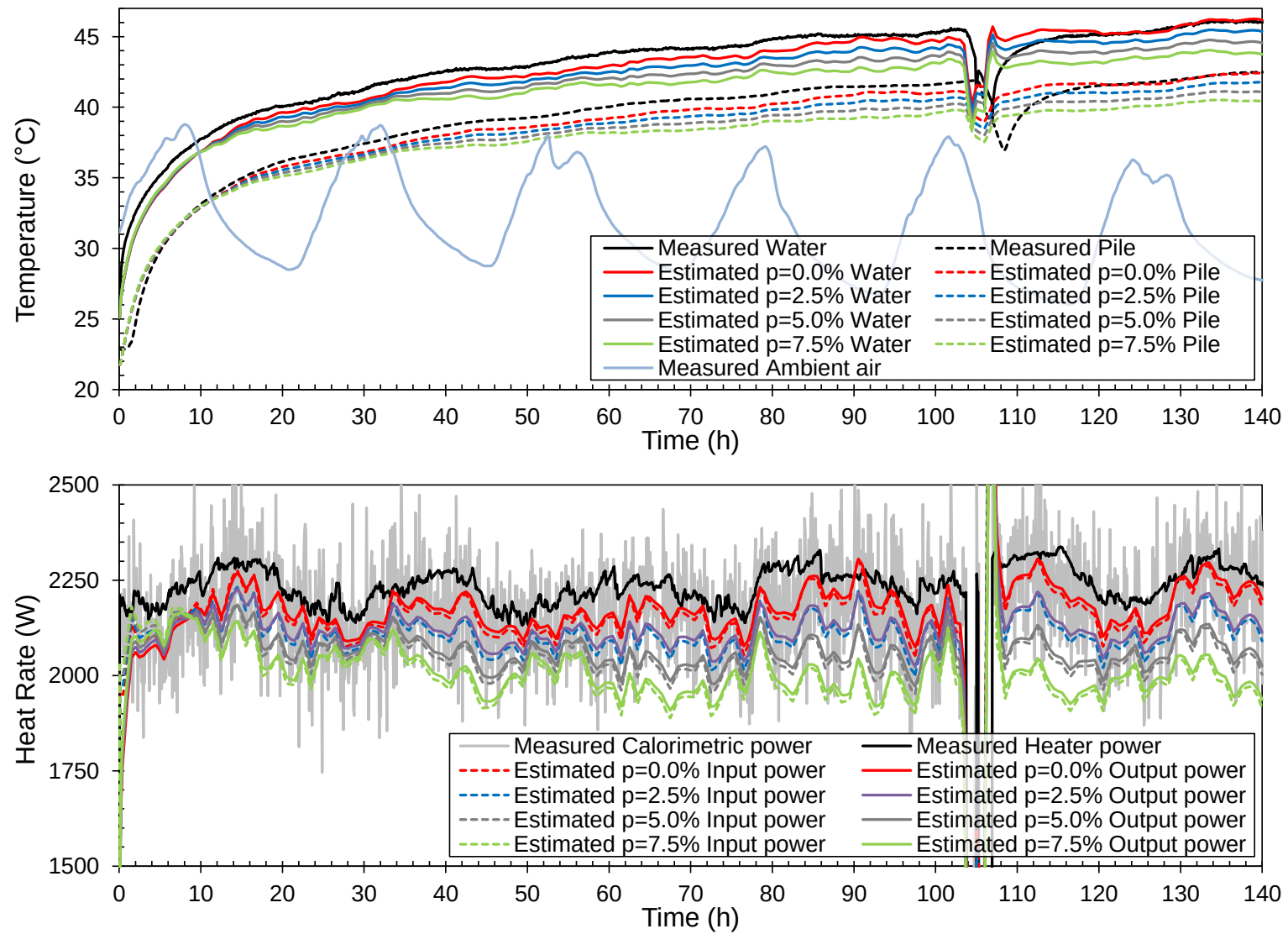


Figure B.4 : Average temperature and heat rate plots of all models for Test #4.

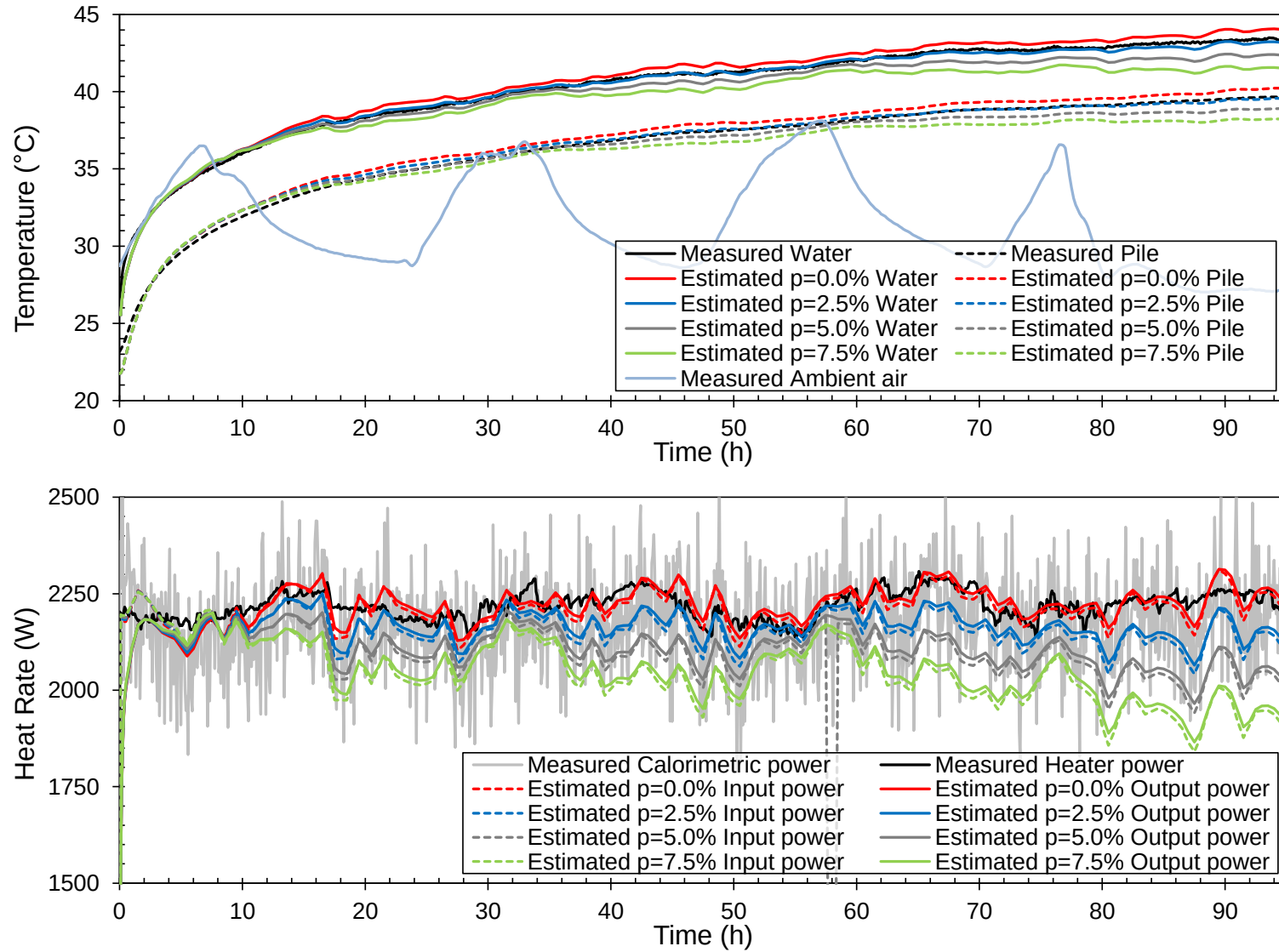


Figure B.5 : Average temperature and heat rate plots of all models for Test #5.

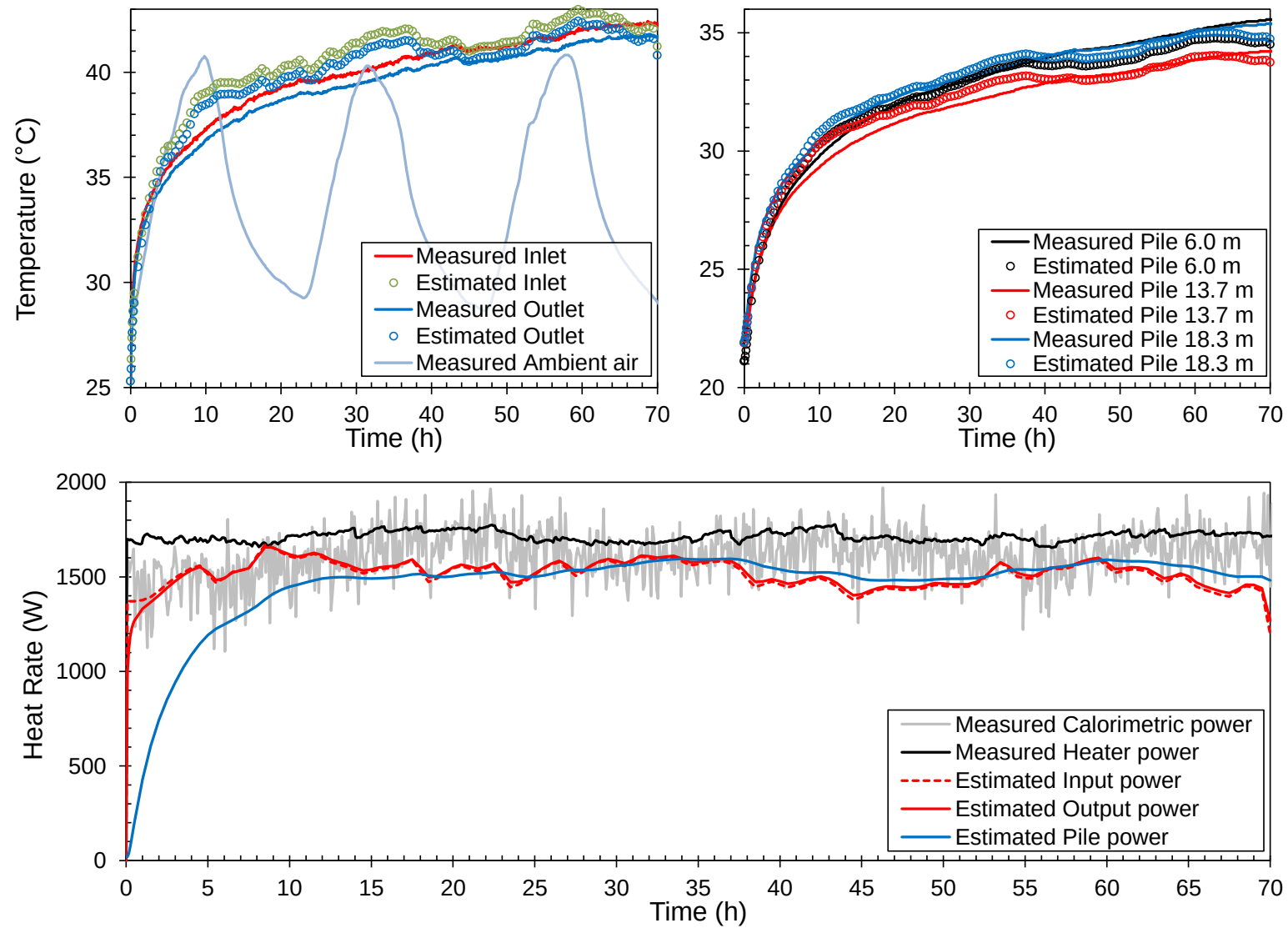


Figure B.6 : Temperature and heat rate plots of best fit analysis for Test #1 ($p = 7.5\%$).

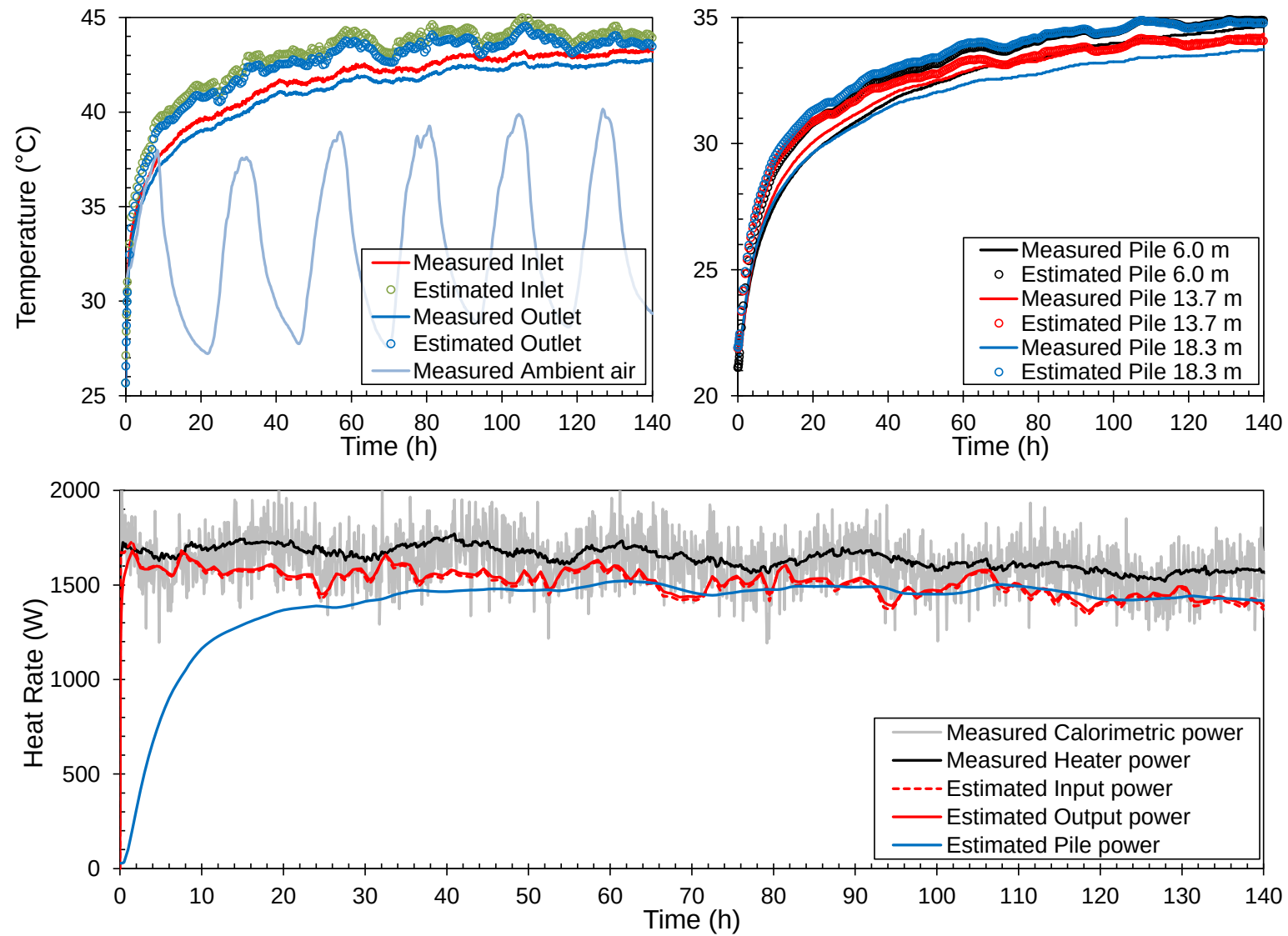


Figure B.7 : Temperature and heat rate plots of best fit analysis for Test #2 ($p = 7.5\%$).

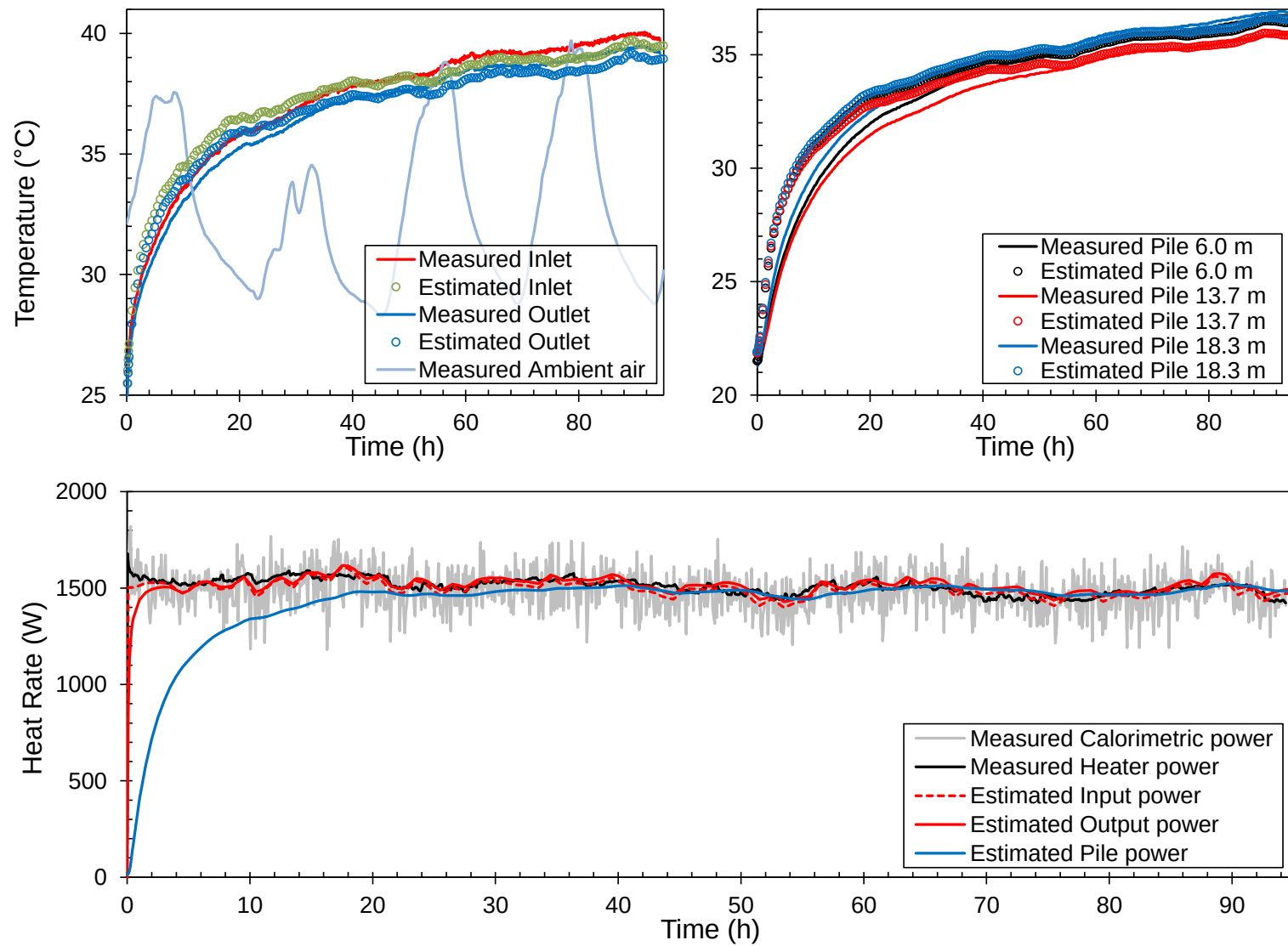


Figure B.8 : Temperature and heat rate plots of best fit analysis for Test #3 ($p = 7.5\%$).

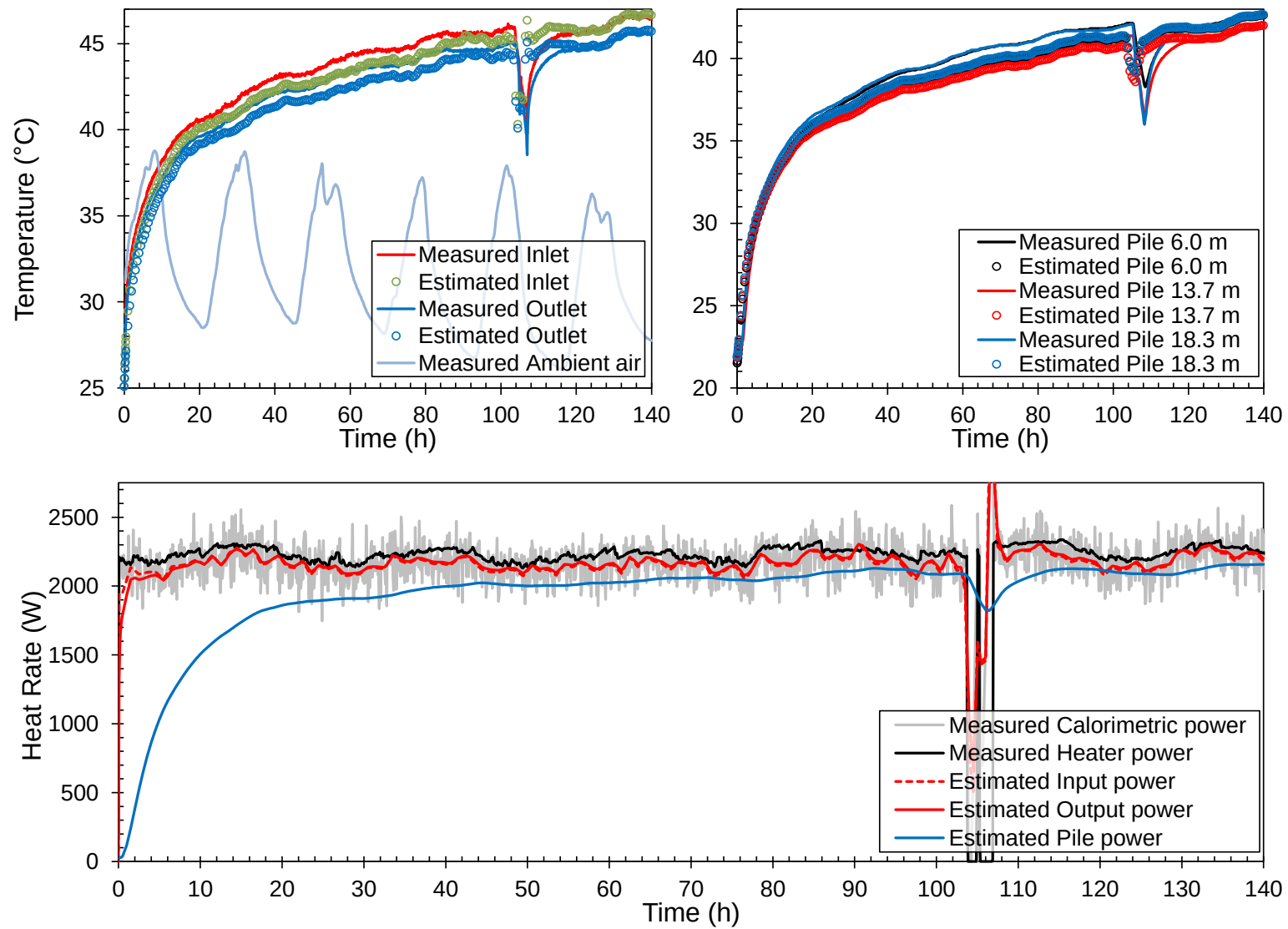


Figure B.9 : Temperature and heat rate plots of best fit analysis for Test #4 ($p = 0.0\%$).

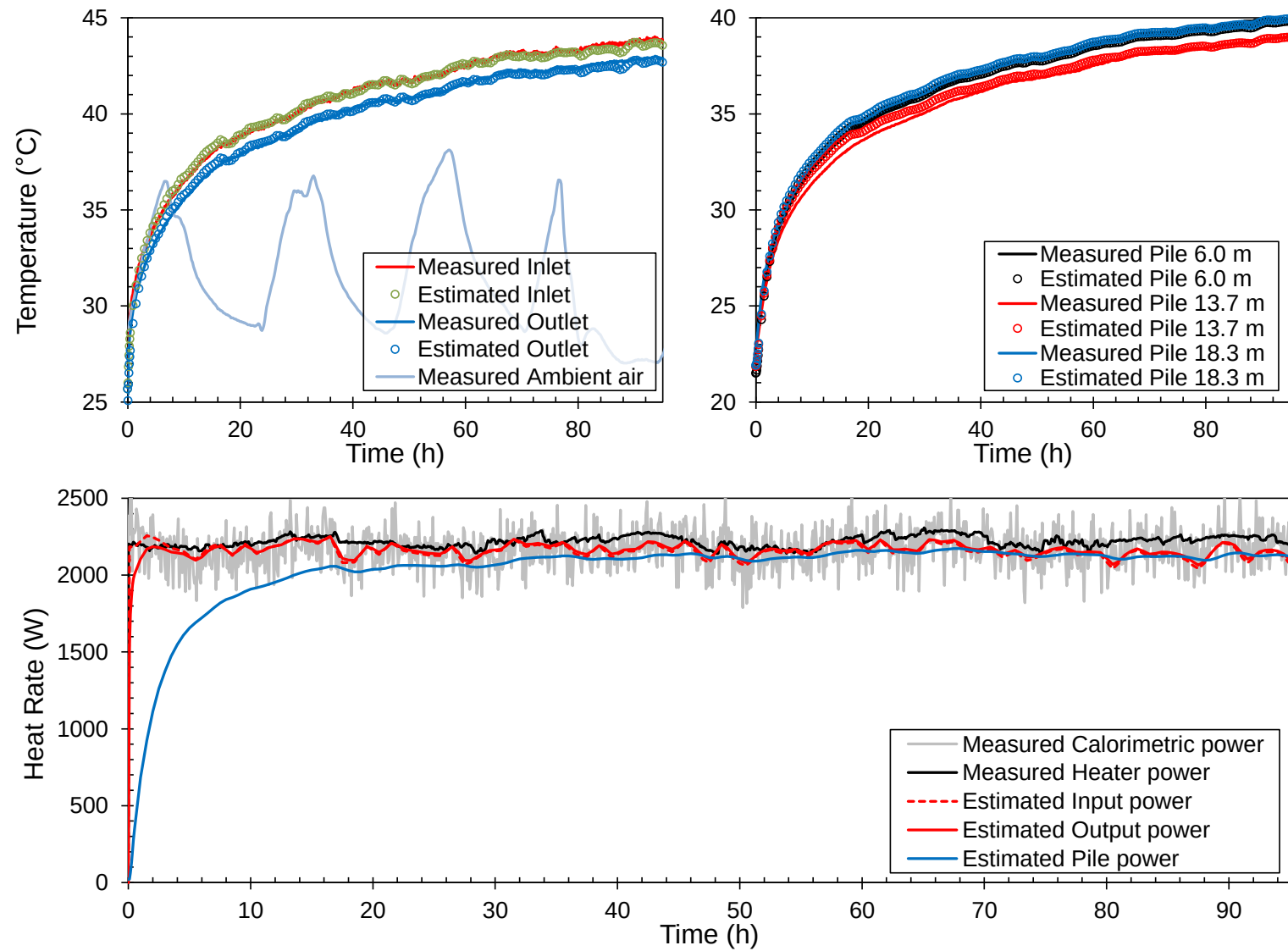


Figure B.10 : Temperature and heat rate plots of best fit analysis for Test #5 ($p = 2.5\%$).

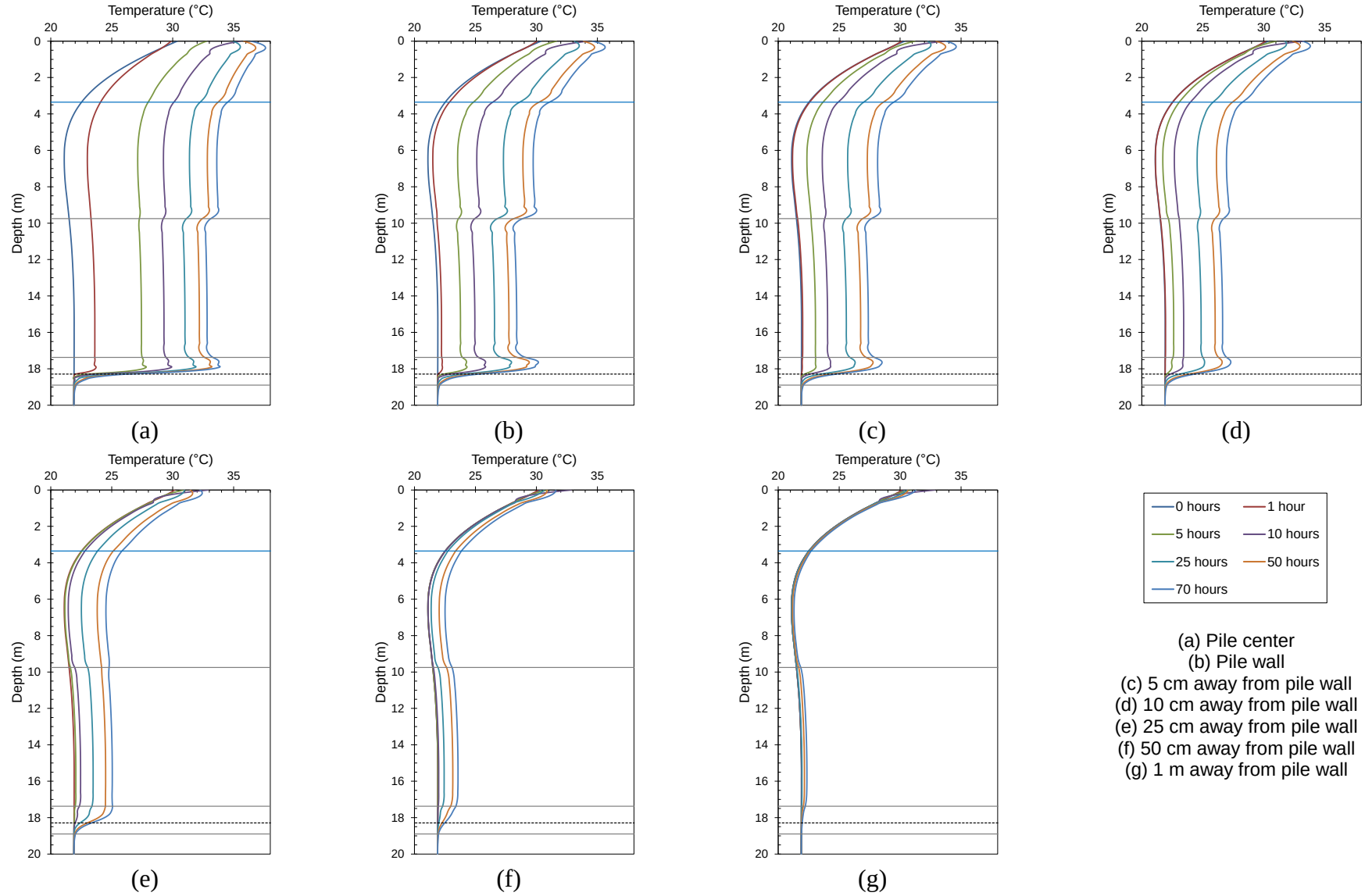


Figure B.11 : Temperature profile plots of best fit analysis for Test #1 ($p = 7.5\%$).

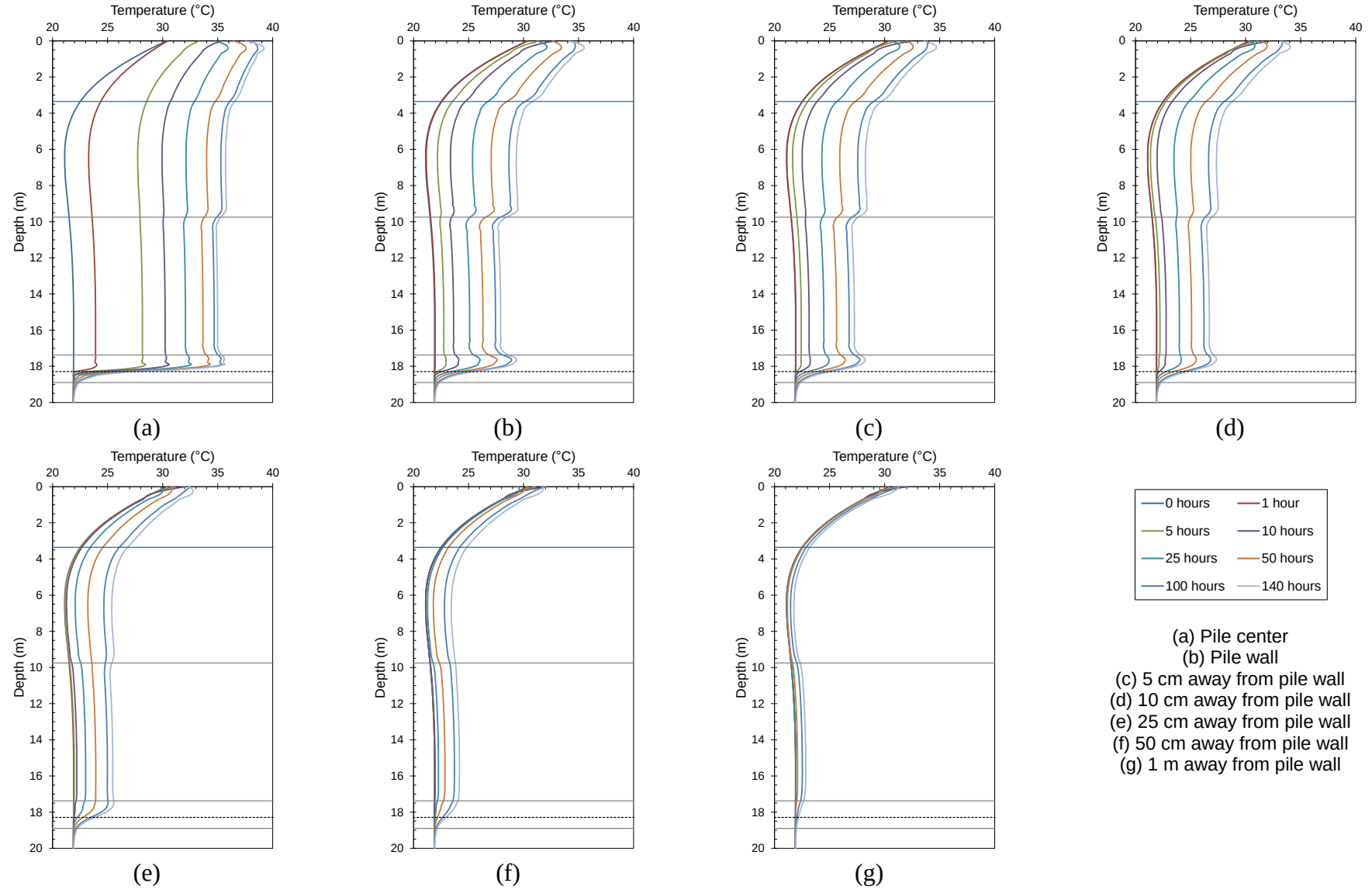


Figure B.12 : Temperature profile plots of best fit analysis for Test #2 ($p = 7.5\%$).

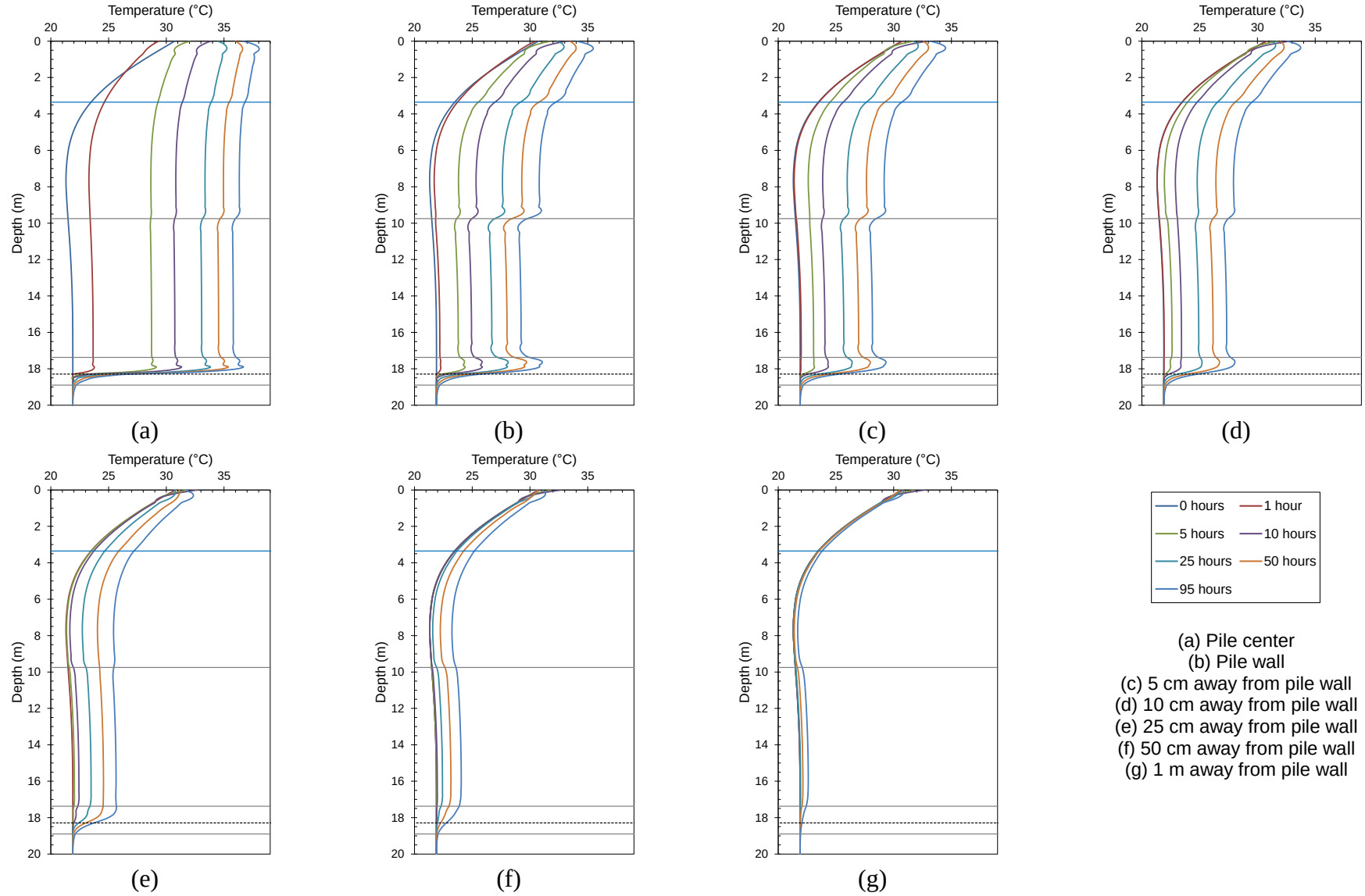


Figure B.13 : Temperature profile plots of best fit analysis for Test #3 ($p = 0.0\%$).

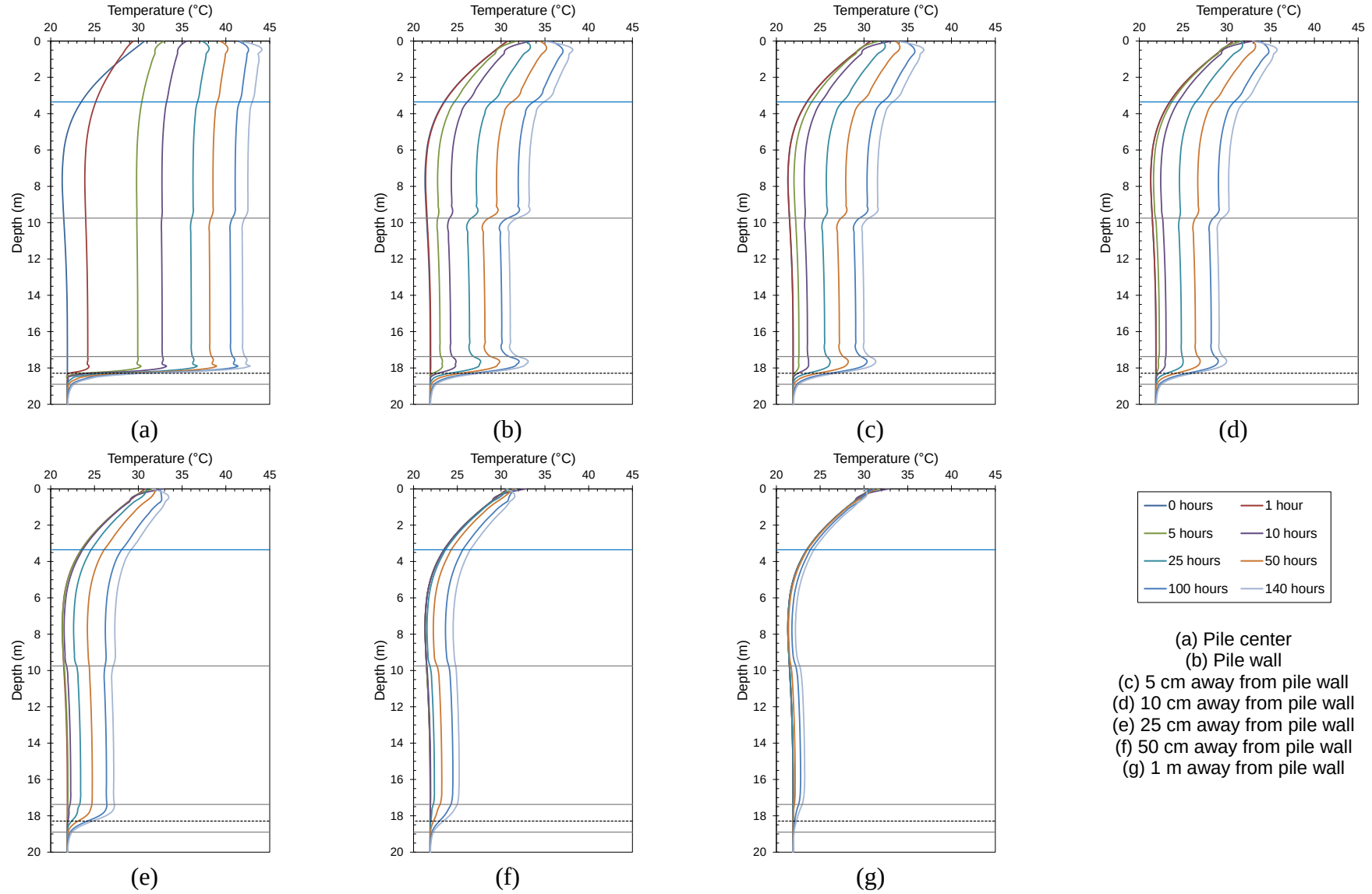


Figure B.14 : Temperature profile plots of best fit analysis for Test #4 ($p = 0.0\%$)

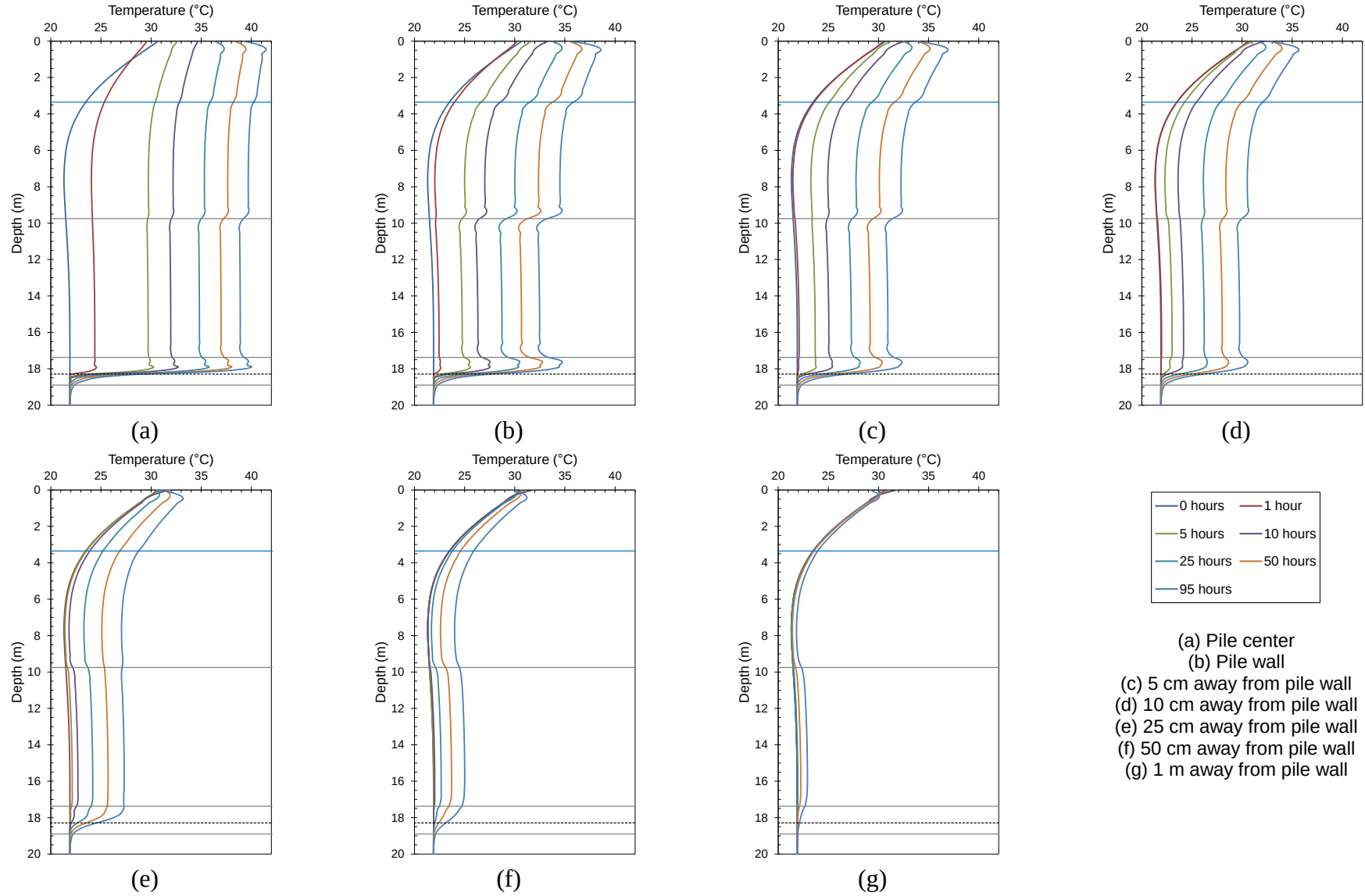


Figure B.15 : Temperature profile plots of best fit analysis for Test #5 ($p = 2.5\%$).

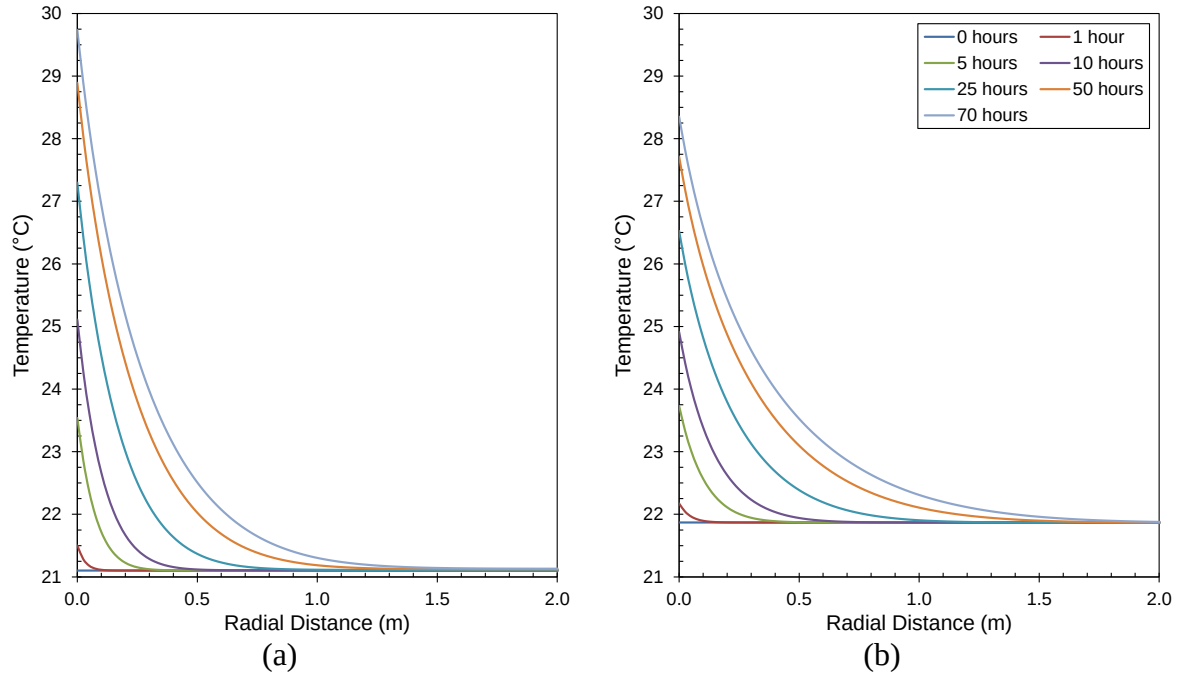


Figure B.16 : Temperature vs. radial distance plots of best fit analysis for Test #1 ($p = 7.5\%$) (a) mid-layer of Saturated Clay 1, (b) mid-layer of Sand 1.

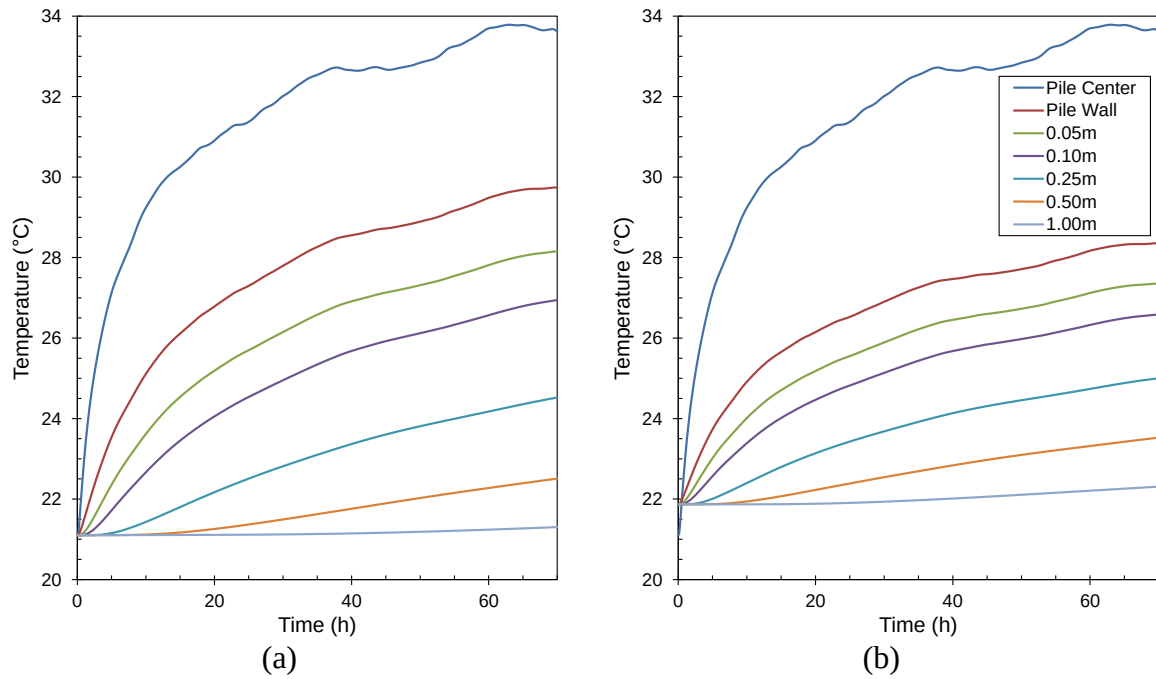


Figure B.17 : Temperature vs. time plots of best fit analysis for Test #1 ($p = 7.5\%$). (a) mid-layer of Saturated Clay 1, (b) mid-layer of Sand 1.

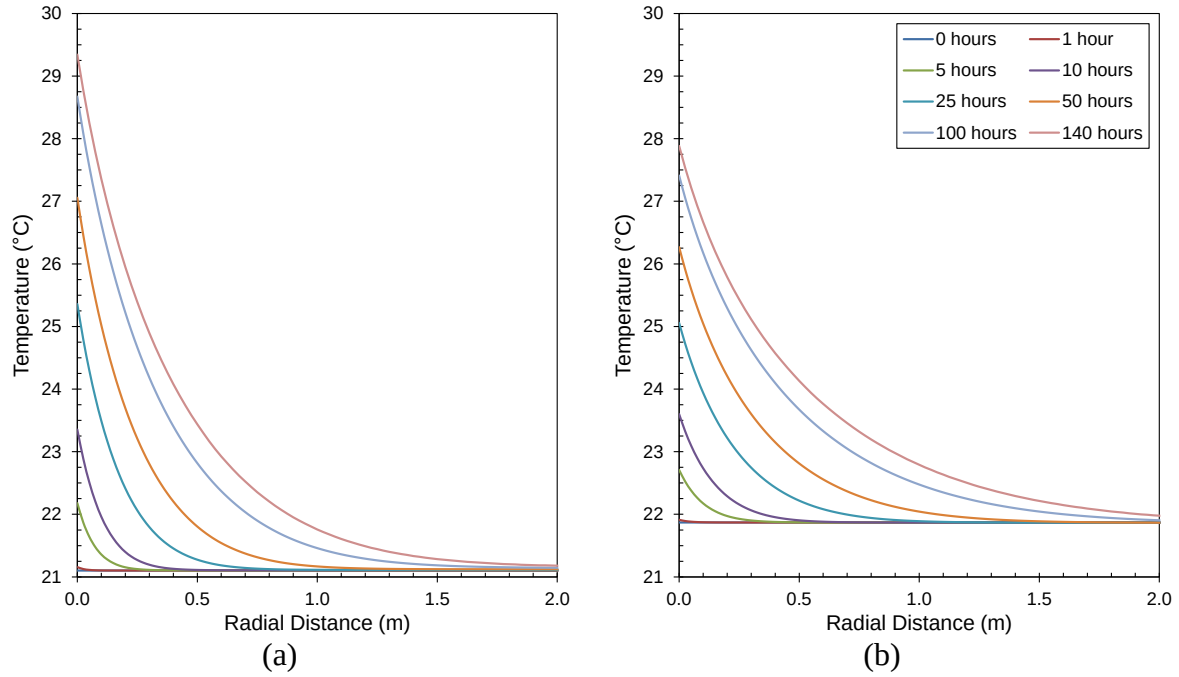


Figure B.18 : Temperature vs. radial distance plots of best fit analysis for Test #2 ($p = 7.5\%$).
(a) mid-layer of Saturated Clay 1, (b) mid-layer of Sand 1.

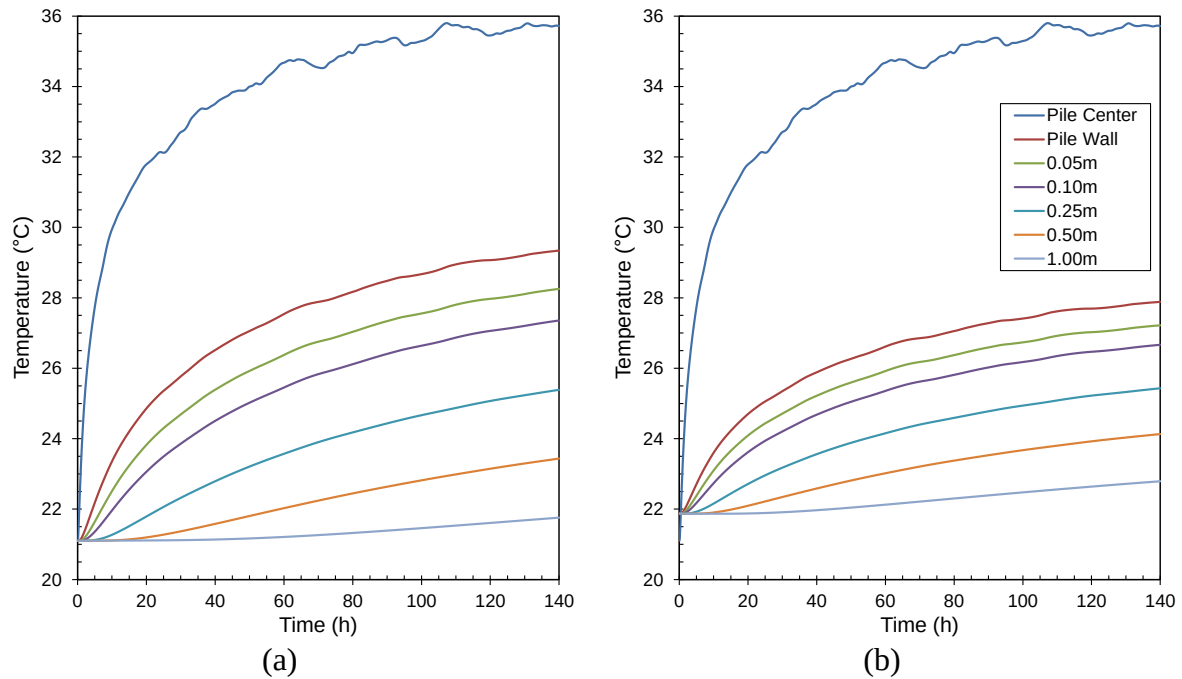


Figure B.19 : Temperature vs. time plots of best fit analysis for Test #2 ($p = 7.5\%$).
(a) mid-layer of Saturated Clay 1, (b) mid-layer of Sand 1.

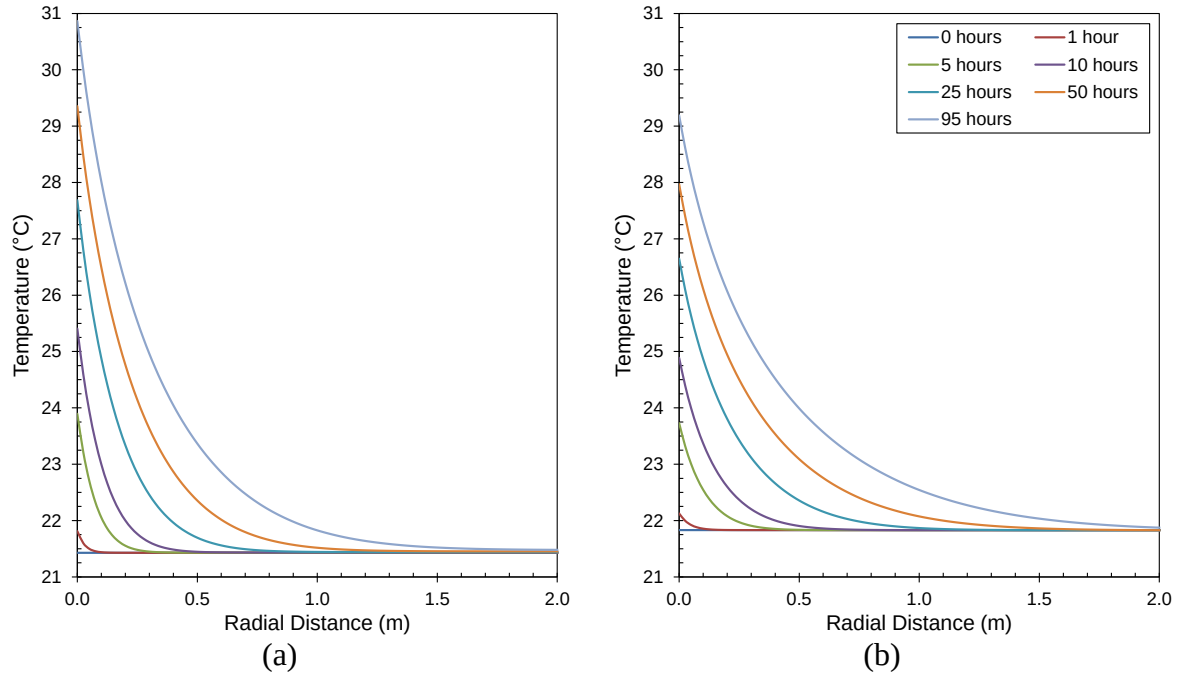


Figure B.20 : Temperature vs. radial distance plots of best fit analysis for Test #3 ($p = 0.0\%$).
(a) mid-layer of Saturated Clay 1, (b) mid-layer of Sand 1.

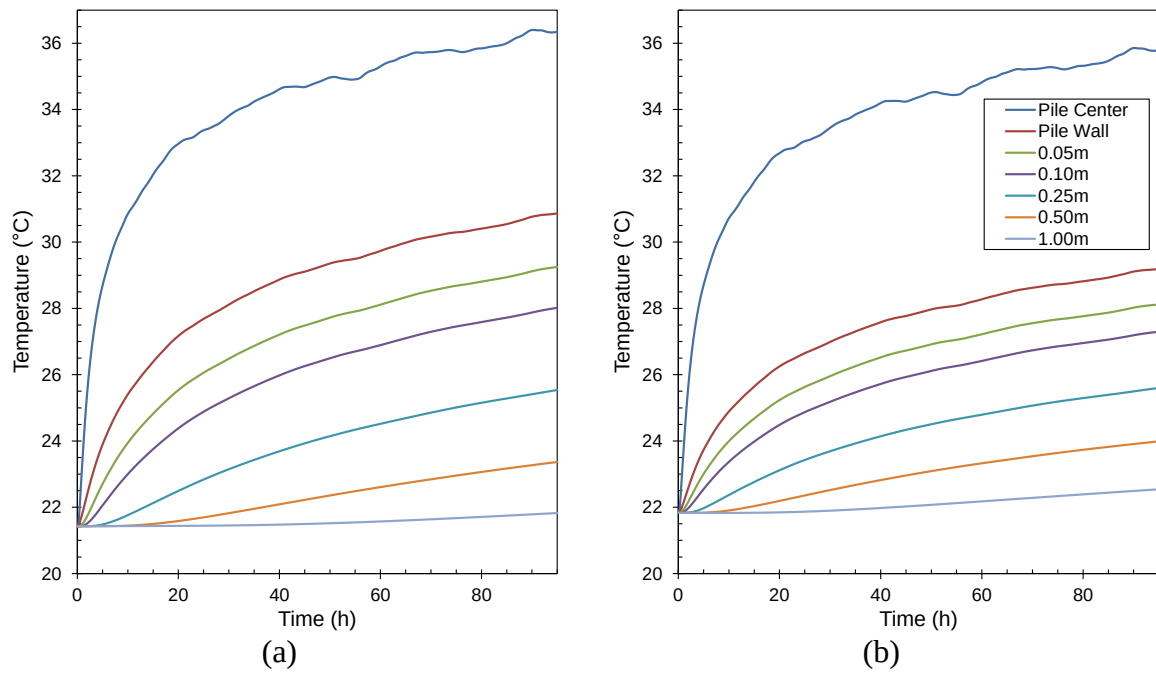


Figure B.21 : Temperature vs. time plots of best fit analysis for Test #3 ($p = 0.0\%$).
(a) mid-layer of Saturated Clay 1, (b) mid-layer of Sand 1.

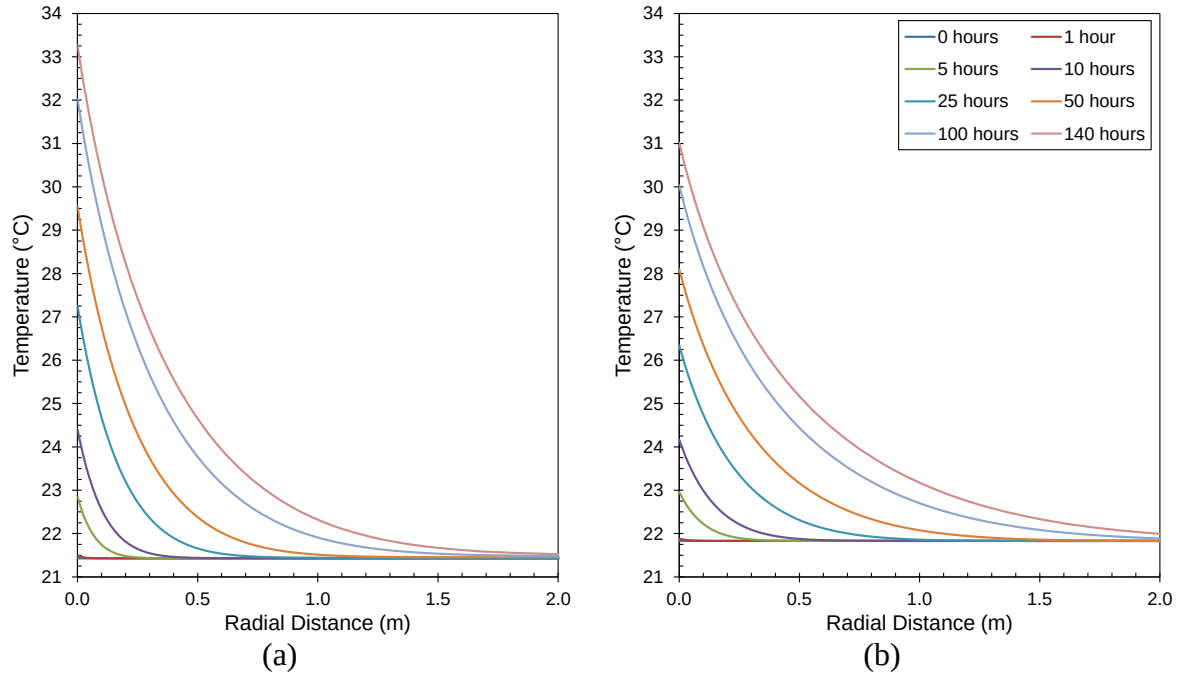


Figure B.22 : Temperature vs. radial distance plots of best fit analysis for Test #4 ($p = 0.0\%$).
(a) mid-layer of Saturated Clay 1, (b) mid-layer of Sand 1.

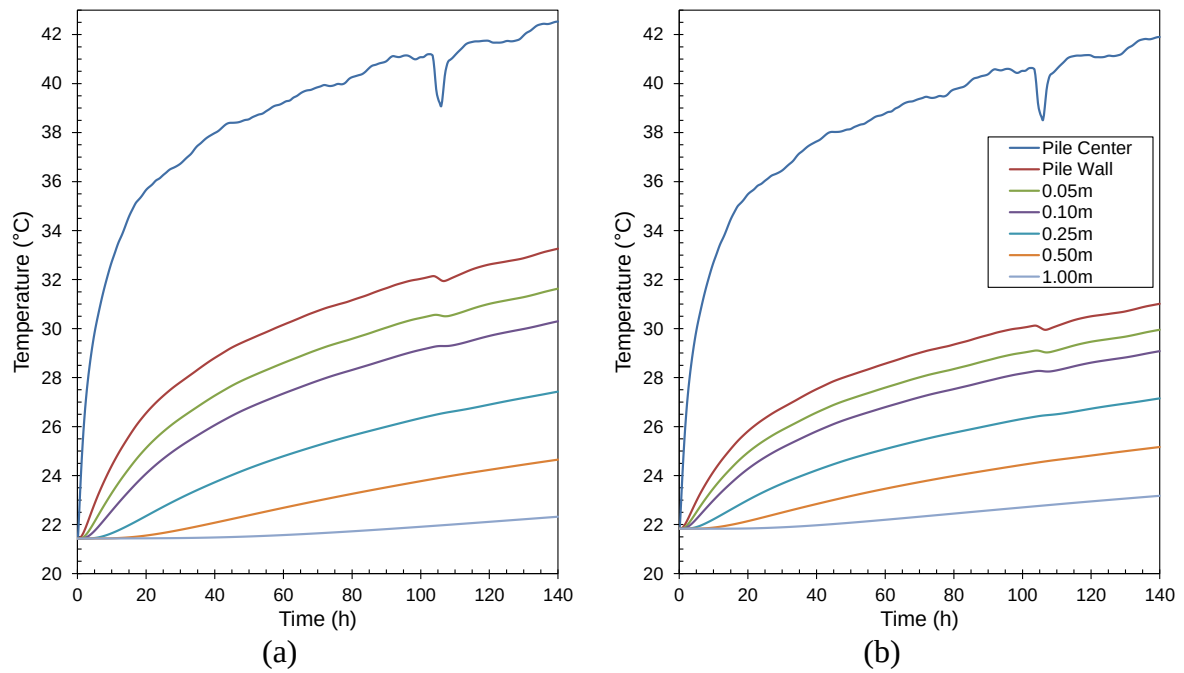


Figure B.23 : Temperature vs. time plots of best fit analysis for Test #4 ($p = 0.0\%$).
(a) mid-layer of Saturated Clay 1, (b) mid-layer of Sand 1.

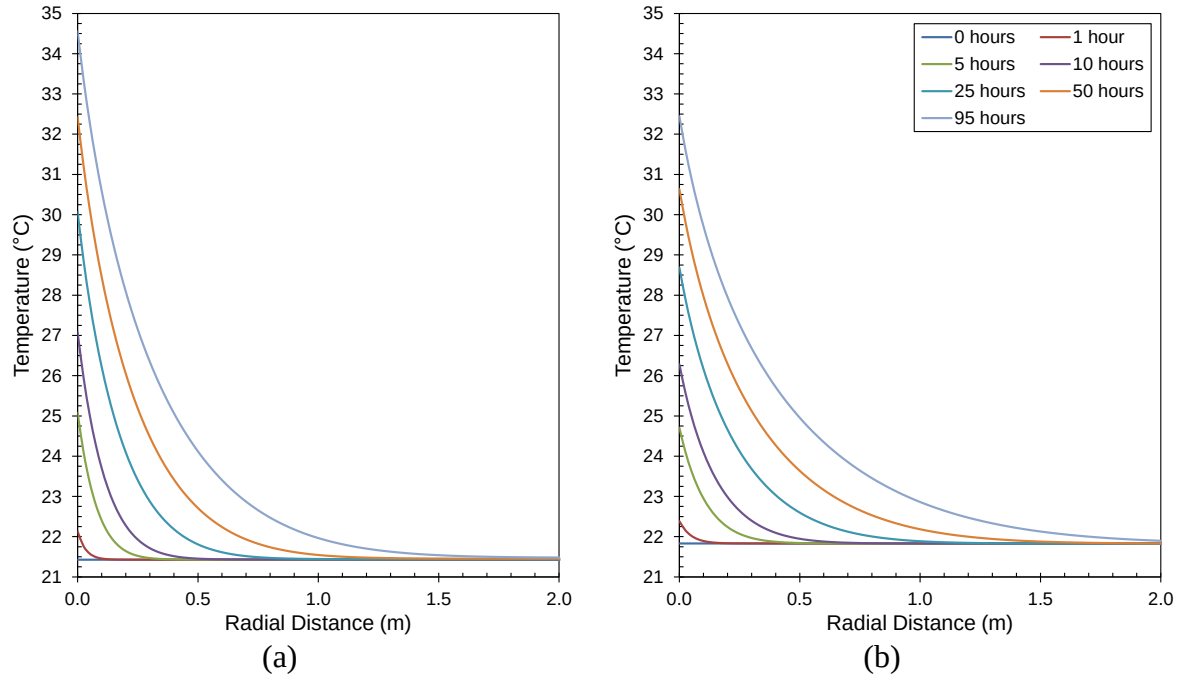


Figure B.24 : Temperature vs. radial distance plots of best fit analysis for Test #5 ($p = 2.5\%$).
(a) mid-layer of Saturated Clay 1, (b) mid-layer of Sand 1.

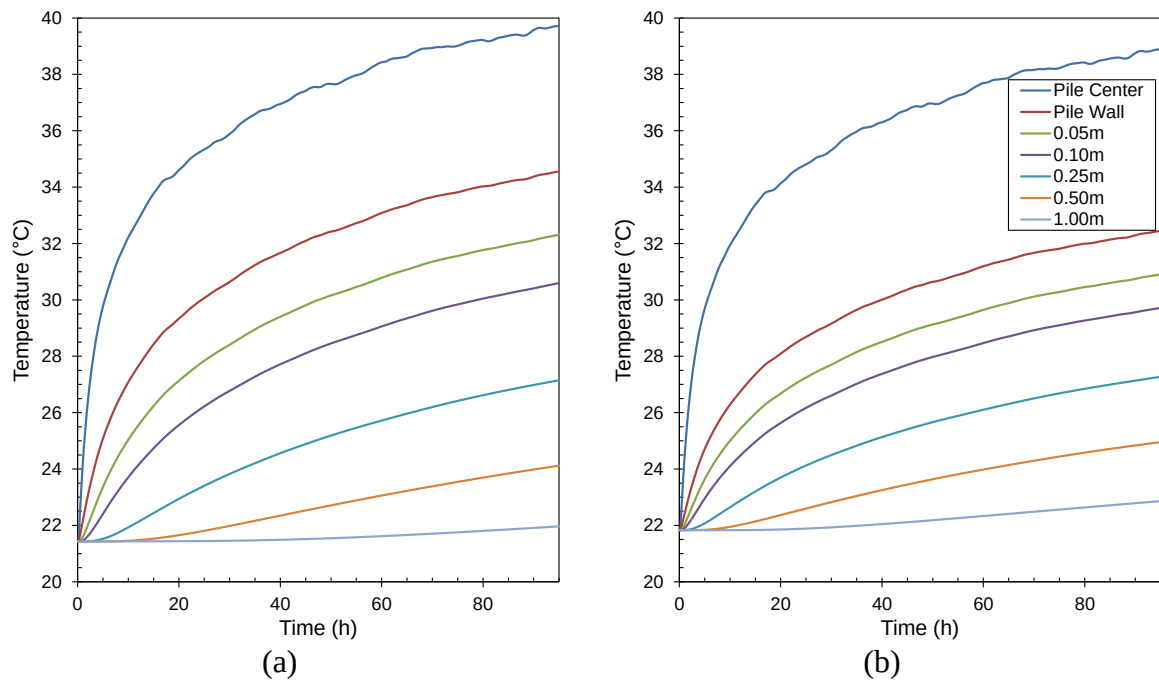


Figure B.25 : Temperature vs. time plots of best fit analysis for Test #5 ($p = 2.5\%$).
(a) mid-layer of Saturated Clay 1, (b) mid-layer of Sand 1.

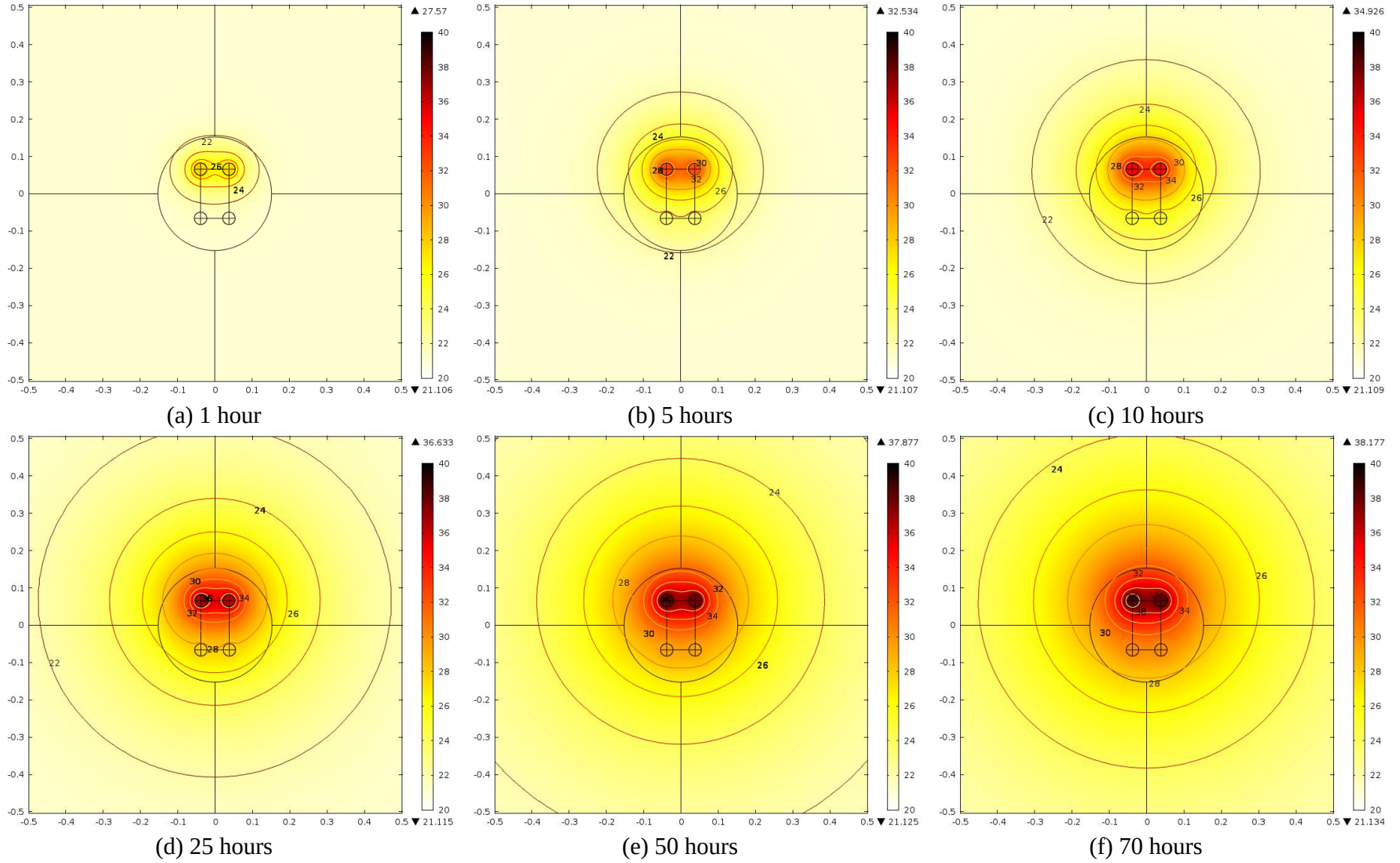


Figure B.26 : Temperature contour plots of best fit analysis at mid-layer of Saturated Clay 1 for Test #1 ($p = 7.5\%$).

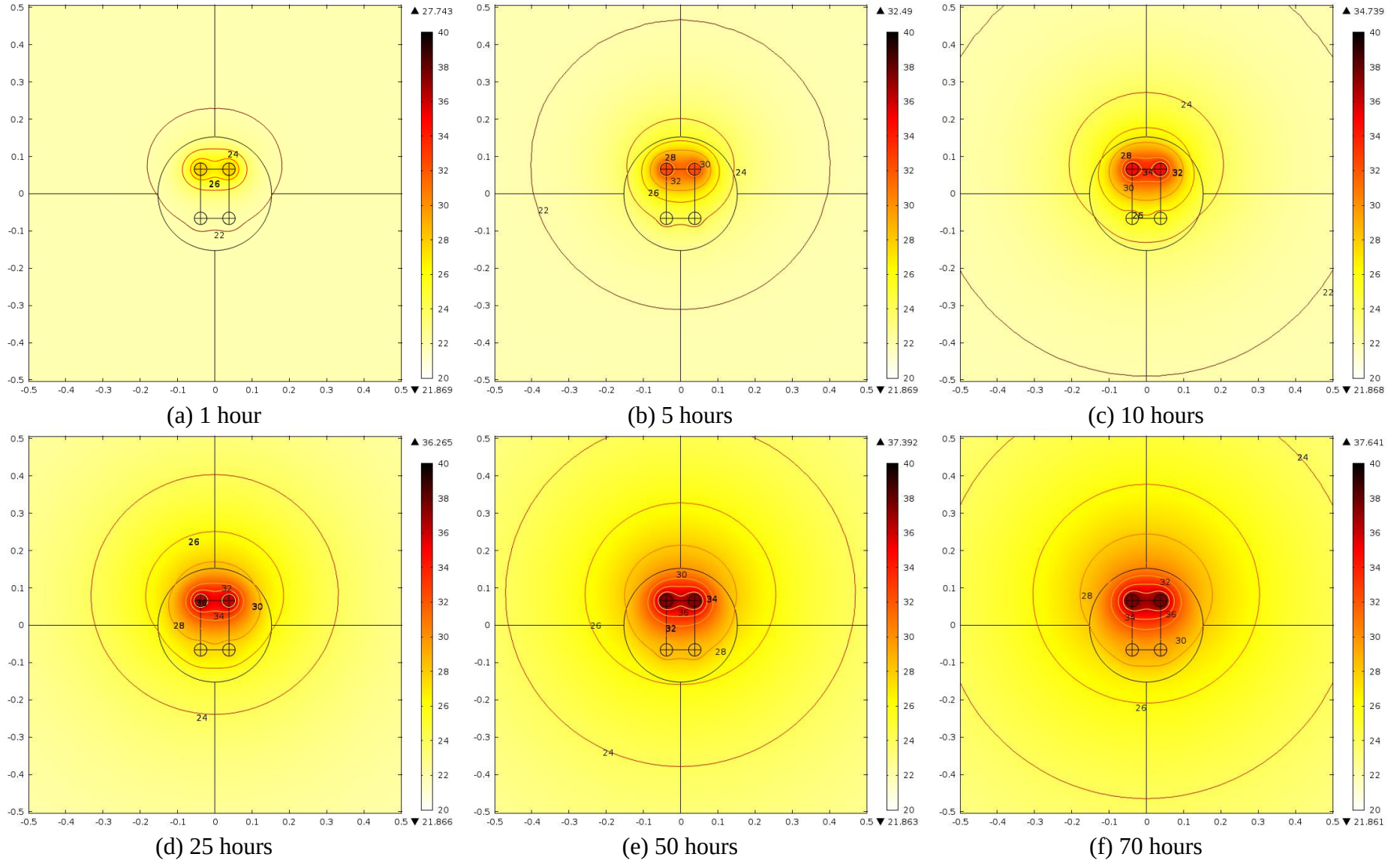


Figure B.27 : Temperature contour plots of best fit analysis at mid-layer of Sand 1 for Test #1 ($p = 7.5\%$).

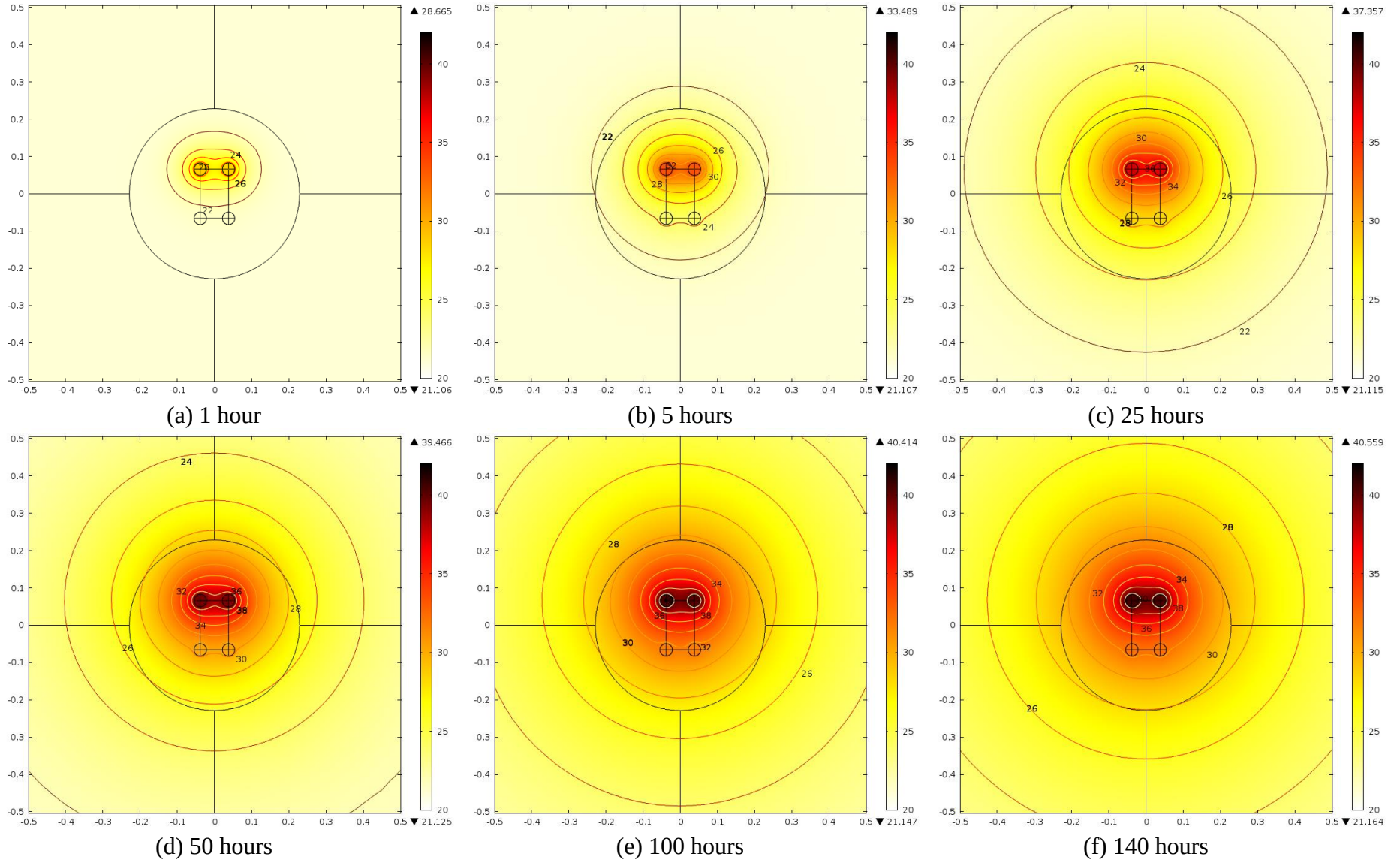


Figure B.28 : Temperature contour plots of best fit analysis at mid-layer of Saturated Clay 1 for Test #2 ($p = 7.5\%$).

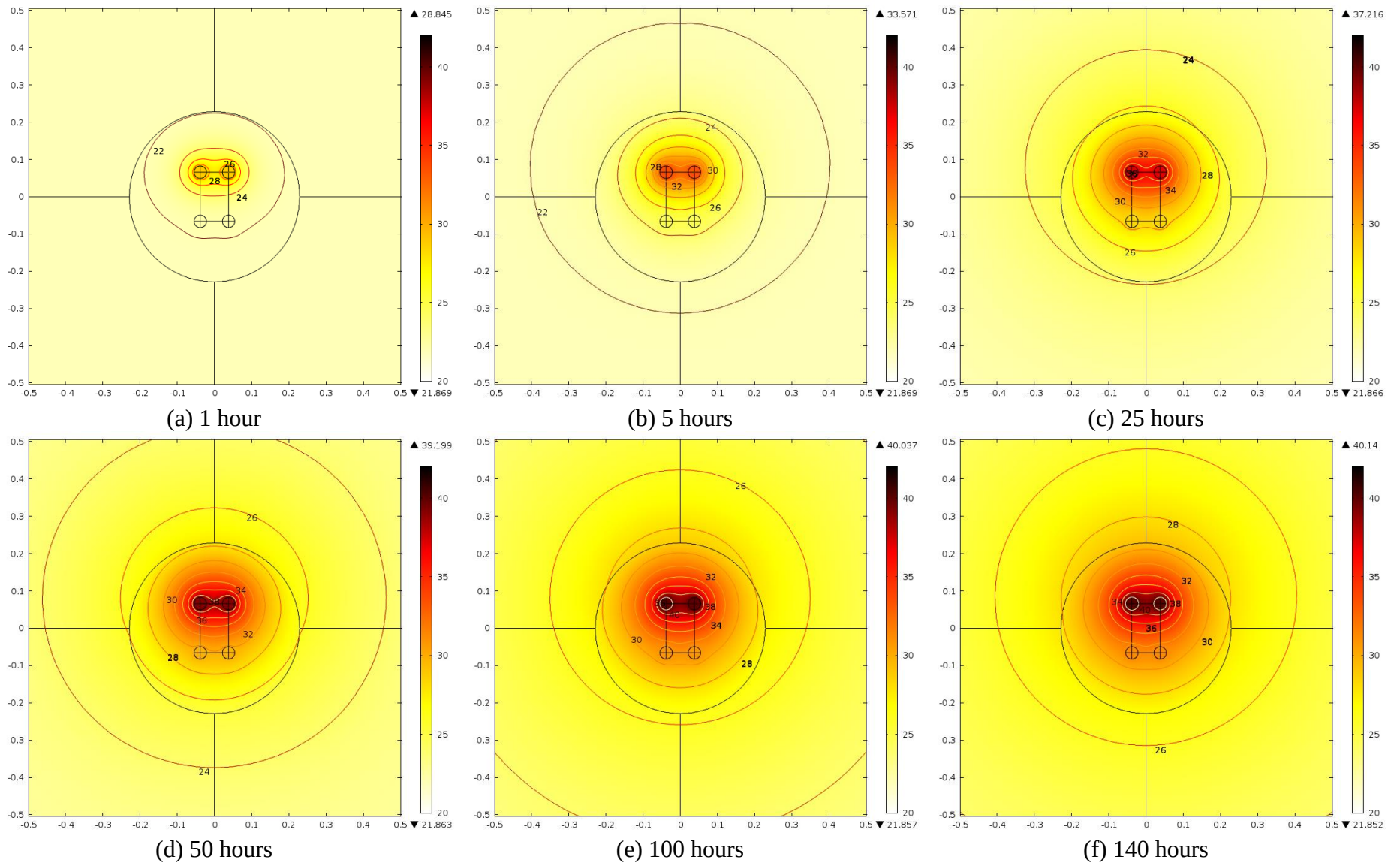


Figure B.29 : Temperature contour plots of best fit analysis at mid-layer of Sand 1 for Test #2 ($p = 7.5\%$).

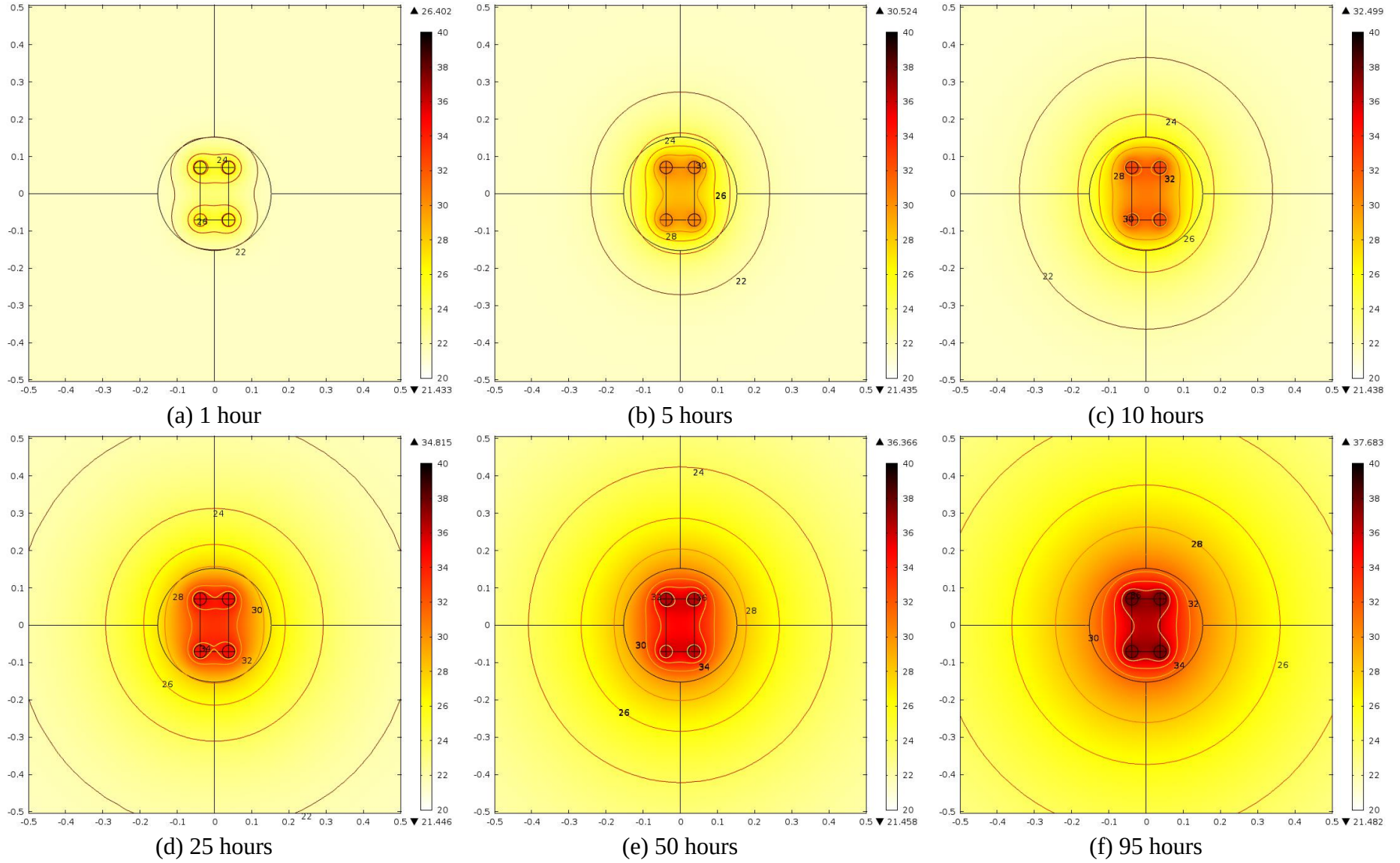


Figure B.30 : Temperature contour plots of best fit analysis at mid-layer of Saturated Clay 1 for Test #3 ($p = 0.0\%$).

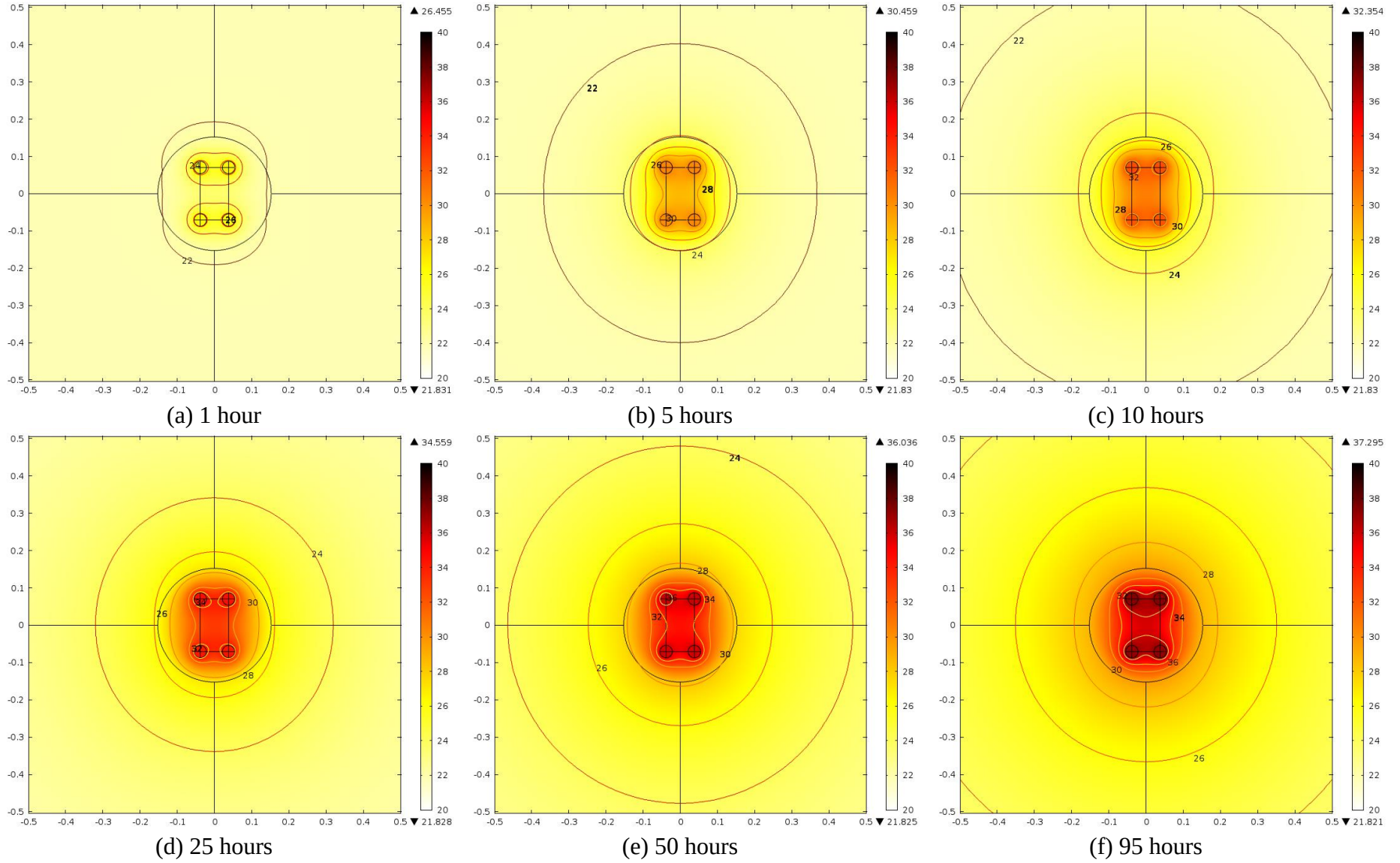


Figure B.31 : Temperature contour plots of best fit analysis at mid-layer of Sand 1 for Test #3 ($p = 0.0\%$).

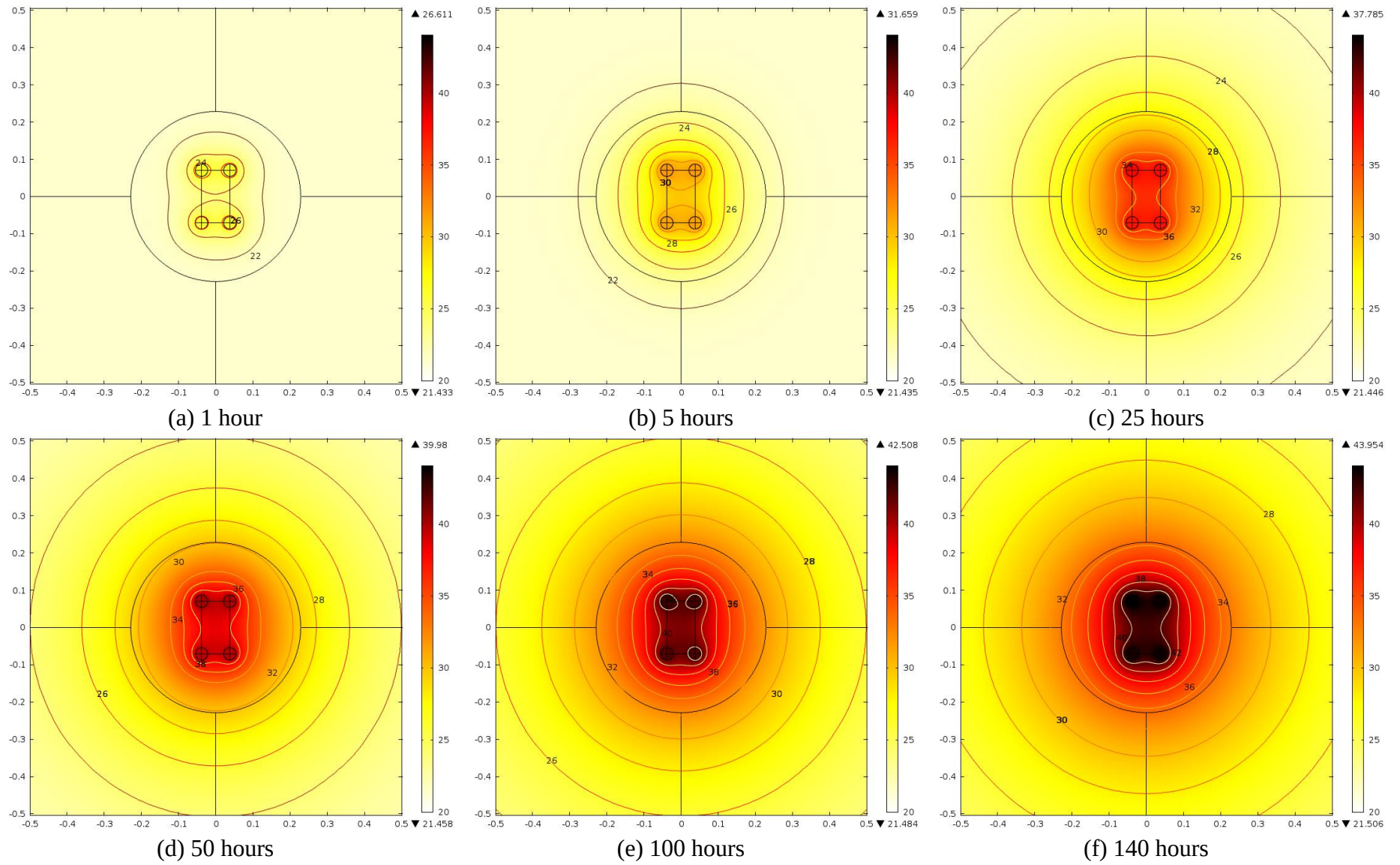


Figure B.32 : Temperature contour plots of best fit analysis at mid-layer of Saturated Clay 1 for Test #4 ($p = 0.0\%$).

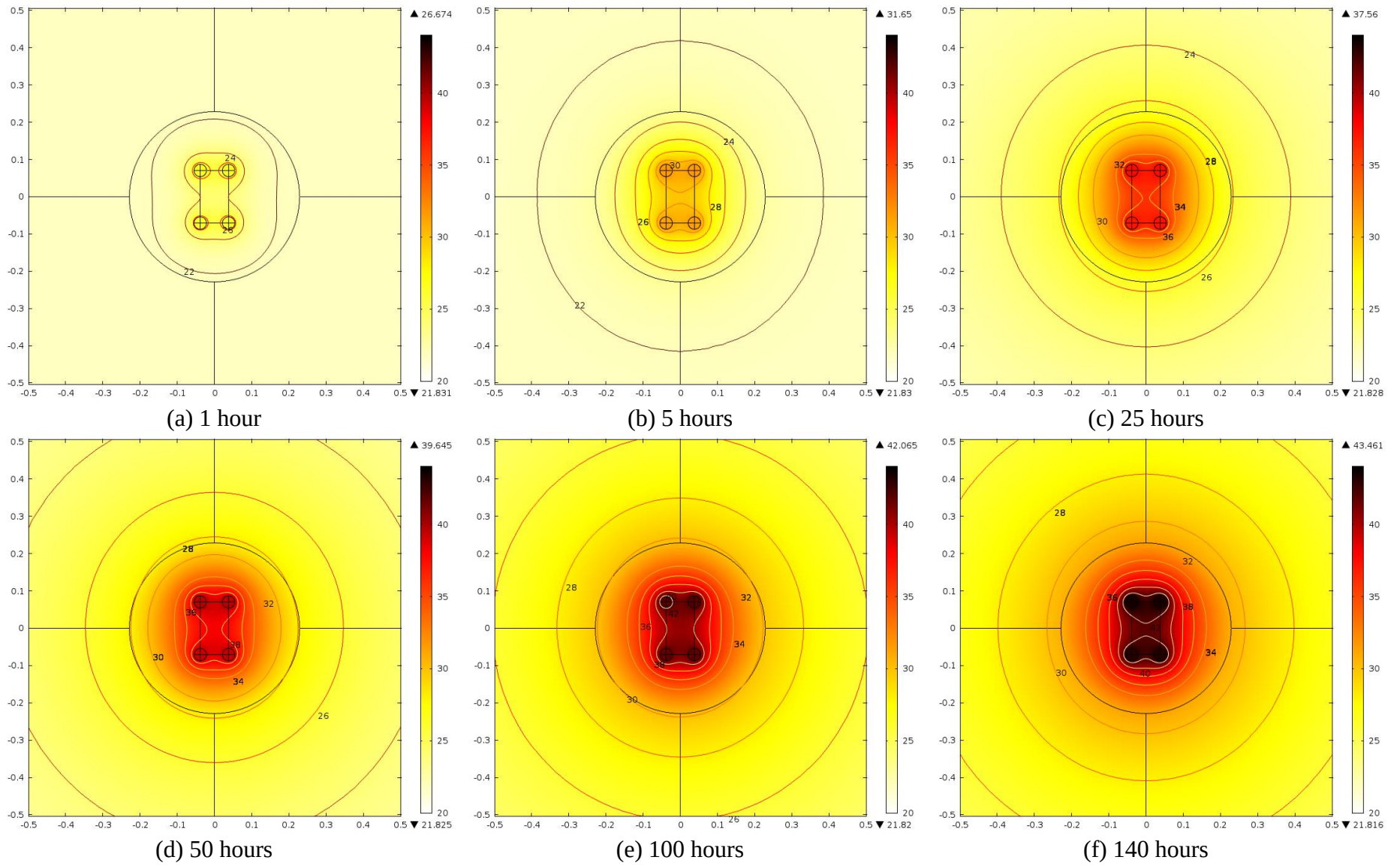


Figure B.33 : Temperature contour plots of best fit analysis at mid-layer of Sand 1 for Test #4 ($p = 0.0\%$).

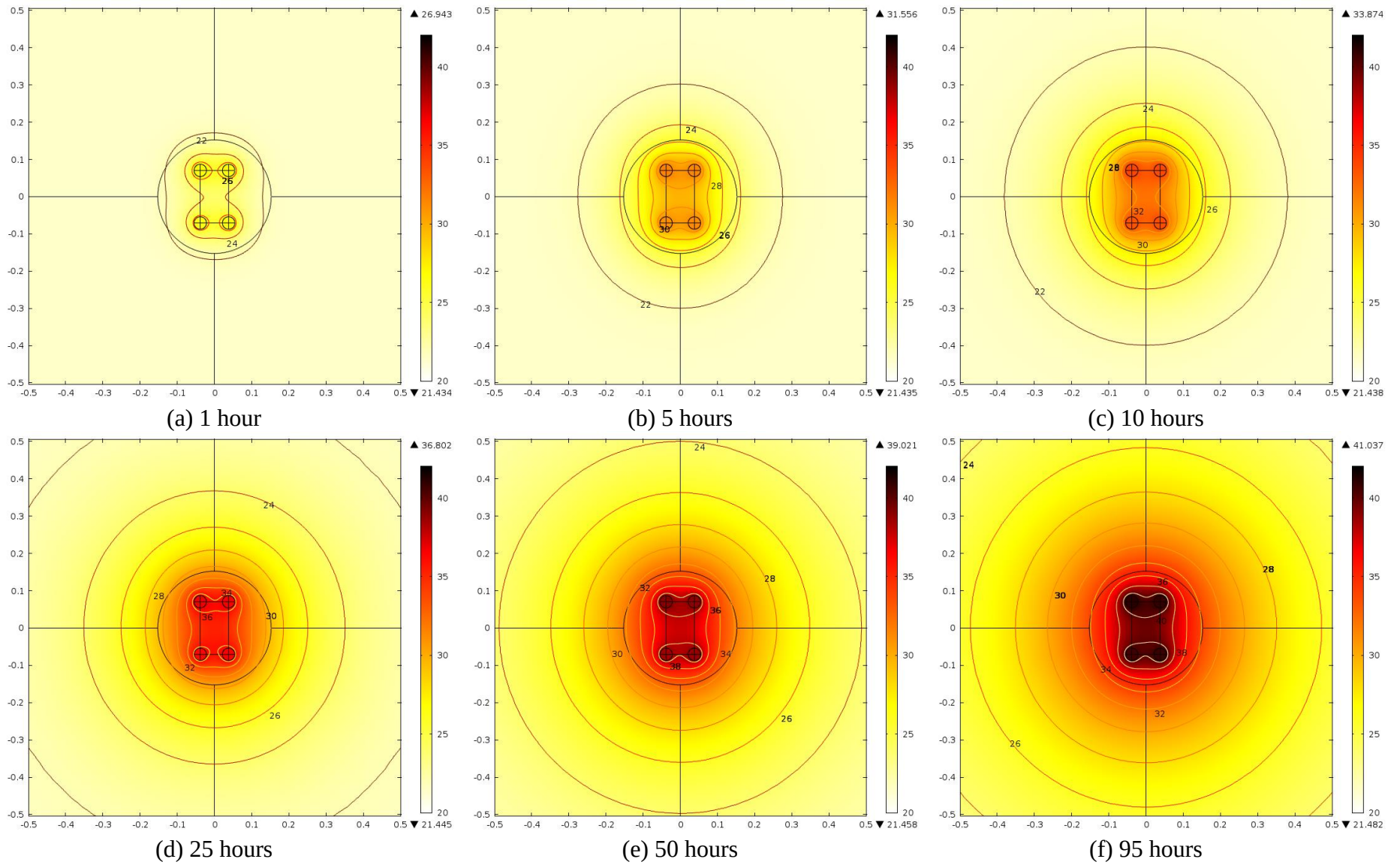


Figure B.34 : Temperature contour plots of best fit analysis at mid-layer of Saturated Clay 1 for Test #5 ($p = 2.5\%$).

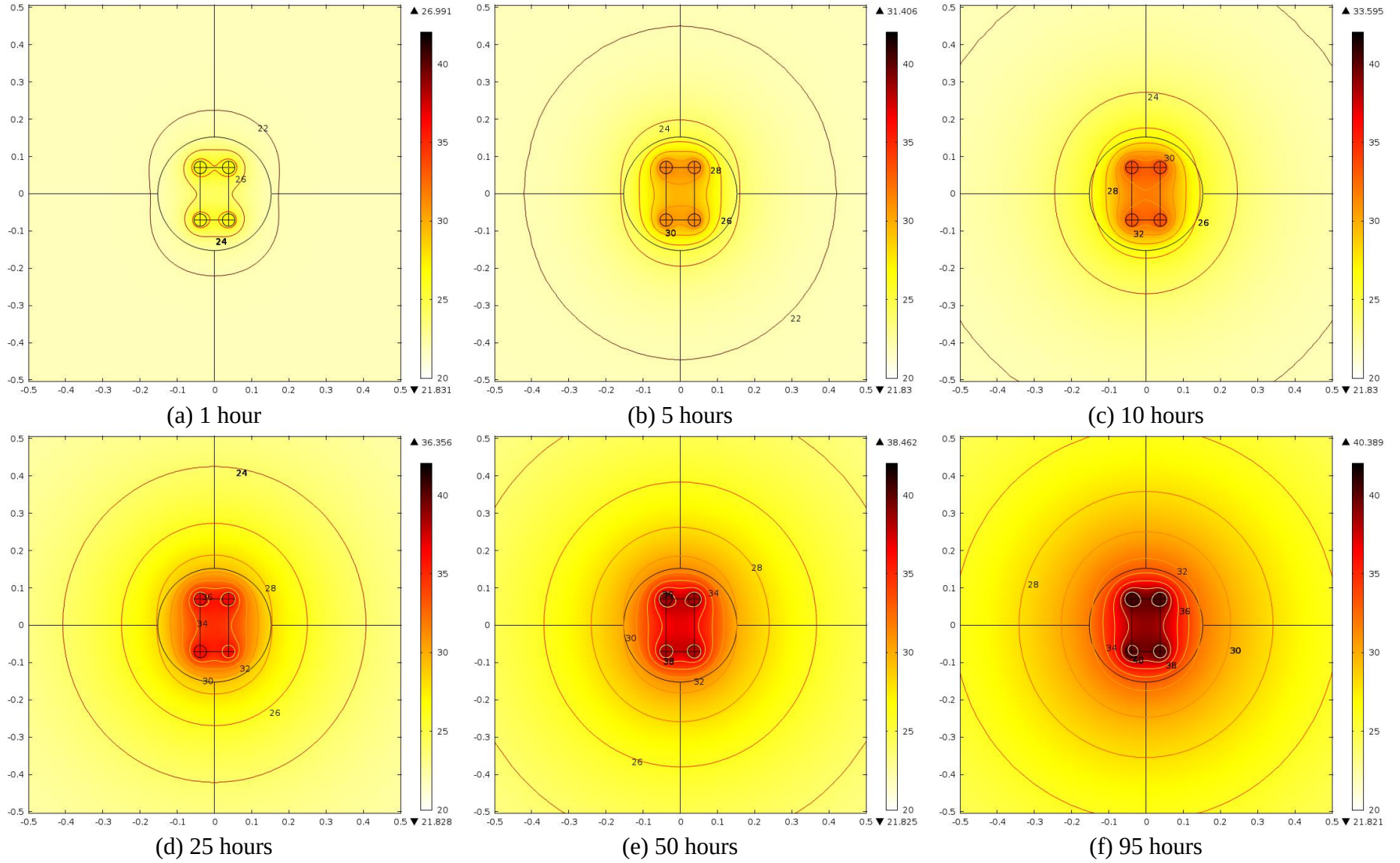


Figure B.35 : Temperature contour plots of best fit analysis at mid-layer of Sand 1 for Test #5 ($p = 2.5\%$).



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- Olgun, C. G., **Ozudogru, T. Y.**, Abdelaziz, S. L. and Senol, A. (2014). Long-term performance of heat exchanger piles. *Acta Geotech.* doi:10.1007/s11440-014-0334-z.
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- **Ozudogru, T. Y.**, Brettmann, T., Olgun, C. G., Martin II, J. R. and Senol, A. (2012). Thermal Conductivity Testing of Energy Piles: Field Testing and Numerical Modeling. *Geo-Congress 2012*, Geo Institute, Oakland, CA, USA, pp. 4436–4445. doi:10.1061/9780784412121.456.
- **Ozudogru, T. Y.**, Ghassemi-Fare, O., Olgun, C. G. and Basu, P. (2015). Numerical modeling of vertical geothermal heat exchangers using finite difference and finite element techniques. *Geotech. Geol. Eng.*, 33, 291-306. doi:10.1007/s10706-014-9822-z.
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- **Ozudogru, T. Y.**, Olgun, C. G. and Senol, A. (2014). 3D numerical modeling of vertical geothermal heat exchangers. *Geothermics*, 51, 312-324. doi:10.1016/j.geothermics.2014.02.005.