THE DEVELOPMENT OF AN OFF-PEAK GROUND PRE-COOL CONTROL
STRATEGY FOR HYBRID GROUND-SOURCE HEAT PUMP SYSTEMS

by

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Abstract

THE DEVELOPMENT OF AN OFF-PEAK GROUND PRE-COOL CONTROL STRATEGY FOR HYBRID GROUND SOURCE HEAT PUMP SYSTEMS

Master of Applied Science, 2016

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Hybrid Ground-Source Heat Pump (HGSHP) systems have been introduced as a remedy to overcome the current financial hurdles associated to the installation of geo-exchange technology. However, there still remains potential for increased economic feasibility of geo-exchange through proactive operation prefaced with higher level control. This study introduces a control strategy referred to as an off-peak ground pre-cool, employing time-of-use conscious operating logic capable of facilitating artificial bore-field pre-conditioning to improve a geo-exchange system's cooling mode performance. Artificial pre-conditioning of a system's bore-field introduces the potential to improve the bore-field's thermal characteristics in a controlled manner. With improved thermal characteristics a bore-field can be exploited more efficiently during the following peak periods; introducing additional economic incentives by reducing peak power consumption attributed to space cooling. This study presents a multifaceted approach which intends to concurrently address improving system economics and aid in the balancing of the electrical grid.

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List of Terms and Abbreviations

ACC	Annual Cooling Cost
AC	Air-Conditioner
ACE	Annual Carbon Emissions
AECC	Annual Electricity Consumption, Cooling
AFL	Adaptive Fuzzy Logic
AHU	Air-Handling Unit
ASHP	Air-Source Heat Pump
ANN	Adaptive Neural Network
BES	Building Energy Simulation
BHE	Bore-Field Heat Exchanger
CL_{max}	Peak cooling demand (kW)
COP	Coefficient of Performance
COP_{GSHP}	Predicted ground-source heat pump coefficient of performance
$COP_{\Delta t}$	Ground-source heat pump coefficient of performance as a function of cycle time
COP_{AUX}	Auxiliary system coefficient of performance
CT	Cooling Tower
DHW	Domestic Hot Water
DS	Distribution System
DSM	Demand Side Management
EER	Energy Efficiency Ratio
EIA	Energy Information Administration
EST	Entering Sink Temperature
FL	Fuzzy Logic
FS	Full Season
f_{PC}	Off-peak performance correction factor
GCHP	Ground-Coupled Heat Pump
GHG	Green-House Gas
GSHP	Ground-Source Heat Pump

HDPE	High-Density Polyethylene
HGSHP	Hybrid Ground-Source Heat Pump
HVAC	Heating, Ventilation, and Air-Conditioning
MPC	Model Predictive Control
OEB	Ontario Energy Board
OGPC	Off-Peak Ground Pre-Cool
PCM	Phase Change Material
PID	Proportional-Integral-Derivative
PS	Peak Season
Q_{GSHP}	Ground-source heat pump cooling capacity (kW)
Q_{AUX}	Auxiliary system cooling capacity (kW)
RL	Reinforcement-Learning
SAGSHP	Solar Assisted Ground-Source Heat Pump
SS	Shoulder Season
SSC	Split-System Configuration
TCD	Total Cooling Demand (MWh)
TCDM	Total Cooling Demand Met by the ground-source heat pump (%)
TES	Thermal Energy Storage
THD	Total Heating Demand (MWh)
TOU	Time-of-Use
TRCA	Toronto Region Conservation Authority
VFD	Variable Frequency Drive
ΔCOP	Predicted increase in ground-source heat pump coefficient of performance due to a pre-cooled bore-field
ΔQ_{out}	Predicted increase in ground-source heat pump cooling output due to a pre-cooled bore-field
α	Hybrid system shave factor for peak cooling demand

CHAPTER 1 – Introduction

1.1 Motivation

As the global population rises, the world energy demand increases to compensate for the consumption habits distinctive to the modern millennial. There is a prominent need for the development and integration of sustainable energy alternatives to alleviate our dependence on energy sources that produce substantial carbon footprints. Geothermal energy is an alternative source which is considered an abundant and inexhaustible resource. It is predicated that the thermal energy that is annually produced within the earth is virtually double the global yearly energy demand [1].

According to the U.S. Energy Information Administration (EIA), the building sector consumed 47.6% of the total energy in the United States, as of 2012 [2]. Space heating/cooling requirements contribute to a significant portion of a building's energy usage, typically accounting for 50% of end use consumption [3, 4]. Harnessing free and abundant geothermal energy sources help contribute to a reduction in unsuitable energy consumption, improving building energy efficiency, which mitigates our carbon footprint.

Ground-Source Heat Pump (GSHP) systems are a sustainable energy alternative for conventional space heating/cooling equipment. GSHP technology utilizes the ground's thermal mass as a source/sink resulting in improved operating efficiency, due to its stable and consistent temperature [5]. Although promising, GSHP system popularity in industry is restricted due their undesirable economics; stemming from large capital investments and resulting in unappealing payback periods [6]. Hybridizing GSHP systems with conventional Heating, Ventilation, and Air-Conditioning (HVAC) units is a technique used to remedy the aforementioned financial barriers; enabling this technology to be implemented in a more economically feasible manner [7]. However, GSHP and

traditional HVAC systems commonly adopt lower level control techniques, due to controller predictability and reliability [4]. In the literature as well as in practice, there still remains potential for the further improvement of GSHP system economics, through the development of intelligent/proactive control strategies complemented with higher level control schemes.

1.2 Objectives

The aim of the present work is to propose a newly developed proactive control strategy, intended for the operation of HGSHP systems. The suggested scheme is referred to as an Off-Peak Ground Pre-Cool (OGPC) control strategy, which employs Time-of-Use (TOU) conscious control logic to facilitate an off-peak artificial bore-field pre-cool. This strategy aims to create improved bore-field thermal characteristics during peak cooling hours resulting in increased GSHP system operating efficiency (cooling mode); with the end goal being a reduction in HGSHP system operating cost by means of peak power shaving. The proposed control methodology presents an alternative means of operating a Hybrid GSHP (HGSHP) system, wherein conventional operation is reactive to cooling loads, using auxiliary cooling units as supplementary demand relief. The OGPC strategy employs a proactive performance enhancing operating perspective, using the auxiliary cooling system as a flexible load to provide a controlled bore-field pre-condition. The OGPC strategy not only address improving system economics through an increase in operating efficiency, but also presents a multifaceted approach which intends to concurrently aid the balancing of the overloaded electrical grid.

With the aid of a newly developed HGSHP numerical performance prediction model, as part of this thesis, a HGSHP systems annual performance can be evaluated. A preliminary analysis of the proposed OGPC control scheme is presented for three HGSHP system cases (corresponding to three unique buildings), where the numerical performance prediction model was used to evaluate

hybrid system performance. The numerical model developed in this study has been constructed as a platform for further expansion, and is aimed to be used as a computational tool in evaluating the impact that process control logic has on the performance of HGSHP systems.

1.3 Outline

Chapter 2 of this thesis presents background research and literature review of GSHP systems, conventional HVAC technology, and a summary of HVAC related control methodologies/techniques as it pertains to energy/cost efficient operation. Chapter 3 corresponds to a manuscript currently under review with Energy and Buildings, presenting a preliminary analysis of the potential benefit the proposed OGPC control strategy poses on the performance of a HGSHP system. Chapter 4 presents an analysis studying the OGPC control strategy's sensitivity to both hybrid system proportions and geographic location. Lastly, in Chapter 5, the overall conclusions for the thesis and recommendations for future work are presented.

CHAPTER 2 – Background Research and Literature Review

2.1 Ground-Source Heat Pump Technology Overview

With building energy consumption on the rise, power consumption saving strategies have become a global priority in many energy usage policies. Space heating and cooling contributes to a large portion of a building's net energy consumption, typically accounting for 50% of annual usage [3, 4]. As environmental awareness and potential resource shortages progress to the forefront of our concerns, an amplified need for sustainable alternatives will arise. GSHP systems are an alternative to conventional HVAC systems, as this technology provides the capability for sustainable and high efficiency building climate control.

'Ground-Source Heat Pumps' is a term that is used for a verity of systems that utilize the ground, groundwater, or surface water as a thermal source or sink. The most commonly used system makes use of the ground which acts as a thermal battery. Also commonly referred to as a 'ground coupled heat pump' (GCHP) or a 'geo-exchange system' [8]. GSHPs are based on an old technology that was initially developed for application in the residential sector [5]. Similar to conventional HVAC systems, for which heating/cooling is accomplished through a two-stage process comprising the furnace/air-source heat pump (ASHP) stage and the distribution system stage. The furnace/ASHP would provide the function of heating/cooling the working fluid (water-air/refrigerant-air) before it is distributed to the respective zones by the circulation system. The most commonly used distribution systems are forced-air and hydronic. For a GSHP system, a three-stage process is required to meet the necessary climate control thresholds. The difference in this configuration is based on the fact that the conventional heating/cooling system is substituted for a water-source heat pump, coupled to a ground heat exchanger and interfaced with a distribution system [8]. This configuration uses the ground as a thermal source/sink, extracting/rejecting heat between the

ground heat-exchanger and the soil interface; whereas conventional systems rely on ambient outdoor air as a thermal sink. With the variability of the outdoor air temperature, which can fluctuate drastically on an hourly basis, unavoidable deviations in conventional system performance commonly occur. Alternatively, a GSHP system provides improved operational performance due the stability of the ground temperature, acting as a predictable thermal reservoir [9].

2.1.1 Ground-Source Heat Pump Component Outline

GSHPs fall under the classification of a closed loop system, where the working fluid is of a fixed volume and is continually recirculated within the ground-loop [9]. Figure 1 provides a schematic illustration of the components essential to the operation of a GSHP system. The system consists of a vapor compression cycle, which is made reversible by a bi-directional control valve. The mechanics of a heat pump (water-to-air) included four major components: a refrigerant-to-water heat exchanger, an indoor coil, a compressor, and an expansion valve. The refrigerant-to-water heat exchanger (acting as an evaporator/condenser) transfers heat from/to the thermal source/sink (ground). The indoor coil (acting as a condenser/evaporator) is used in conjunction with a blower to transmit warm/cool air into the respective zone that is being heated/cooled. Finally, the compressor and expansion valve components are used to boost and relieve pressure, respectively, at critical points in the vapor compression cycle [8]. In many applications, heat pump units implement a desuperheater heat exchanger (illustrated in Figure 1) to preheat incoming potable (for hot water production) through the use of high-temperature refrigerant at the high pressure side of the compressor [8].

Figure 1: Closed loop GSHP system (reproduced from [8])

The most commonly implemented unit is a water-to-air heat pump, a water-antifreeze solution is circulated through a liquid-to-refrigerant heat exchanger and a ground-loop heat exchanger, buried below the earth's surface. The ground-loop provides the function of thermal energy injection/extraction to/from the ground [8]. The ground-loop heat exchanger is a critical component in the design of a GSHP system. There are two primary geo-exchange system types utilized in the industry: open loop and closed loop systems.

2.1.1.1 Open Loop Systems

Open loop geothermal systems function by the means of pumping ground water from natural wells/aquifers, acting as the system's working fluid. The entering working fluid has two potential routes when the system is operating, either direct application or heat pump assisted operation. For direct use applications, if the entering fluid temperature is sufficient to accommodate the building's load, it is directly used to provide heating/cooling (commonly referred to as free-exchange or passive heating/cooling systems). In the case where the entering fluid temperature is insufficient to effectively provide the load requirement, the fluid is routed to the heat pump unit where it is upgraded to the correct operating temperature. After the heat transfer fluid has been spent, it is rejected back into the well allowing for thermal regeneration of the working fluid.

Open loop systems are not as frequently installed due to the environmental regulations set on their implementation, which are meant to avoid the potential contamination and reduction of water quality [10]. Other limitations with this system type are the potential increase in maintenance costs due to mineral build-up, resulting in mechanical damage attributed to the use of natural spring water as the working fluid [11]. A schematic of a common open loop system configuration is illustrated in Figure 2, representing the heat pump assisted configuration.

Figure 2: Open loop groundwater heat pump system (reproduced from [8])

2.1.1.2 Closed Loop Systems

Closed loop systems are referred to as ‘fixed volume’ systems. These systems operate by utilizing a fixed volume of working fluid contained in an underground piping network. This configuration is typically more costly to implement due to the installation of the piping network, but is a more favoured arrangement. Unlike open loop systems, the popularity of the closed loop configuration is attributed to it being sealed, posing no significant risk of environmental contamination [5].

The ground-loop component of the closed loop GSHP system consists of high-density polyethylene (HDPE) pipes buried in the ground, with water or a water-antifreeze solution used as the working fluid [5, 12]. The two most commonly implemented ground-loop configurations are the horizontal and vertical heat exchangers, illustrated in Figure 3 and Figure 4, respectively. The horizontal ground-loop configuration is typically employed in installations that have limited

restrictions on available land area, where substantial trenching (proportional to the ground-loop length) is required to install the piping network. The typical installation depth for the ground-loop is between 4-6 feet, which provides stable ground temperatures and sufficient protection from extreme weather conditions [12]. The vertical ground-loop configuration is implemented when there is limited land area available, introducing unavoidable barriers for the geo-field installation. The vertical configuration is the most expensive ground-loop configuration to install due the costly rates associated with drilling [9].

Figure 3: Horizontally configured ground-loop system (reproduced from [8])

Figure 4: Vertically configured ground-loop system (reproduced from [8])

2.1.2 Deficiencies in Ground-Source Heat Pump Technology

The current economic viability of GSHPs and associated knowledge gaps rely on several factors such as: geographical location (weather patterns), control strategies, utility rates, and inflation rates. There is great potential for GSHP systems to improve energy conservation, as it relates to space heating/cooling, but due to high upfront costs and long payback periods, their ability to penetrate the market has been throttled. As a result, GSHP systems are best suited for large buildings with high thermal demands in order to overcome the associated capital investments [12, 13]. The long-term performance characteristics of GSHP systems have not been adequately characterized, thus the design and integration of future systems is critical to the success and viability of this technology. For example, an oversized system will result in longer payback periods and inadequate design has the potential for thermal saturation of the bore-field from long-

term operation. Figure 5 illustrates the ground becoming thermally saturated in the first few years of operation, after which a much more gradual change results as the temperature gradient in the soil adapts. The recuperation period occurs when the GSHP system is deactivated and the soil is given time to re-equilibrate.

Figure 5: Ground temperature change and recuperation (reproduced from [10])

It is evident that GSHP systems provide a sustainable method to attain our climate control needs, but their success is limited by their large financial commitment. The economic viability of these systems (compared to conventional units) depends heavily on the utility rates of natural gas and electricity [14]. As illustrated in the study performed by *Nguyen et al.* [14], the variability in the utility rates in North America have a substantial effect on the size of the resulting ground-loop for a GSHP system. Due to inflated electricity rates, GSHP systems are more favourable for cooling dominant builds and less favourable for heating dominant buildings, assuming natural gas is the fuel source utilized to provide space heating. Due to the current low price of natural gas, in most cases, it is not economical to install a GSHP system [9].

2.2 Conventional HVAC System Overview

In this section, four commonly implemented HVAC systems will be outlined to characterize their working principles. HVAC systems are composed of primary and secondary systems. Primary systems refer to the mechanical units that convert an energy source into thermal energy for heating/cooling applications, and the secondary system refers to the distribution system (DS). In this section, the interest is on the primary system characterization, for the following systems: boilers/furnaces and air-conditioners/cooling-towers (AC/CT).

2.2.1 Boilers and Furnaces

There are a variety of mechanical systems that are utilized for space heating applications; two of the most commonly implemented are boilers and furnaces. Boiler systems are commonly used in the industrial sector to produce steam (or hot water) for various applications, subject to a facility's needs. Depending on a plant's required heating capacity, boilers are conventionally arranged in a parallel configuration feeding a common header, allowing allocation of load to be distributed among boiler units. In the residential sector, furnaces are the favoured system, providing heating to an interior space by means of a working fluid, utilizing an air medium. It is noted that residential heating appliances that use water as the system's working fluid are commonly referred to as residential hot water boilers. Common boiler/furnace configurations are fire-tube and water-tube systems. Fire-tube systems burn a fuel source creating hot gas which travels through tubes submerged in a fluid reservoir, where heat is transmitted through the tube wall into the fluid. Water-tube systems operate in the reverse manner; a fluid (typically water) is circulated through tubes located in a combustion chamber where a fuel source is burned to release the thermal energy necessary to heat the fluid [15].

The operational principle of boilers and furnaces are the same. These systems utilize the heat released from the combustion of a fuel source to provide space heating (boiler/furnace) and hot water/steam production (boiler). Currently, natural gas is the favoured fuel source consumed by these systems due to its low cost and accessibility [16]. The main deficiency with these systems is that they rely on the energy released from the incomplete combustion of the fuel source; never fully utilize all of the potential energy stored in the fuel being consumed. The typical fixed capacity boiler/furnace is confined to produce efficiencies below 100%, in the range of approximately 75% - 90% [9, 10]. The efficiency of a boiler/furnace (variable capacity) is proportional to the systems firing rate, where low-fire relates to the lower end boiler efficiency and high-fire relates to higher end boiler efficiency. When sizing a boiler/furnace, it is critical to select a system with the appropriate capacity to avoid system short-cycling; short-cycling has been known to contribute to efficiency degradation of approximately 15% lower than the efficiency accomplished at low-fire. Short-cycling affects system efficiency because fixed losses are magnified under reduced load conditions. For example if the boiler/furnace system's fixed radiant losses at full-load account for 1% of total heat input, at half capacity this loss doubles. In addition, short-cycling contributes to an increased number of boiler/furnace pre-purge/post-purge cycles which removes heat from the systems envelope, increasing the amount of useful heat lost [17, 18].

2.2.2 Air-Conditioners and Cooling Towers

For space cooling applications, two of the most commonly implemented mechanical systems are ACs and CTs [17]. AC systems are regularly operated in both commercial and residential sectors. The operating principle of the AC is the same as the aforementioned heat pump vapor compression cycle (illustrated in Figure 1), composed of four major components: compressor, condenser coil, evaporator coil, and expansion valve. AC systems are also known as non-reversible air source heat

pumps because of their identical operating principle and common components, excluding the directional control valve. Hence, air source heat pumps (ASHPs) are capable of providing both space heating/cooling, whereas AC units are limited only to space cooling due to the mechanical restrictions (absence of directional control valve). AC units are commonly paired to forced-air distribution systems, arranged in a split-system configuration (SSC) or a self-contained unit. SSC designation was established to describe the placement of the refrigeration cycle's primary components, not located in a self-contained unit. For SSC ACs with a forced-air distribution systems, the evaporator coil is located in the air-handler unit (AHU), where the condenser coil, compressor, and expansion device are contained in a separate unit, commonly located outdoors [17, 18].

CT units are categorized into two system types, wet sump (evaporative) coolers and dry coolers. Wet sump CT systems are commonly utilized in large commercial buildings to provide heat rejection capabilities, cooling the condenser side of central HVAC chillers. Figure 6 presents an illustration of the working principle of a typical wet sump CT. These systems operate by pumping warm working fluid (water) from a heat source (chiller sink fluid or industrial equipment) to the top of the CT. The warm water is then evenly dispersed, typically by means of spray nozzles, over the CT fill. The CT fill spreads the water over a larger surface area, which increases evaporation and removes heat. Simultaneously, CT fans are engaged to induce air flow across the fill and accelerate evaporation process, further cooling the working fluid. The cooled water continues to travel down the CT's fill, into the sump and is circulated back through the working fluid network to cool the heat source. As the working fluid evaporates, only pure water is lost, resulting in a build of 'system elements' (suspended solids) due to the reduction of water volume. To account for the loss of volume, a make-up water line supplies fresh water into the cooling tower sump. Once the

concentration of system elements reaches the threshold limit, a bleed off valve opens to eject a portion of the systems water to the drain, which is replenished by the fresh water make-up. During CT evaporation, bleed-off and make-up are balanced to maintain the appropriate water volume, while creating a continuous open loop [17].

Figure 6: Wet sump cooling tower principle sketch (recreated from [19])

Figure 7 presents a descriptive schematic of the working principle of a dry cooler system. Different from an evaporative cooling tower, dry coolers rely solely on sensible heat transfer, whereas wet sump systems rely on both sensible and latent heat, due to the phase change of the cooling fluid. In essence, a dry cooler's working principle is the same as that of a fan coil unit; a high temperature fluid enters a finned-tube heat exchanger, where heat is rejected from the fluid to the surrounding thermal sink. This process is accelerated by a fan blowing air over the heat exchanger surface, increasing the rate of convective heat transfer due to the forced air-flow.

Figure 7: Descriptive schematic of a dry cooler (recreated from [20])

The major deficiency in both air-conditioners and cooling towers is that they both rely on the outdoor ambient air to act as a heat sink to reject the unwanted thermal energy from a process. It is evident that the outdoor ambient air is unstable in terms of both air temperature and humidity [12]. Peak cooling demands (for space cooling applications) occur during hours of suboptimal ambient outdoor air (high temperature and humidity). As a result, both ACs and CTs operate at higher temperatures to allow for effective heat rejection, producing lower operating efficiency [8].

2.3 Ground-Source Heat Pump System Hybridization

The growth in popularity of GSHP systems is currently throttled by the large capital investments associated to the implementation of this technology. Current rules-of-thumb utilized in industry do not correspond to optimized designs, as outlined by Alavy *et al.* [21] and Nguyen *et al.* [13]. A solution that can be implemented to alleviate the high upfront costs and long payback periods associated to this technology is referred to as a system hybridization. This method couples a GSHP system with auxiliary heating and cooling units, or in the case of a retrofitted installation, the

existing conventional heating and cooling systems are utilized [13]. Employing a hybrid system configuration provides the system's designer with flexibility in the sizing of the both the geo-exchange and conventional system components. With this flexibility, the HGSHP system can be designed to reduce the upfront costs and payback periods in a manner that is acceptable and appealing to investors. For hybrid systems, the GSHP unit is sized to meet a percentage of the total heating/cooling demand. When the system is undersized, in hours of peak demand, the auxiliary heating/cooling systems provide supplementary assistance, increasing the systems net capacity to meet the required load [22, 7]. In addition to HGSHPs, significant savings can be obtained through the integration of an improved control paradigm; with potential to reduce operating costs and payback periods, making this technology a more appealing investment [11, 23, 24].

GSHP systems typically achieve payback periods in the range of 5 to 15 years, but the upper limit can be further constrained to 10 years if design efforts are made to lower ground-loop requirements and/or a significant carbon tax is imposed [25, 26]. A payback period of five-years is typically considered a "threshold" payback for widespread market adoption of energy-saving technologies. Payback periods exceeding five years lead to limited market penetration; with payback periods of ten years or greater suggesting that the application is limited to niche markets [25]. To further the economic viability of geo-exchange technology, optimized design methods are necessary to reduce the capital costs requirements and improve the appeal of geo-exchange to a larger share of the renewable technology market [25].

Although the current hybrid geo-exchange design methods adopted in industry follow standardized sizing approaches, the use of simulation tools can improve the accuracy in assessing the energy consumption of hybrid subsystems and ensure a cost-effective design. A study conducted by Hénault *et al.*, [27] proposes a strategy to optimize the net present value of a HGSHP utilizing a

spectral-based simulation tool that predicts the heat pump performance on an hourly basis with an internal optimization algorithm. The analysis revealed that the net present value of a project was highly sensitive to the number of boreholes, operating temperature limit, and the number of installed heat pumps. In addition, the hybrid system configuration indicated significant benefits for reducing both electricity consumption and peak electrical load. Although optimization based design techniques are more complex, these methods clearly indicate benefits over the standardized sizing approach [27]. To improve the economic appeal of geo-exchange and reduce financial risk, the geothermal industry should adopt higher level integrated design approaches.

2.3.1 Benefits of HGSHS Systems versus Conventional Technology

When considering a building's heating requirements, the typical technology that is utilized to supply the required demand is a modern furnace/boiler. This system utilizes the thermal energy released from the combustion of a fossil fuel source, typically natural gas, for space heating and in some cases domestic hot water (DHW) production. For common systems in use today generate operating efficiencies of approximately 75%-90%, which can be translated to a coefficient of performance (COP) of 0.75-0.90 [9, 10]. Alternatively, GSHP systems are able to produce average COPs in the range of 3-5 [10, 8, 28, 16]. The primary deficiency of many conventional technologies is that they consume non-renewable fuel sources releasing the necessary thermal energy to meet a buildings heating demand. For example, a natural gas furnace releases the thermal energy through the means of combustion of the fuel source. From a sustainability perspective, this process performance is unsatisfactory because it fails to utilize all of the potential energy stored in the fuel source. In addition, due to the operating principle of these systems, their maximum theoretical performance is restricted to 100%, translating to a COP of 1. GSHP on the other hand move thermal energy stored below the earth's surface to provide space heating, consuming

electricity only when necessary to amplify the temperature of the working fluid to effectively meet a building's thermal demand. This sustainable method of utilizing useful thermal energy from below the earth's surface provides an effective and efficient means of space heating, resulting in high COPs.

When considering a building's cooling requirements, the typical technologies that are implemented to supply the required demand is modern AC/CT. Conventional systems such as AC units and CTs operate by utilizing the outdoor ambient air as a thermal sink to reject unwanted thermal energy from the zone being cooled. Due to the variability of the outdoor air temperature and humidity, these systems operate at higher temperatures and thereby produce lower energy efficiency ratios (EER). Overall, cooling can be provided more efficiently via HGSHP as opposed to conventional systems, due to minimal fluctuation in ground temperatures then experienced in the ambient air [12]. GSHP systems excel in cooling mode when compared to conventional systems, due to the fact that GSHPs are designed to effectively supply the required building demand at a lower operating temperature [10]. It is noted in the study conducted by Alzahrani [28], the COP of a GSHP system (in cooling mode) is at its maximum when the ground-loop working fluid is at its minimum temperature (equilibrated with the ground temperature).

2.3.2 GSHP Systems Coupled with Alternative Energy Source Devices

Hybrid energy systems facilitate the use of renewable and sustainable energy sources, and are expected to become more prevalent in the years to come. With GSHP systems at the core of the hybrid configurations, HGSHP systems have been used and studied frequently in the literature [29, 30, 31]. Hybrid systems are typically composed of the following energy source devices: solar collectors, boilers, electric heaters, waste energy devices, cooling towers, chillers, and natural hot/cold thermal storage [29].

The ground-loop component of a GSHP system has conventionally been used as a heat exchanger to transfer heat to/from the ground to accommodate a building's demand. However, due to high utility rates during peak hours of operation along with ground thermal imbalance (attributed to unbalanced operation and inadequate design), inflated operating cost and long payback periods can result. Recent studies have been conducted to assess the potential to couple thermal energy storage (TES) systems with GSHP technology. As studied by *Nguyen et al.*, [13] cooling dominant buildings located in warm climates make GSHP systems less appealing due to higher ground temperatures such that larger ground-loop lengths are required. As such, implementing a TES system such as ice storage is an option to counter the high operation costs during peak hours of operation [30]. Alternatively, when utilizing the ground's mass as a thermal battery, the ground can be cooled; the action of cooling can prevent thermal imbalance of the ground and thereby extends the mechanical life of the GSHP system [20]. In a study conducted by *Allaerts et al.*, [31], a HGSHP system is analyzed with a split bore-field configuration. The system's bore-field is partitioned into a warm and cold section, allowing for seasonal energy storage. In addition, auxiliary dry-coolers are used to store energy, storing heat during the summer and removing heat during the winter. The results of the analysis indicated that with the proposed hybrid configuration, a significant bore-field size reduction of 47% was achieved in a cost-optimal configuration. Additionally, the seasonal energy storage strategy introduced the ability to maintain/restore the bore-field's thermal profile when heating and cooling demands are unbalanced.

Geo-exchange systems are utilized by buildings in winter months by extracting thermal energy from the ground to provide space heating. Whereas in summer months thermal energy is rejected to the ground, providing space cooling. It is evident that from the operation of a GSHP system, there are adverse effects produced that contribute to thermal saturation of the ground's natural

temperature equilibrium. It is critical to the life expectancy of these systems that they are designed to balance the heat injection/extraction annually to reduce the potential for thermal saturation [10, 32]. On an annual basis, many large commercial buildings experience an imbalance in thermal energy storage due to their cooling-dominant characteristics [13]. With this imbalance present, the ground becomes thermally saturated, which results in the soil temperature varying from the design specifications. Consequently, thermal saturation limits the lifespan of GSHPs and degrades the system's overall performance. For buildings that have imbalanced heating/cooling demands, it is critical that supplementary systems are implemented to make use of the excess thermal energy that would otherwise contribute to the ground temperature change. For example, a desuperheater can be coupled to a GSHP, used to extract additional thermal energy to provide DHW. Also, hydronic in-floor heating/snow melt technologies can be coupled to the geo-exchange system to utilize the excess thermal energy stored in the ground, caused by the imbalance of the building's heating/cooling loads [8, 10]. A study conducted by You *et al.*, [33] proposed the coupling of a heat compensation unit with a GSHP to build a hybrid GSHP system. The integration of the heat compensation unit provided improved energy savings, allowing for direct DHW and space heating. The proposed hybrid configuration efficiently reduced energy consumption due to the multi-functional operation, producing 9.8% - 12.9% savings from heat compensation and DHW combined. The proposed hybrid system with multi-functional heat compensation provided significant energy savings potential while maintaining the underground thermal balance in cold regions.

Solar assisted ground-source heat pump (SAGSHP) systems have been proposed in the literature as an alternative hybrid system configuration, allowing for improved system efficiency by means of a bore-field TES operating strategy. During heating mode operation of a GSHP system, heat is

extracted from the soil which results in a soil temperature decrease as a consequence of long operating cycles. With the geo-exchange system's performance weighing heavily on soil temperature, SAGSHP systems allow for improved system efficiency in heating mode by a controlled increase in soil temperature. These systems operate in two modes; heating mode (supplying building thermal requirements) and solar thermal storage mode. The solar thermal storage mode uses a solar thermal collector to heat the bore-field working fluid during peak hours of solar radiation. The heated fluid circulates between the solar collector and the bore-field allowing the heat in the fluid to transfer to the bore-field/soil, effectively pre-heating the ground prior to a dwelling requiring heat [34]. Early research conducted by Bose *et al.*, [35] verified the potential for SAGSHP systems as an operational strategy for improving the geo-exchange system's energy efficiency. Since then many studies have been conducted simulating the thermal benefit brought forth by the SAGSHP system configuration. Nam *et al.* [34] simulated the performance benefits of a SAGSHP system with a predicted 28.1% and 9.3% increase in bore-field heat exchange rate and system COP, respectively. Verma *et al.* [36] conducted a SAGSHP system simulation revealing an 8.74% increase in in the solar assisted systems COP.

2.4 Control Methodologies for HVAC Applications

There are a variety of control methodologies that have been implemented in HVAC applications. Figure 8 illustrates the hierarchical classification of the various control methods implemented in HVAC applications. The classification of the various control techniques are sectioned into four categories: 1) classical control, 2) hard control, 3) soft control, and 4) hybrid/other control methods.

Figure 8: HVAC control methods flowchart (recreated from [4])

The original goal of control system design for building HVAC systems was primarily focused on occupancy comfort. The current expectation of building HVAC control places emphasis on both system performance as well as occupancy comfort. The increase in controller expectation and system restrictions has resulted in a multidimensional control task, which currently remains an open ended issue [37].

2.4.1 Classical Control

Classical controllers are the most commonly implemented system types in industry to date, due to their robust and simplistic principles of operation [4, 37]. As the favoured controller type, these systems are used for various applications in building automation systems, such as: room temperature control, supply pressure/temperature control, and primary HVAC system control [37]. As illustrated in Figure 8, the classical control category comprises discrete control and proportional-integral-derivative (PID) control.

Discrete control techniques are the most fundamental control methods utilized in HVAC applications. Discrete control is also referred to as “on/off” or “bang-bang” control, due to its two state operating principle. Classical “on/off” control is commonly implemented in thermostat controllers for set-point feedback management of a building’s HVAC equipment. To avoid frequent switching between the two states of the thermostat, a dead band is implemented. Dead bands are implemented in thermostats as an acceptable temperature deviation from the reference set-point, defining the amount of overshoot/undershoot permitted before a control action is made. The primary deficiency with this method is that overshooting in the control state is unavoidable, resulting in unnecessary energy consumption [37]. Research has been conducted utilizing discrete controllers as the foundation of the operation, coupled with set-back strategies to reduce energy consumption. Research conducted by Moon *et al.* [38] involved a discrete thermostat partnered with a nighttime temperature setback strategy; the analysis showed a significant impact on the heating energy consumption in a cold climate, where approximately 3% savings per degree Celsius was achieved. Occupancy predicting algorithms combined with the previous configuration was studied by Kleiminger *et al.* [39]. It was learned that energy consumption savings of 6-17% (on average) were achievable but there is a trade-off with the risk of comfort loss within the dwelling.

PID controllers are commonly implemented in HVAC system designs due to their ability to tune the system’s response, mitigating the resulting overshoot [37]. The implementation of PID control techniques produces promising results in energy conservation of HVAC systems, but tuning these controllers is a daunting task and system performance can degrade substantially if operating conditions vary from the calibration reference [4]. PID controller gains have an enormous impact on system energy consumption as well as the controller’s ability to maintain the set-point, potentially resulting in controller instability [40]. Wemhoff *et al.*, [40] illustrates the importance

of PID gain calibration, utilizing a simple calibration methodology to conduct successive optimizations on the three controller gains. The gain optimization process revealed a 29% reduction in energy consumption for the proportional gain, and a 35% savings in energy for the integral gain [40].

2.4.2 Hard Control

Hard control methods are well established techniques. The classification of these methods is based on control systems theory and contains the following methodologies: gain scheduling, nonlinear control, robust control, optimal control, and model predictive control (MPC) [4]. Only nonlinear control and MPC techniques are considered in this literature review.

Nonlinear control techniques cover a wide range of systems with a dynamic structure. This control methodology applies to more realistic problems; real-world control problems are nonlinear in nature. This method requires complex mathematical analysis in the design process to determine the stability of the system's state. If a system's state is near the stable point of interest, the system is linearized. Once linearized, linear controller design techniques can be directly applied to the corresponding system [4, 41]. A study conducted by Homod *et al.*, [42] adopted the use of a hybrid model using both physical and empirical modeling schemes to render a product that accurately represents a building and HVAC system. Although the research scope did not include the controller design, the foundation was established and the model resulted in a large number of measurable variables that can be controlled to obtain a strong transient and steady-state response.

MPC is a control technique that uses a developed system model to forecast the future state of a process. This control method allows for constraints to be set on the rates and limits of process state. MPC allows for both external and internal disturbances (such as occupancy, appliance use, and weather) to be accounted for in the estimated effects built into the controller [4]. The nature of

MPC control is a system that is robust to parameter variation and is capable of handling time-varying disturbances; capable of regulating a process tightly within predefined bounds. Afram *et al.*, [4] reviewed a variety control techniques, with an emphasis on the theory and application of MPC; comparing the performance of MPC with other control approaches. From [4], benefits of the MPC method have been established, such as: energy/cost reduction, peak load shift capabilities, improved regulation, and enhanced efficiency/COP. Further work done by Liang *et al.*, [43] contributed to the solidification of MPC as an alternative method for HVAC application; numerically simulating a potential 27% energy savings compared to the baseline controller implemented in an AHU.

2.4.3 Soft Control

The soft control category is classified into two methods: Adaptive Neural Network (ANN) control and Fuzzy Logic (FL) control, as illustrated in Figure 8. These two control methodologies are new and have been made feasible by the introduction of digital controllers [4]. Soft control methods provide many benefits over the previous control categories outlined, due to the introduction of model free adaptive control.

The ANN control sequence is a black box modeling technique in which the controller is trained on the performance data of a process, upon which a nonlinear mathematical model is fitted to the process data. A benefit of implementing an ANN controller is that it does not require any understanding of the control processes' underlying physics [44]. A drawback to this method is that to successfully implement this controller, training data on a wide variety of operating conditions is required. Process data may not be readily available for many applications; as a result, industry has been hesitant to adopt this black box design approach [4, 44]. Despite industry's lack of support in ANN control, there has been an ample amount of research conducted showing a strong potential

for this technique. A study by Gang *et al.* [45] proposed a new strategy to control a HGSHS system (GSHP running in parallel with a cooling tower) using an ANN predictive foundation. The study revealed that the ANN can always realize the process predications accurately, regardless of the operation mode. It was also revealed that the ANN predictive control strategy is more energy efficient than the other methods studied, resulting in the lowest end use consumption.

The FL control is a model-free design method providing superior results when compared to the frequently implemented classical control strategies. This controller is capable of conflict resolution from local level controllers, by prioritizing certain processes in order to accomplish a common goal [46]. The implementation of FL controllers is relatively simple, utilizing basic if-then-else logic statements to obtain the desired controller actions [4]. Adaptive FL (AFL, a FL foundation coupled with learning algorithms) controllers have the ability to self-regulate and adapt to the varying process state conditions. AFL controllers have been regarded as the most promising adaptive control system for buildings [4, 37]. AFL controllers have recently been integrated in a new generation of furnace control units that apply adaptive heating control to both maximize energy efficiency and thermal comfort for the residential sector [4, 46]. A recent study of FL control has been applied to the management of a photovoltaic/thermal (PVT) and GSHP hybrid system by Putrayudha *et al.*, [47]. This study implemented FL control to optimize system energy consumption, with a discrete controller acting as a baseline for comparison; results showed the FL controlled system reduced energy by approximately 20%.

2.4.4 Hybrid and Other Control Methodologies

Hybrid control techniques are a combination of both hard and soft control methodologies. This fusion was carried out as a means to address issues that arose in process control that would otherwise be unfeasible to overcome by the application of either technique separately. The reinforcement-learning (RL) technique is an example of a controller type that falls under the hybrid category. RL controllers are able to use information from system inputs/outputs to gain knowledge about the controlled process. RL is a model-free approach that learns from a process' previous action, making adjustments to increase the system's performance [4, 37]. A study conducted by Gracia *et al.*, [48] made use of a RL algorithm to control the operating schedule of a ventilated façade with a phase change material (PCM) in its air chamber. The control system for the ventilated PCM configuration was based on the weather forecast, which was used to optimally solidify the PCM to release the cold stored to the inner zone during hours of peak demand. This research discovered that the control system was sensitive to an accurate weather forecast, which resulted in an 18% variation in electrical energy savings between predicated and real weather data. A drawback to RL method is an ample amount of time is required to learn about the controlled process, making them impractical to implement in many applications [4, 37].

CHAPTER 3- Characterizing the Effect of an Off-Peak Ground Pre-Cool Control Strategy on Hybrid Ground-Source Heat Pump Systems

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Abstract

Geo-exchange systems are a sustainable alternative to conventional space conditioning systems due to their high operating efficiency, resulting in reduced energy consumption and greenhouse gas emissions. However, geo-exchange's ability to penetrate the market has been throttled by large capital investments, resulting in undesirable payback periods. Optimized HGSHP systems have been introduced as a remedy to overcome the current economic hurdles associated with the installation of geo-exchange systems. In both the literature, as well as in practice, there still remains potential for increased economic feasibility of this technology through integration of intelligent operational strategies. This manuscript presents a novel control methodology referred to as an off-peak ground pre-cool strategy, employing a TOU-conscious operating logic which artificially pre-conditions the system's bore-field. A reduction in peak power consumption is achieved by creating improved thermal characteristics during mid-peak/peak TOU operating brackets. A comprehensive numerical model was developed to characterize the operation of HGSHP systems for three real case studies. The model implements a base case set-point control scheme, used as a reference to assess the operational benefit of the proposed off-peak ground pre-cool control strategy. The preliminary analyses indicated potential for operational cost savings in all cases, under specific pre-cool scheduling. The strategy substantially contributes to a reduction in peak power consumption while increasing cooling supplied by the hybrid geo-exchange system.

3.1 Introduction

With building energy consumption on the rise, power conservation strategies have become a global priority in many energy usage policies. Space heating/cooling contributes to a large portion of a building's net energy consumption, typically accounting for 50% of annual usage [3, 4]. According to the U.S. energy Information Administration (EIA), the building sector consumed 47.6% of the total energy in the United States, as of 2012 [2]. With environmental awareness and potential resource shortages progressing to the forefront of concerns, an amplified need for sustainable alternatives will arise.

GSHP systems are an alternative means to conventional HVAC systems. Although geo-exchange technology provides a high-efficient of means of building climate control, its success has been limited due to a history of poorly designed systems and undesirable system economics. The current economic viability of GSHPs and associated knowledge gaps rely on several factors such as: geographical location (weather patterns and soil conditions), system control, utility rates, and inflation rates [12]. There is great potential for GSHP systems, but due to high upfront costs and long payback periods, their ability to penetrate the market has been throttled. The economics of geo-exchange systems (compared to conventional HVAC units) depends heavily on the utility rates of natural gas and electricity. The study performed by *Nguyen et al.*, [14] highlighted that the variability in the utility rates in North America have a substantial effect on the size of the resulting ground-loop for a GSHP system. As a result of incremental inflation of electricity rates, GSHPs are a favourable alternative for buildings with large cooling demands. GSHP systems show less financial incentive for buildings with large heating loads due to the current low cost of natural gas [26].

Current rules-of-thumb utilized in industry for system sizing do not correspond to an optimized design, as outlined by Alavy *et al.*, [13] and Nguyen *et al.*, [14]. Optimally sized hybrid systems are a remedy that can be implemented to alleviate the large capital investments associated with geo-exchange. This method couples a GSHP with an auxiliary heating/cooling system, or in the case of a retrofitted installation, the existing conventional heating/cooling systems are utilized [12, 13, 14]. In a hybrid arrangement the GSHP is sized to meet base loads, with the auxiliary systems providing supplementary heating/cooling during hours of peak demand [7, 9].

In addition to geo-exchange hybridization, an increase in operating efficiency can be obtained through the integration of improved control paradigm; resulting in the potential for a reduction in energy usage, operating cost, and greenhouse gas (GHG) emissions [11, 23, 24]. With increased controller expectations and operating restrictions on HVAC systems, a multidimensional task has been established, which remains an open ended issue. Geo-exchange and traditional HVAC systems commonly implement lower level control techniques due to simplicity and reliability [4]. Although classical methods are expected in industry because of their performance characteristics, they operate in a suboptimal manner and contribute to a restriction in system performance. Through the implementation of higher level control prefaced with intelligent operational strategies, there is potential for improved system efficiency and economics.

A performance assessment study [49], conducted by the Toronto and Region Conservation Authority (TRCA) of urban geo-exchange projects in the Greater Toronto Area, presented valuable insight on various research areas to further the potential of geo-exchange. It highlighted TOU based control strategies as an area for further research and development, indicating a potential to reduce electricity costs from 20-25% (excluding regulatory and distribution charges). Aside from the cost savings associated with TOU control, additional benefits are provided to utilities in the

form of electrical load leveling, which helps alleviate the pressure placed on the grid during peak hours [49]. The study presented by Carvalho *et al.*, [50] conducted out of Coimbra, Portugal, proposed a TOU-conscious demand side management (DSM) strategy; using a GSHP as a flexible load to artificially consume energy in off-peak periods to pre-heat a service building. The thermal response of the building was analyzed through numerical simulation, which was validated by experimental results. The building pre-heat method allowed for a portion of the GSHP's operating cycles to be isolated within off-peak brackets, taking advantage of lower electricity rates; resulting in a 34% reduction in electricity costs [50].

The aim of this chapter is to perform a preliminary analysis of a proactive control strategy for cooling mode operation of HGSHP systems. Referred to as the 'off-peak ground pre-cool' control strategy, this technique utilizes the auxiliary cooling system as a flexible load to artificially consume electricity during off-peak TOU brackets when energy costs are most economical. This analysis aims to demonstrate that a HGSHP system operated with an OGPC strategy can exploit the bore-field's/ground's thermal mass with a pre-cool, creating improved thermal characteristics during the following mid-peak/peak periods. When regular cooling mode operation commences, with the aid of the pre-conditioned bore-field, the GSHP efficiency will increase as a result of a reduction in local ground temperatures. Introducing a pre-conditioned bore-field helps to reduce peak energy usage, due to an increase in the geo-exchange system's cooling output, accomplished at a higher degree of efficiency. With a TOU-conscious operating strategy, the proposed methodology will not only address improving system economics through an increase in operating efficiency, but also presents a multifaceted approach which intends to concurrently aid the balancing of the electrical grid.

The analysis in this chapter was accomplished by use of a newly developed numerical performance prediction model, characterizing the operation of a HGSHP system. The model was used to simulate the response of the hybrid geo-exchange system when controlled by two different methods, being; a base case set-point scheme, and an OGPC control strategy. Numerical simulations were conducted to predict the impact the proposed strategy has on electrical energy consumption, annual operating cost and peak power reduction for three pre-cool operating schedules; shoulder, peak, and full season. A comparative analysis was performed for three real buildings located in Toronto, Canada.

3.2 Methodology

The methodology applied in this paper consists of a three-part procedure, which is illustrated in the flowchart diagram presented in Figure 9.

Figure 9: Flowchart of proposed methodology

First, building energy simulations (BES) were conducted to generate estimates of various building's annual hourly heating and cooling loads. Once the estimated annual thermal loads are processed, they are utilized as an input variable to design an optimally sized HGSHS system, following the rigorous computational approach outlined in Alavy *et al.*, [13]. After the BES and hybrid design procedure have been completed, numerical simulations are conducted with the aid of a performance prediction model that has been newly developed to quantify the proposed OGPC control strategy effect on the overall performance. During the BES stage of this research the building thermal loads were determined using eQuest [51], with the remaining design and numerical modeling being carried out using MATLAB [52]. The loads generated from the eQuest simulations are used in both the HGSHS system design procedure and as an input to the numerical model. The following sub-sections provide a detailed outline of the numerical model process logic and the representation of the presented control scheme.

3.2.1 Building Loads and Hybrid System Design

In this study, annual hourly thermal loads associated with the various buildings cases were generated in building energy simulations using eQuest. Positive loads indicated a space cooling demand, whereas negative loads represented space heating demand. The energy simulation initially allows for both heating and cooling to be supplied in a simultaneous fashion during each time interval. However, the heating and cooling loads are corrected under the assumption that each building's demand can be satisfied with an internal heat distribution mechanism before relying on the compensation from the geo-exchange system. In the analysis a common water loop distribution system has been assumed, allowing for internal load compensation to occur. For example, if an operating time step has a large cooling demand and a small heating demand; the heating demand can be met by removing heat from the zone requiring cooling, and into the zone requiring heating.

By neglecting the power consumption of the internal mechanisms, the net demand will be provided by the HGSHP to/from the common water loop distribution system. For the purpose of this analysis, three real buildings were examined, as highlighted in Table 1.

Table 1: Building information

Building Info	Sector
1. Mid-rise, multi-residential	Residential
2. High-rise, multi-residential	Residential
3. School	Educational

The parameters used in the HGSHP optimal design process were held constant with the analysis conducted in [13, 14]. The configuration of the ground-loop is a closed-loop system with a single U-tube vertically-oriented heat exchanger in each borehole, with a bore-field aspect ratio of 11 across by 4 down. The following Table 2 provides a summary of the parameters utilized in the design process.

Table 2: Design parameter summary

Parameters	Value
Cooling design entering sink temperature	25.1 °C
Heating design entering source temperature	1.7 °C
Heat Pump	ClimateMaster Tranquility TT (COP=3.1, EER= 13.6)
Soil thermal conductivity	2.94 W/m K
Soil thermal diffusivity	0.072 m ² /day
Duration of operation	20 years
Borehole thermal resistance	0.136 m K/W
Pipe resistance	0.06 m K/W
Pipe size	32 mm
Borehole diameter	127 mm
Grout thermal conductivity	1.47 W/m K
Number boreholes across	11
Number of boreholes down	4
Inflation rate	4%
Interest rate	8%

The heat pump selected for the hybrid system is the ClimateMaster Tranquility Series TT with an assumed fixed COP of 3.1 and EER of 13.6. For the HGSHS system design the ClimateMaster Tranquility TT heat pump is considered, with a dual stage heat pump unit capable of full load and 67% part load operation. A dual stage heat pump was selected in this study after consulting with GeoSource Energy Inc. personnel [53], stating that this is the most commonly encountered HP configuration in the Southern Ontario, Canada market.

The auxiliary cooling system selected is a non-revisable air source heat pump (water-air) with an assumed fixed EER of 10.8, selected from the experimental study conducted in [54]. The auxiliary heating system is a natural gas furnace, with an assumed fixed efficiency of 78%, held constant with the design specifications outlined in [14]. The following capital cost parameters were held constant with analyses conducted in [13, 14] and are summarized in Table 3.

Table 3: General capital cost of hybrid system components

Item	Cost
Ground heat exchanger (install and materials)	\$65.5/m
Auxiliary cooling system, plate heat exchanger including controls and auxiliary equipment	\$14/kW of auxiliary cooling design capacity
Auxiliary heating system	\$20/kW of auxiliary heating design capacity

3.3 Numerical Model

In this study, in order to simulate and analyze the off-peak ground pre-cool control strategy, a numerical model was developed using MATLAB to characterize the operation of a HGSHP system (presented in Appendix C). The model was developed in a modular manner allowing for simple manipulation of the system's control algorithms, providing flexibility for modification and study of the proposed control techniques.

The numerical model was developed to simulate a HGSHP system (GSHP and auxiliary HVAC systems) operating over the course of a year. A yearly operational analysis was selected over a commonly used 20 year life cycle because the annual savings of the proposed control scheme were being assessed, which was more clearly realized with the selected time frame. The numerical model simulates a load-based analysis determining which mechanical system has operational

authority in each simulated time step. The model runs a yearly simulation with a time-step of 20 minutes for a total of 26,280 iterations.

The numerical procedure is presented in the flowchart schematic in the following Figure 10. When the algorithm is initiated, all input parameters are specified, such as annual hourly heating/cooling loads (eQuest BES), optimal HGSHS system specifications (design procedure from [21]), and time-of-use electrical cost structure. The input data is then processed to determine important characteristics of the buildings thermal requirements, such as total heating/cooling demand required, peak heating/cooling loads, and whether the building under consideration is heating or cooling dominant. The capacities of the GSHP and auxiliary systems are then determined based on the optimal shave factor (α), defined as the percentage of peak cooling demand met by the geo-exchange system, with the remaining load ($1 - \alpha$) being supplemented by the auxiliary systems. A numerical simulation of an optimally sized HGSHS system operating for one year is then initiated, with a set-point control algorithm fitted to the model to determine the operational authority of the sub-systems depending on the building's thermal requirement in the current time-step. For every time-step of the numerical simulation the portion of demand supplied by the geo-exchange system and the auxiliary systems are determined; the variation in the GSHP COP, and the bore-field average temperature response is predicted simultaneously, with this function being explained later. Once the simulation expires, a data post-processing procedure is carried out to determine the resulting performance of the HGSHS system over a year of operation; upon completion of this process the program is terminated.

Figure 10: Process flowchart of the numerical performance prediction model

3.3.1 Variable COP Characterization

In [28], Alzahrani conducted a detailed experimental performance analysis of a GSHP system coupled to both a vertical and horizontal ground-loop at the TRCA Archetype Sustainable Twin Houses located in Vaughan, Canada. An observation made in this study highlighted that a heat pump's COP varies substantially with respect to system cycle time. From the experimental data collected, a clear correlation was represented; as the cycle on-time increases, the heat pump COP decreases, observing that the GSHP experiences peak COPs during the early transient portion of the on cycle [28]. This occurrence is attributed to the ground-loop working fluid having thermally equilibrated with the temperature gradient in the bore-field/soil. For example, cooling mode operation in the early transient stage of an on-cycle, assuming the ground-loop working fluid has equilibrated with the soil temperature, results in a low inlet sink fluid temperature to the heat pump.

As the cooling cycle progresses the geo-exchange system transitions toward steady-state operating conditions (increasing inlet sink fluid temperature), causing the variation in the system's COP to stabilize.

A GSHP's cycle time plays a vital role in system performance; as the cooling cycle duration increases and entering sink temperature increases, resulting in greater heat pump compensation to meet load requirements. Utilizing the experimental data collected by Alzahrani in [28], a variable COP correlation was generated to characterize the GSHP system's COP as a function of the on-cycle duration. The following Figure 11 is an illustration of the trends that occur for varying system COP, as a function of cycle time.

Figure 11: COP variation with respect to GSHP system cycle time (recreated from [28])

The numerical model employs COP ranges of 4.90 - 3.98 in cooling mode and 3.66 – 2.64 in heating mode. It is evident from the COP ranges, the GSHP system is more efficient in cooling mode when compared to heating mode. The higher efficiency is due to smaller temperature difference requirements between entering load and sink temperatures to meet thermal demands.

A thermal continuity condition was assumed upon integration of the presented performance correlations. This continuity assumption implies that the ground-loop fluid has a thermal recharge time that is equal to the previous on-cycle duration. For example, if the GSHP is operating for one on-cycle time-step, it takes one off-cycle time-step for the two media to reach steady-state. Thus when the heat pump's on-time is equal to the systems off-time the ground and ground-loop working fluid is assumed to have thermally equilibrated.

3.3.2 Time of Use Rates

The 2015 TOU utility structure from Toronto, Canada was adopted as a means to estimate the cost savings associated with the application of the OGPC control technique. TOU brackets are used as a primary control variable in the OGPC algorithm in an attempt to reduce peak power consumption to ease pressure on the grid during high-demand hours and decrease annual operating cost. The following Table 4 presents the TOU brackets and electricity rates used in the numerical simulations presented in this study.

Table 4: TOU electricity rates Ontario 2015 [55]

Period Type	Rate (¢/kWh)	Summer Periods	Winter Periods
Peak	16.1	11 a.m. – 5 p.m.	7 a.m. – 11 a.m. 5 p.m. – 7 p.m.
Mid-Peak	12.2	7 a.m. – 11 a.m. 5 p.m. – 7 p.m.	11 a.m. – 5 p.m.
Off-Peak	8.0	7 p.m. – 7 a.m.	7 p.m. – 7 a.m.

The Ontario Energy Board (OEB) classifies two peak seasons; summer weekdays ranging between May 1st to October 31st, and winter weekdays ranging between November 1st to April 30th. For weekends and holidays, an off-peak rate is designated for the full day [55].

3.3.3 CO₂ Emissions

The total annual CO₂ emissions associated to the operation of a HGSHP system are dependent on the electricity consumed by the GSHP, the auxiliary systems, and the CO₂ emissions factors for the local electricity supply mix. The electricity supply mix breakdown used in this analysis corresponds Ontario's power generation data for 2015. Over the 2015 year, Ontario's average supply mix is categorized into the following sources: 60.56% from nuclear, 23.28% from hydroelectricity, 9.96% from natural gas, 5.77% from wind, 0.10% from solar, and 0.23% from biofuel [56].

In this analysis, the 2015 hourly electricity supply mix for Ontario is used to produce an average hourly CO₂ emissions factor. In each numerical time-step, the supply mix factors for the aforementioned energy sources are determined. The product of each supply mix factor and their corresponding fuel type emissions factor is computed. The summation of the product of each supply mix factor and corresponding fuel type emissions factor produces an average hourly CO₂ emissions factor. This process is iterated 8,760 times, corresponding to each hour of the 2015 year. The result of this computation is used to predict the amount of CO₂ emitted (kg), for the operation of the HGSHP system. Utilizing an hourly average CO₂ emissions factor in the computation allows for the ability to evaluate the potential impact that the proposed peak power shaving strategy poses on annual emissions for the three building cases. With the efficiency of power generation depending on electrical demand (varying with the TOU operating brackets), there is potential for reducing emission associated to operating the HGSHP system.

In the present study, the CO₂ emissions factors are in terms of kg-CO₂ per kWh (kg-CO₂/kWh) of electricity consumed. The emissions factors used in this analysis are 0.002 kg-CO₂/kWh for nuclear power [57], 0.023 kg-CO₂/kWh for hydroelectricity [57], 0.553 kg-CO₂/kWh for natural gas [58], 0.002 kg-CO₂/kWh for wind power [57], 0.035 kg-CO₂/kWh for solar power [59], and 0.0261 kg-CO₂/kWh for biofuel [60]. The biofuel emissions factor is determined by taking the average for wood chips (0.0149 kg-CO₂/kWh) and wood pellets (0.0373 kg-CO₂/kWh), assuming equal mix of the two sources [60].

3.3.4 Off-Peak Ground Pre-Cool Control Strategy

HGSHP systems are commonly connected either in a parallel or series configuration, depending on system operational requirements [61]. The proposed system orientation required to implement a ground pre-cool is illustrated in a system schematic in Figure 12.

Figure 12: Hybrid ground source heat pump system schematic

The assumed hybrid geo-exchange configuration uses a series connection of the auxiliary cooling system with the ground-loop. Utilizing this configuration allows for three operating loops, through the manipulation of four three-way-two-position directional control valves (V1 – V4). Loop one is activated under base load operating conditions, when the GSHP unit is the only active load coupled with the bore-field heat exchanger (BHE). Loop one is the primary circulation loop, accomplished when V1 – V4 are in their rest state, allowing for undisturbed BHE fluid flow. Loop two is activated under peak operating conditions, when the GSHP and auxiliary cooling systems are both active loads, interacting with the BHE to meet the building's thermal demand. Loop two is accomplished by actuating V1 and V2 into their flow diverting state, allowing for the BHE fluid to be circulated through the auxiliary ASHP's evaporator. Loop three is activated for OGPC operation, with system activity depending on the selected auxiliary technology implemented. For this study, a non-reversible air source heat pump is selected; loop three would result in the ASHP being the only active load interacting with the ground heat exchanger. Loop 3 circulation is accomplished by actuating V1 – V4 into their flow diverting state, resulting in the decoupling of the GSHP from the BHE circuit. Alternatively, a cooling tower can interface with the GSHP operating in heating mode to reject heat from the refrigerant entering the condenser to the outdoor air, permitting ground pre-conditioning.

The primary control technique used in the operation of the HGSHS system is a conventional set-point control scheme. A conventional set-point control strategy is used in this study as a base case scheme to compare system performance benefits that an OGPC has on the operation of a HGSHS system. The OGPC algorithm's operational authority is restricted to the control of only the auxiliary cooling system, under the condition that specific logic indicators are satisfied.

The preliminary stage of the OGPC strategy utilizes two control variables to determine the beginning of an OGPC cycle, the control variables being: time of the year and TOU operating bracket. The time of year was utilized as a control variable in the OGPC algorithm to permit flexibility in pre-cool operation, allowing for time spans of various cooling load densities to be targeted by the algorithm. In this study, three operating schedules were analyzed; ‘shoulder season’ (SS) operation (April 1st – May 31st), ‘peak season’ (PS) operation (July 1st – August 31st), and ‘full season’ (FS) operation (April 1st – September 30th). The TOU control variable is a critical component in the OGPC algorithm where a pre-cool operation is strictly restricted to off-peak TOU brackets, taking full advantage of low electrical energy prices.

The following Figure 13 presents a flowchart diagram outlining the OGPC control logic. For every iteration of the numerical model, the OGPC algorithm evaluates the state of the two control variables. First the algorithm considers the operating day of the year, attempting to classify the current time-step into one of the three proposed pre-cool schedules; represented by the first three decision blocks (FS, SS, and PS schedule) illustrated in Figure 13. If none of the three decision blocks are satisfied, the HGSHP system maintains primary operation without the use of the OGPC strategy. Once the algorithm determines that the current time-step satisfies the requirements of one of the three proposed schedules, the next control variable is evaluated. The second control variable considers the hour of operation, attempting to classify the current time-step into one of the three TOU operating brackets; represented by the fourth decision block in the logic flowchart presented in Figure 13. The fourth decision block in Figure 13 indicates a time range from 19:00-7:00 hours, which refers to an off-peak TOU bracket for the adopted utility rate structure presented in Table 4. When both logic conditions are satisfied; day of the year (either SS, PS, or FS operation) and an off-peak TOU time bracket, a pre-cool cycle is initiated. First the bore-field circulation loop 3 is

activated, represented by the first process block in Figure 13; resulting in the GSHP being decoupled from the ground-loop, with the only active load being the auxiliary ASHP. The auxiliary cooling system is then initiated to operate at full load for a pre-cool cycle; represented by the second process block in Figure 13. The pre-cool operating duration (Δt) is varied to evaluate the impact it has on the HGSHP system's performance.

Figure 13: Flowchart diagram of simplified pre-cool control logic

To characterize the bore-field/ground's thermal response to an OGPC, the results presented in the study conducted by Nam *et al.*, [34] were used. In [34], a performance predication model was developed to analyze a solar-assisted GSHP (SAGSHP), using TRNSYS [62]. In [34], a solar thermal collector was used to implement a ground pre-heat thermal storage strategy to enhance the performance of the GSHP system. The numerical model for the vertical U-tube ground exchanger (DST model) that was used in [34], was developed at the Lund Institute of Technology in Sweden [63] and later integrated into TRNSYS by Pahud and Hellström [64]. From the results presented in [34], four correlations were developed to characterize the bore-field thermal response to heat injection/extraction, GSHP cooling output and COP variation in response to inlet sink temperature change (presented in Appendix D). The correlations were developed in the form of linear regressions based on the trends observed in the experimental correlations presented in [28].

The present model simulates two assumed physical responses to the thermal benefit resulting from the OGPC; an increase in the rate of heat rejection to the ground during cooling operation (increase in cooling output) and an increase in the GSHP system COP (decreases in compressor compensation). The increase in cooling output is due to the reduction in ground temperature, resulting in an average increase in heat injection per meter length of the ground-loop. The increase in heat pump COP in cooling mode is due to a lower inlet sink temperature, resulting in the reduction of compressor power consumption. These assumptions were validated by the studies presented in both [34] and [28], which provide both an experimental and numerical confirmation.

The presented numerical model predicts the GSHP system's performance by means of Eq. 3.1 and Eq. 3.2, in which Q_{GSHP} and COP_{GSHP} are the resulting heat output and system efficiency, respectively. In Eq. 3.1, the product of α and CL_{max} represents the base load supplied by the GSHP (system's cooling capacity), where CL_{max} is the peak cooling load, and α is defined as the shave

factor, which is the percentage of the peak load met by the GSHP. The ΔQ_{out} term is the incremental increase in the system's rate of heat exchange with the ground when exposed to a pre-cooled bore-field. The magnitude of the ΔQ_{out} increase is predicted by the aforementioned cooling output correlation, presented in Appendix D.

$$Q_{GSHP} = \alpha \cdot CL_{max} + \Delta Q_{out} \quad (\text{Eq. 3.1})$$

In Eq. 3.2, the $COP_{\Delta t}$ terms represent the variation of the GSHP efficiency during transient operation, which is characterized by the cooling curve presented in Figure 11. The ΔCOP term represents the incremental increase in system efficiency when exposed to a pre-cooled bore-field. The magnitude of the ΔCOP term is quantified by the aforementioned COP correlation generated from the results presented in [34], presented in Appendix D.

$$COP_{GSHP} = COP_{\Delta t} + \Delta COP \quad (\text{Eq. 3.2})$$

Eq. 3.3 defines the auxiliary system's cooling capacity (Q_{AUX}), sized to provide supplementary cooling under peak operating conditions. The proposed model simulates the auxiliary cooling unit as a single-stage fixed capacity system.

$$Q_{AUX} = (1 - \alpha) \cdot CL_{max} \quad (\text{Eq. 3.3})$$

Eq. 3.4 introduces the variable f_{PC} , which is the auxiliary system performance correction factor. The magnitude of f_{PC} is determined by implementing the step function in Eq. 3.5. The f_{PC} factor has a value of 1.0 when the simulated system is operating within a mid-peak to peak TOU bracket, or a value of 1.5 when operating within an off-peak TOU bracket. A performance correction factor has been introduced to modify the auxiliary system's performance when operating during an off-peak TOU bracket to account for improved operating conditions. It is assumed that there is a 50%

improvement in auxiliary system efficiency (COP_{AUX}) when operating in an off-peak TOU bracket, when compared to mid-peak and peak durations. This assumption was integrated into the model because the average daily maximum - minimum dry bulb temperature difference for a cooling season in Toronto was 8 °C; the average difference in maximum - minimum relative humidity is approximately 30% [65]. It is noted that in a cooling season, maximum temperature and humidity typically occur in a peak TOU bracket, while minima occur during off-peak durations. This assumed off-peak performance increase (for the auxiliary cooling unit) is an additional sensitivity parameter in the presented model. It was assumed constant and was not varied for the following studies.

$$COP_{AUX} = f_{PC} \cdot COP_{AUX} \quad (\text{Eq. 3.4})$$

Where,

$$f_{PC} = \begin{cases} 1.0, & t \rightarrow \text{mid} - \text{peak or peak} \\ 1.5, & t \rightarrow \text{off} - \text{peak} \end{cases} \quad (\text{Eq. 3.5})$$

3.3.5 Summary of Assumptions

This section provides a summary of the assumptions integrated in the numerical model, newly developed in this thesis (presented in Appendix C). The summary is as follows:

Variable COP Characterization:

- The experimental data presented in [28], characterizing the variation in GSHP COP (heating and cooling mode) as a function of cycle time was assumed valid as a baseline for model development.

- From [28], second order polynomial regressions were fitted to the experimental data set, presented in Figure 11, and used in the numerical computation.
- The COP of the GSHP system is averaged over the numerical time-step. The variation in COP is integrated in a discrete manner (refer to COP module in the model presented in Appendix C) with system performance ranging from 4.90 – 3.98 and 3.66 – 2.64, in cooling and heating mode, respectively.
- The variation in system COP is a function of cycle time, when on-cycle time is equal to off-cycle time, the COP is assumed to have reset to its maximum value.

HGSHP System Performance with OGPC Operation:

- Two physical responses are assumed when the GSHP is exposed to a pre-cooled bore-field; an increase in GSHP cooling output and COP.
- Utilizing numerically predicted data in [34], the correlations presented in Appendix D are assumed relationships used to predict the impact of the OGPC strategy.
- The increase in GSHP cooling output (ΔQ_{out}) is predicted using the correlation presented in Figure 34, Appendix D.
- The increase in GSHP COP (ΔCOP) is predicted using the correlation presented in Figure 35, Appendix D.
- The variation in the auxiliary cooling system COP is assumed 50% greater in off-peak operating brackets, versus mid-peak/peak periods.

3.4 Results: Characterizing the Effect of an OGPC Control Strategy on HGSHS Systems

The numerical performance prediction model developed in this research considers full seasonal operation (both heating/cooling). However, only cooling season operating was analyzed, as the proposed OGPC strategy's scope is limited to improving system performance in cooling mode. The results from this study are presented as follows: building load characteristics/ hybrid system specifications, base case control strategy, shoulder season schedule, peak season schedule, and full season schedule.

3.4.1 Building Loads and Hybrid System Specifications

In this study, three buildings were considered: a mid-rise multi-residential facility, high-rise multi-residential facility, and a school. Each of the buildings utilized in the research are real, and either had a GSHP installed or considered [13, 14]. The following Figure 14 illustrates the predicted building thermal loads as a result of the BES conducted using eQuest. In Figure 14, the cooling demand is presented as a positive load (light gray lines) and the heating demand is presented as a negative load (dark gray lines). The three presented building cases are all cooling dominant facilities, where the peak demand for cooling is greater than the heating requirement. Both the mid-rise (Figure 14a) and high-rise (Figure 14b) facilities require substantially greater amounts of cooling than heating, whereas the school's (Figure 14c) total cooling and heating requirements are comparatively even. In Figure 14c, the unique cooling demand distribution is attributed to the specific occupancy pattern distinctive to an educational institution, with a reduction in occupancy occurring in peak summer months (end of June to the beginning of September).

Figure 14: Predicted thermal loads for buildings (a) mid-rise, (b) high-rise, and (c) school

With the hourly loads presented in Figure 14, the design methodology presented in [13] was conducted to optimally size the hybrid geo-exchange systems for the three presented cases. Table 5 presents a summary of important details of the building thermal characteristics and hybrid geo-exchange systems specifications for the mid-rise, high-rise, and school buildings. The shave factor parameter summarized in Table 5 refers to the optimal specification. In the hybrid design procedure, the shave factor (α) refers to the percentage of peak cooling demand met by the GSHP system. As a result, this parameter is based solely on peak cooling required by the three building

cases. Referring to Table 5, it can be seen the annual total heating demand (THD) to total cooling demand (TCD) ratios reveal that neither the mid-rise nor the high-rise buildings are thermally balanced requiring four times the amount of cooling as they do heating, with the school being relatively balanced.

Table 5: Summary of building/hybrid system characteristics for a mid-rise, high-rise, and a school building

Building	Peak Cooling Load (kW)	Shave Factor (α)	Ground-Loop Length (m)	THD (MWh)	TCD (MWh)	THD/TCD
Mid-Rise	274.5	0.23	2115.5	143.0	573.7	0.25
High-Rise	568.4	0.21	3920.5	315.7	1212.2	0.26
School	174.0	0.21	791.9	295.9	252.1	1.17

3.4.2 Base Case Set-Point Control Strategy

The numerical performance prediction model was fitted with a standard set-point control strategy to develop a baseline result with which to compare the proposed OGPC operating strategy. The base case strategy utilizes the GSHP as the primary mechanical system, meeting base load demand, and is only assisted by the auxiliary cooling system when the demand exceeds the GSHP system capacity. Table 6 represents the predicted hybrid system performance over a year of operation when the system is controlled with a standard set-point control scheme. The annual cooling cost (ACC) is given in Canadian dollars, the annual electrical consumption for cooling (AECC) is presented in terms of megawatt hours, and the annual carbon emissions (ACE) is presented in terms of kilograms of CO₂. The total cooling demand met (TCDM) is given in terms of a percentage, representing the portion of cooling load supplied by the GSHP system. In Table 6 it can be seen that the under-sized hybrid geo-exchange systems are still capable of meeting a large portion of the total cooling requirement of each of the three presented buildings.

Table 6: Hybrid system performance for a year of operation with a set-point control strategy

Building	ACC (\$)	AECC (MWh)	ACE (kg-CO ₂)	TCDM (%)
Mid-Rise	20,381	185.4	12,694	41.7
High-Rise	48,127	423.3	29,231	35.9
School	8,791	75.8	5,374	34.5

In Table 7, a summary of the energy consumption distributed over the TOU operating brackets is presented for the three building cases. It can be seen that the mid-peak and peak hours of consumption account for a 48.5%, 55.5%, and 58.7% of total yearly energy consumption from space cooling for the mid-rise, high-rise, and school, respectively. It is noted that the higher consumption occurring in the off-peak TOU bracket is attributed to inclusion of weekend/holiday operation (categorized in the off-peak cost bracket).

Table 7: Summary of power consumption in TOU brackets, for a year of cooling operation with a set-point control strategy

Building	Off-Peak (MWh)	Mid-Peak (MWh)	Peak (MWh)
Mid-Rise	95.4	44.6	45.3
High-Rise	188.4	122.1	112.8
School	31.3	22.5	22.9

3.4.3 OGPC Control Strategy

To characterize the effect of the OGPC control strategy on the presented hybrid geo-exchange systems, a primary analysis was conducted in which 36 different annual simulations were carried out. In these analyses only the pre-cool duration is being varied; the simulations were conducted with 20 minute increments of pre-cool time, up to a maximum of 12 hours (full off-peak operating bracket). Upon completion of this study, the OGPC strategy was analyzed to determine the peak power reduction potential resulting from the introduced thermal benefit of the pre-conditioned bore-field.

3.4.3.1 Shoulder Season Operation

The simulated results for OGPC control strategy restricted to SS operation are presented in Figure 15 and Figure 16. Figure 15 presents the variation in AECC with respect to pre-cool duration, and Figure 16 illustrates the variation ACC as a function of pre-cool duration. Each data point represents an annual cooling season simulation, with the horizontal line acting as a datum reference for the same system operating without an OGPC scheme. Note that the fluctuation in the simulated data for both AECC and ACC occurs because of the simplicity of the set-point control algorithm operating the hybrid geo-exchange, resulting in set-point overshoot.

Figure 15: AECC for SS operation versus pre-cool duration for buildings (a) mid-rise, (b) high-rise, and (c) school

Figure 16: ACC for SS operation versus pre-cool duration for buildings (a) mid-rise, (b) high-rise, and (c) school

The impact that the OGPC strategy has on the three present cases is clearly characterized in Figure 15. All three cases depict a similar trend, in which energy consumption increases for small periods of bore-field pre-conditioning. After reaching a peak in annual energy consumption, all three systems begin to experience the thermal benefit imposed by longer periods of pre-cool. The mid-rise (Figure 15a) and high-rise (Figure 15b), required longer periods of pre-conditioning before a positive impact is realized on system energy consumption; approximately seven and six hours,

respectively. When analyzing Figure 15c it can be seen that the school required approximately three hours of pre-cool before the system's operation conditions were improved enough to reduce energy consumption. For all three cases, the greatest impact on energy consumption occurs at approximately 12 hours (maximum pre-cool duration), resulting in minor increases in AECC: 1.0%, 0.9%, and 0.3% for the mid-rise, high-rise, and school, respectively.

The economic impact of the proposed OGPC strategy is illustrated in Figure 16. Implementing the TOU structure presented in Section 3.3.2, the annual cooling cost was determined for the three presented cases operating within a shoulder season period. Both the mid-rise (Figure 16a) and high-rise (Figure 16b) required a minimum pre-cool duration of approximately nine hours to break even with the operating cost associated with the base case control scheme. The school (Figure 16c) showed financial gain for a pre-cool duration of only five hours. The largest ACC savings accomplished for all three buildings were realized for a full 12 hour pre-cool, accomplishing ACC savings of: 3.7%, 3.6%, and 7.2% for the mid-rise, high-rise, and school, respectively.

The cost saving potential illustrated in Figure 16 is accomplished by the proposed methodology's ability to shift a percentage of mid-peak/peak power consumption into off-peak TOU brackets, taking advantage of the low electricity cost. Table 8 presents a summary of the mid-peak/peak power reduction potential for the proposed shoulder season OGPC, showing a significant reduction potential which contributes to a more evenly distributed electrical consumption profile. When evaluating the ACE for each building case, there is no added benefit from an emissions perspective, resulting in a minor increase in annual CO₂ emitted. In addition, the pre-conditioning technique allows for an artificial increase in cooling output from the GSHP, meaning a larger percentage of the cooling demand can be met by the geo-exchange system. The simulations indicate a significant

increase in TCDM by the GSHP, resulting in an additional 11.9%, 12.2%, and 21.8% of the total cooling demand being met by geo-exchange for the mid-rise, high-rise, and school, respectively.

Table 8: Shoulder season summary of mid-peak/peak power reduction potential, ACE increase, and resulting TCDM for the mid-rise, high-rise, and school buildings

Building	Peak Power Reduction (%)	Mid-Peak Power Reduction (%)	ACE Increase (%)	TCDM (%)
Mid-Rise	16.3	16.6	0.5	53.7
High-Rise	14.4	14.9	0.9	48.2
School	24.4	24.7	0.4	56.4

3.4.3.2 Peak Season Operation

The same analysis was repeated for a peak season operating schedule to determine how the bore-field pre-conditioning affects the hybrid systems in a period of typically high cooling demand. It can be seen in Figure 17 that pre-cool strategy impact on the systems annual energy consumption is sensitive to the time in which it is implemented in a cooling season. Both the mid-rise (Figure 17a) and high-rise (Figure 17b) exhibit similar trends (as a SS schedule), where energy consumption increases for lower pre-cool durations and begins to decrease for longer periods. What differs from peak season results is that both the mid-rise and high-rise AECC is greater than the shoulder season simulation, resulting in an AECC increase of 8.0% and 6.5% for the mid-rise and high-rise, respectively. This consumption increase is a result of the high cooling load density during the peak season, degrading the thermal impact the pre-cool has on the bore-field due to a greater amount of heat being injected into the ground to satisfy cooling demands. The school (Figure 17c) shows a positive impact for a peak season pre-cool, showing a degrading energy consumption trend for pre-cool durations greater than four hours, resulting in a minor AECC increase of 0.7%. The school is able to benefit from peak season operation due to lower cooling

load density in this time frame (illustrated in Figure 14c), as a result of the unique occupancy pattern attributed to the educational facility. With the reduced thermal demand and the integration of an OGPC strategy, the geo-exchange system was able to supply a larger percentage of cooling demand in a more efficient manner.

Figure 17: AECC for PS operation versus pre-cool duration for buildings (a) mid-rise, (b) high-rise, and (c) school

Figure 18 presents the annual cost of cooling trends with respect to the variation in pre-cool duration. For both the mid-rise (Figure 18a) and high-rise (Figure 18b), peak season pre-

conditioning provides no economic benefit in the form of a reduction in operating costs when the TOU structure in Section 3.3.2 is implemented. The school (Figure 18c) shows a significant cost reduction for off-peak pre-conditioning with a minimum pre-cool duration of approximately five hours to break even with the base case control scheme. The HGSHHP system shows the greatest cost savings for a full 12 hour pre-cool, with an ACC reduction of 9.1%.

Figure 18: ACC for PS operation versus pre-cool duration for buildings (a) mid-rise, (b) high-rise, and (c) school

Utilizing the pre-conditioning strategy for a peak season operating bracket showed no financial incentive for the mid-rise and high-rise buildings, as depicted in Figure 18a and Figure 18b, respectively. The ability to generate operational cost savings is limited by the small price difference between off-peak/mid-peak and off-peak/peak electricity costs, for the TOU structure adopted. Although the potential savings are limited (for the mid-rise and high-rise) the proposed operating strategy still provides load leveling capabilities for mid-peak/peak power reduction, summarized in Table 9. From Table 9 it can be seen that all three buildings show a positive impact for mid-peak/peak power reduction with the addition of an increase total cooling demand met by the geo-exchange system. Peak season simulation resulted in a TCDM increase of 8.8%, 9.0%, and 21.9% for the mid-rise, high-rise, and school, respectively. When considering the ACE, the mid-rise and high-rise show minor increases in total CO₂ emitted; resulting from the degraded pre-cool benefit caused by higher cooling load density in peak season. When evaluating the school, the simulations indicate a potential 15% reduction in total CO₂ emitted. The school shows a 15% savings in CO₂ emitted due to the reduced cooling demand in peak season (lower building occupancy) in addition to there being a greater difference between the off-peak/peak emissions factors during the peak season timeframe.

Table 9: Peak season summary of mid-peak/peak power reduction potential, ACE increase, and resulting TCDM for the mid-rise, high-rise, and school buildings

Building	Peak Power Reduction (%)	Mid-Peak Power Reduction (%)	ACE Increase (%)	TCDM (%)
Mid-Rise	8.8	10.4	4.4	50.5
High-Rise	8.7	8.8	3.1	44.9
School	33.9	25.5	-15.0	56.4

3.4.3.3 Full Season Operation

The control algorithm was modified to simulate a full season operating schedule, where the predicted results for AECC and ACC with respect to the variation in pre-cool duration are presented in Figure 19 and Figure 20, respectively. It can be seen from Figure 19 that all three buildings show a declining trend for energy consumption with a pre-cool duration greater than eight, nine, and four hours for the mid-rise, high-rise and school, respectively.

Figure 19: AECC for FS operation versus pre-cool duration for buildings (a) mid-rise, (b) high-rise, and (c) school

Figure 20: ACC for FS operation versus pre-cool duration for buildings (a) mid-rise, (b) high-rise, and (c) school

Both the mid-rise (Figure 19a) and high-rise (Figure 19b) show a declining AECC trend for longer pre-cool durations but still result in a significant increase in energy consumption (for a 12 hour pre-cool), with a 9.1% and 7.4% increase, respectively. With the school (Figure 19c) resulting in a minor 1.2% increase in AECC for a maximum pre-cool duration.

A full season operating period positively impacts all three building cases from an operating cost perspective, as illustrated in Figure 20. The mid-rise (Figure 20a) and high-rise (Figure 20b) buildings require close to the maximum pre-cool duration to obtain any operational cost savings, resulting in the largest ACC reduction of 0.4% and 1.7%, respectively. The lower magnitude of annual cooling cost savings are a result of the inclusion of the peak season operation within the full season timeframe, previously shown to be less favorable to the mid-rise and high-rise buildings. To elevate the cost savings associated with both of the multi-residential buildings, it is suggested that peak season operation should be omitted from the pre-cool schedule, putting precedence on shoulder and mid-season operation. From Figure 20c, it is clear that the school is significantly impacted by a full season operating schedule, with the predicted minimum pre-cool duration being approximately five hours, the system accomplishes a predicted 16.4% savings in operating cost for a pre-cool duration of 12 hours.

From the results presented in Table 10 it is evident that a full season operating period shows great potential for mid-peak/peak power reduction, with significant percentages of electrical load being shifted to off-peak periods. In addition the total cooling demand met by the geo-exchange systems increased by 20.7%, 21.3%, and 43.7% for the mid-rise, high-rise, and school, respectively. When evaluating the ACE for the three building cases in full season, consistent trends occur, when compared to peak season operation. The mid-rise and high-rise indicate minor increases in CO₂ emitted, while the school indicates a potential 14.5% reduction. The school with a full season operating schedule shows less potential for CO₂ emissions (compared to peak season) due to the inclusion of the shoulder season pre-cool timeframe.

Table 10: Full season summary of mid-peak/peak power reduction potential, ACE increase, and resulting TCDM for the mid-rise, high-rise, and school buildings

Building	Peak Power Reduction (%)	Mid-Peak Power Reduction (%)	ACE Increase (%)	TCDM (%)
Mid-Rise	25.3	26.6	4.9	62.5
High-Rise	23.6	23.3	4.1	57.2
School	58.5	49.8	-14.5	78.3

3.5 Chapter Summary

The scope of this analysis is to provide a preliminary evaluation of the proposed control strategy for the use in hybrid geo-exchange systems. The study aimed to characterize the impact that proactive pre-conditioning of a bore-field can have on system performance. Employing both a thermal pre-conditioning scheme in tandem with TOU-conscious control, this study proposes a multifaceted methodology to address system economics through improved operating efficiency (during peak hours of demand) contributing to a reduction in mid-peak/peak power consumption.

In this study, a numerical model was developed using MATLAB to predict the performance of HGSHP systems associated to three building cases: a mid-rise, high-rise, and school. The model was used to simulate the hybrid geo-exchange systems response to the proposed OGPC control strategy for a variation in pre-cool duration. This analysis was conducted for three proposed pre-cool schedules: shoulder operation, peak operation, and full season operating. The following is a summary of the significant findings and conclusions determined from this evaluation:

Shoulder Season Operation:

- The mid-rise and high-rise buildings showed the largest financial benefit, obtaining an annual cooling cost savings of just under 4% for less than a 1% increase in energy consumption.
- Simulated reductions of 17% - 25% of mid-peak/peak power consumption were accomplished, with an increase in total cooling demand met by the GSHP of approximately 12% for the mid-rise and high-rise, and 22% for the school.
- Simulations indicate no potential for emissions savings, resulting in minor increase of just under 1% of CO₂ emitted for all three building cases.

Peak Season Operation:

- The impact the OGPC has on the hybrid geo-exchange system's performance is sensitive to the building's thermal load characteristics.
- The mid-rise and high-rise showed no financial incentive due to the high cooling load density (greater heat rejection to the bore-field) for a peak season time frame, diminishing thermal benefit the pre-cool poses.
- Although the mid-rise and high-rise showed no financial incentive to pre-cooling, a notable mid-peak/peak power reduction and increase in total cooling demand met by the GSHP were simulated.
- The school indicates a potential 15% reduction in total CO₂ emitted, due to the combination of reduced cooling demand (lower building occupancy) in addition to there being a greater difference between the off-peak/peak emissions factors, during peak season operation.

Full Season Operation:

- Minor operating cost savings for both the mid-rise and high-rise were observed. For further cost improvement, peak season operation should be excluded and additional focus should be placed on shoulder and mid-season operation.
- The school showed a predicted operating cost savings of over 16% with just over a 1% increase in energy consumption, while reducing mid-peak/peak power consumption by approximately 50% - 59%, with the GSHP meeting an additional 44% of the total cooling demand.
- The OGPC strategy significantly impacts the school due to the unique thermal load distribution, resulting from the occupancy pattern specific to an educational institution (lower summer loads, meaning lower heat rejection to the bore-field).

CHAPTER 4 – Pre-Cool Sensitivity to Hybrid System Proportions and Geographic Location

To expand on the study conducted in Chapter 3, two analyses were carried out to examine the proposed OGPC strategy's sensitivity to various factors. The first study examined the impact the pre-cool strategy has on a system's performance for a variation of hybrid system proportions. For the first study, three buildings were considered; a mid-rise multi-residential facility, high-rise multi-residential facility, and a school. The second study evaluated the impact the proposed OGPC strategy has on a hybrid system's performance for a change in geographic location. For this analysis only the school building was examined.

4.1 Methodology

In this study, the methodology outlined in Chapter 3 was used to carry out the two aforementioned sensitivity analyses. Utilizing the developed eQuest BES models for the three building cases, a set of annual hourly heating/cooling loads were generated. The resulting building load data is then used as an input to generate an optimized HGSHP design using the rigorous computerized design algorithm presented in [13], and studied further in [21, 14, 6]. Following the process logic outlined in Figure 9, the results of the BES and hybrid system optimization are used as input parameters to the HGSHP numerical performance prediction model. The numerical model is used as a tool to characterize the affect that the proposed operating strategy has on the hybrid system's performance, with the model's process logic illustrated in Figure 10. In continuing to examine the potential behind the proposed operating strategy, two variations of the numerical algorithm were developed for the present sensitivity studies as discussed in Sections 4.1.1 and 4.1.2.

4.1.1 Pre-Cool Sensitivity to Hybrid System Proportions

The previous study analysed the impact the proposed control strategy has on the operation of an optimally sized HGSHS system. This optimization was conducted by varying the shave factor from zero to one - a two dimensional optimization (total cost as a function of shave factor) – resulting in the economically optimized system being determined. The resulting optimal design corresponds to an optimal shave factor (portion of peak cooling demand met by the geo-exchange system) and ground-loop length, for a fixed entering source/sink temperature pair corresponding to heating/cooling mode operation.

For the three buildings considered the eQuest models and building energy simulation results remained consistent with the analyses conducted in Chapter 3, simulating the three facilities located in Toronto, Canada. The optimization methodology in [13] was used to determine the systems' technical specifications; the assumed design and capital cost parameters were held constant, represented in Table 2 and Table 3, respectively. The result of the hybrid geo-exchange design process generates a set of system specifications for each building case, for a shave factor value ranging from zero to one; a shave factor value of zero represents a conventional system and a value of one represents a full geo-exchange system. Any shave factor value specified between the upper and lower limit of zero and one represents a hybrid geo-exchange configuration. The following Figure 21 was generated from the results of the optimization methodology in [13] conducted on the three previously mention building cases. Figure 21 represents the ground-loop length in meters (ordinate), required to satisfy the systems shave factor parameter (abscissa); the percentage of peak cooling demand met by the geo-exchange system.

Figure 21: Ground-loop length verses shave factor (α)

4.1.1.1 Subroutine Process Logic

The current study evaluates how the hybrid system size affects the potential benefit the proposed off-peak ground pre-cool control strategies pose. This study was accomplished by modifying the numerical performance prediction model with the addition of a subroutine to allow for the variation in the hybrid geo-exchange systems' design specifications. To quantify the potential benefit the proposed operating strategy introduces for variable system size, the process logic presented in Figure 22 is employed.

Figure 22: Algorithm process logic for hybrid proportion sensitivity analysis

When the proposed algorithm is initiated (referring to Figure 22), a shave factor initial condition of zero is established. The module increments the shave factor parameter by a value of 0.01, representing the first hybrid geo-exchange system design. The input data (annual hourly heating/cooling loads, TOU rates, hybrid design parameters) of the simulated building case is then

processed to determine the hybrid geo-exchange system specifications; establishing the design capacities of the GSHP/auxiliary systems and the required ground-loop length (from correlations presented in Figure 21), for the current shave factor value. Once the input parameters have been determined, the algorithm performs four numerical simulations of the resulting HGSHP system, for an operating duration of one year. The four simulations use the numerical model to forecast the performance result of a simulated building's HGSHP system for the following: base case set-point control scheme, shoulder season (SS), peak season (PS), and a full season (FS) pre-cool schedules. For the three scheduled pre-cool simulations, a maximum pre-cool duration of 12 hours was implemented, due to the trends observed in the analysis conduct in Chapter 3. Once the four simulations have expired, the resulting performance data of the three scheduled pre-cool models are processed to determine the performance trends with respect to the base case model. The algorithm then evaluates if the current shave factor limit has been reached, referring to a value of 0.99. When the shave factor reaches the limit value and satisfies the logic condition (illustrated by the decision block in Figure 21), the program is terminated. The proposed subroutine varies the hybrid system size for a shave factor range of 0.01 - 0.99, excluding the values of zero and one, referring to HVAC systems that are fully conventional and fully geo-exchange, respectively. The results of this study are presented and discussed in Section 4.2.

4.1.2 Pre-Cool Sensitivity to Geographic Location

In this study, the proposed operating strategy's sensitivity to geographic location is considered. The school was selected as the case to analyse the proposed strategy's sensitivity when implemented in the following locations: Toronto (Ontario), Houston (Texas), and Phoenix (Arizona). The methodology used in this study is consistent with the process flowchart presented in Figure 9, with modifications being made to the building energy model and hybrid design process; the procedure used in this study is discussed in further detail in the following subsections.

4.1.2.1 Building Energy Simulation Modifications

With the use of the previously developed eQuest model of the school, two additional simulations were conducted to characterize the school building's heating/cooling load requirement for the new locations, Houston and Phoenix. In the two modified building energy models the school building's technical specifications (structural information, occupancy patterns, appliance information, etc.) were assumed constant; only the weather data was adjusted to account for the building relocation. Table 11 presents the summary of the building energy simulation results for the school located in Toronto, Houston, and Phoenix. Referring to Table 11, it can be observed that the building relocation, to areas with a warmer climate resulted in a significant impact on the school's annual thermal requirement (compared to Toronto). For the relocation to Houston the school is characterized as a cooling dominant facility with a minor 5.6 kW increase in peak cooling demand; with a more substantial difference realized in the comparison of the total heating and cooling demands, where the total cooling requirement increased by a factor of 10. For the relocation to Phoenix the school maintains its cooling dominant classification with a more substantial 32.4 kW increase in peak cooling demand. When comparing the total heating and cooling demands, a similar trend is recognized, where the total cooling requirement increases by a factor of 12.5.

Table 11: Summary of building thermal characteristics for the school located in Toronto, Houston, and Phoenix

Location	City	Peak Cooling Load (kW)	THD (MWh)	TCD (MWh)	THD/TCD
Ontario	Toronto	174.0	295.9	252.1	1.17
Texas	Houston	179.6	43.8	425.6	0.10
Arizona	Phoenix	206.4	44.2	552.8	0.08

4.1.2.2 Hybrid System Design Modifications

For this study the algorithm presented in [13] was used to design two additional HGSHP systems for the school located in Houston and Phoenix. The algorithm was run for the two additional cases with modifications being made to two aspects in the computation, the building load file (from eQuest simulations in Section 4.1.2.1) and ground temperature. All other parameters were held constant from previous studies, summarized in Table 2 and Table 3. The following Table 12 presents a summary of the modified ground temperature input parameter to the design algorithm in [13], along with the resulting ground-loop length requirements for the additional simulated cases. A shave factor value of 0.21 was selected for all three hybrid designs, corresponding to the optimal value of the Toronto simulation. This parameter was held constant in order to expose the proposed strategy's sensitivity to location change alone.

Table 12: Summary of modified input parameters and simulation results for the school located in Toronto, Houston, and Phoenix

Location	City	Ground Temperature [66] (°C)	Ground-Loop Length (m)
Ontario	Toronto	10	791.9
Texas	Houston	20	2331.2
Arizona	Phoenix	24	4342.0

Upon completion of the of the two processes described in Sections 4.1.2.1 and 4.1.2.2, the updated building load data and hybrid system specifications are loaded to the numerical performance prediction model. Four annual simulations are conducted for each building case, for the school building located in Toronto, Houston, and Phoenix. A base case simulation is generated for each building location; acting as a reference to evaluate the impact of the shoulder season (SS), peak season (PS), and full season (FS) pre-cool operating schedule for varying geographic locations. The results of this study are presented and discussed in Section 4.3.

4.2 Results: Pre-Cool Sensitivity to Hybrid System Proportion

Applying the methodology from Section 4.1.1, the three building cases (located in Toronto) were evaluated to determine the proposed HGSHP control strategy's sensitivity to the hybrid systems proportions. The scope of this analysis is on characterizing the impact on the following outputs: annual electricity consumption for space cooling (AECC), annual cost for space cooling (ACC), peak/mid-peak electricity consumption, and total cooling demand met (TCDM) by the geo-exchange system.

Figure 23 presents the variation of annual electrical power consumption for the three building cases studied. For plots illustrated in Figure 23, the ordinate represents the annual electrical power consumption savings, with respect to the base case model for a shave factor range of 1% - 99% (located on the abscissa). Each data point represents an annual operating period for the respective hybrid shave factor with the corresponding pre-cool schedule; where the meaning of a negative savings nullifies to a percent increase in power consumption. In Figure 23, the vertically oriented lines are a reference indicator of the optimal shave factor parameter associated to the three building cases; 23%, 21%, and 21%, for the mid-rise, high-rise, and school, respectively.

The energy savings profiles for the mid-rise (Figure 23a), high-rise (Figure 23b), and the school (Figure 23c) exhibit similar trends with regard to the trajectory of energy consumption savings verses hybrid system size; in all cases the slope of AECC savings has a positive trajectory for small shave factor values (in all cases under 20%). In all three building cases the trajectory of the AECC savings profiles become negative for a shave factor range of approximately 20% - 60%. After transitioning into a shave factor range of approximately 60% - 99% the slope of the AECC savings profiles becomes positive, before converging at a shave factor of 99% with a predicted AECC savings percentage of zero. As illustrated in Figure 23, the previous discussion of the AECC savings profiles provide a general overview of the trends occurring for all three pre-cool schedules studied. Each pre-cool scheme results in a unique savings profile (for each building case) for the variation in hybrid system shave factor, which is clearly depicted in Figure 23a (mid-rise), Figure 23b (high-rise), and Figure 23c (school).

Figure 23: AECC savings potential versus shave factor (α) for buildings (a) mid-rise, (b) high-rise, and (c) school

For the mid-rise (Figure 23a) it can be seen that the proposed OGPC control strategy has the smallest impact on system energy consumption for a shave factor range of approximately 5% - 20%; where there is a negligible increase in total energy consumption for space cooling. Similarly, the school (Figure 23c) shows a positive impact on system energy consumption; all three pre-cool schedules show an AECC savings of approximately 2.5% for the lower end of the shave factor interval, with the savings degrading as a shave factor approaches the upper limit of 20%. For the

high-rise (Figure 23b) all three pre-cool schedules result in an increase in AECC for a shave factor range of 8% - 20%; starting at an approximate consumption increase of 2.5% for the SS/PS and 5% of the FS schedules (for the lower limit of the shave factor interval), with energy consumption increasing further as the upper limit is approached. For all building cases it can be seen that the potential pre-cool benefit proposed by the three operating schedules is sensitive to hybrid system proportions, with the most ideal hybrid system size (from an energy consumption perspective) corresponding to a shave factor range of 5% - 20%. When examining Figure 23, the optimal shave factor (vertical datum line) resulting from the design procedure specifies a hybrid system size that falls outside of ideal AECC shave factor range. Incorporating the pre-cool control strategy into the hybrid design procedure can allow for an additional dimension of optimization from a power consumption perspective. Optimizing power consumption will impact the operating cost component of the hybrid design process, along with the annual carbon emissions; introducing additional economic incentive for systems implemented in areas with a significant carbon tax. Of the three suggested pre-cool schedules, Figure 23 clearly illustrates that the shoulder season operating schedule is the most resilient to variations in hybrid system sizing.

To conceptualize the physical meaning of trends addressed in Figure 23, the significance of the hybrid system shave factor must be understood. The shave factor parameter represents the percentage of a building's peak cooling demand met by the GSHP, with the difference being supplied by the auxiliary cooling system. For example, if a building requires 100 kW of peak cooling and a shave factor design of 10% is selected, the resulting cooling capacity of the GSHP and auxiliary cooling system are 10 kW and 90 kW, respectively. When the shave factor is varied, three hybrid system parameters change: ground-loop length, GSHP system cooling capacity, and auxiliary system cooling capacity. From Figure 23, undersized hybrid systems are most promising

for this strategy due to smaller ground-loop lengths (meaning smaller bore-field/soil volumes) and large auxiliary cooling capacity, which would translate to faster ground temperature response. This concept is further illustrated for shave factor values of 1% and 99%; 1% representing the smallest hybrid system configuration (large auxiliary cooling capacity and small ground-loop length) resulting in excessive energy consumption from pre-cooling with insufficient GSHP operating benefit, due to an undersized bore-field heat exchanger. Alternatively, a shave factor of 99% represents the largest hybrid system configuration (small auxiliary cooling capacity and large ground-loop length) resulting in negligible energy consumption increase from pre-cooling and negligible thermal impact on the bore-field, due to large bore-field/soil volumes.

Figure 24 presents the simulation results corresponding to the sensitivity of the potential economic benefit attributed to the use of the proposed OGPC control strategy. Utilizing the TOU cost structure presented in Table 4, the potential ACC savings (ordinate) sensitivity to hybrid system size is presented for the mid-rise (Figure 24a), high-rise (Figure 24b), and the school (Figure 24c). Figure 24 illustrates similar trends as those present in the AECC savings sensitivity to hybrid system size (Figure 23). For all three building cases the scheduled ground pre-conditioning technique shows potential for improved system economics by reducing operating costs, occurring for undersized hybrid systems (low shave factor values). In all three building cases, the ACC savings potential occurs for a shave factor range of approximately 5% - 40%. With the maximum potential for ACC savings occurs on the lower end of the previously mentioned shave factor range, and gradually degrades as the upper limit is reached. In Figure 24, the optimal shave factors (vertical datum line) for all three building cases fall within the ideal ACC shave factor range. From Figure 24, it is clear that there is potential for further optimization in an attempt to target the maximum operating cost savings. Incorporating this control strategy into the hybrid design

procedure will allow for an additional dimension in the optimization process, introducing the potential for further improved system economics. For the mid-rise (Figure 24a) and high-rise (Figure 24b), a shoulder season pre-conducting schedule shows the least sensitivity to variation in hybrid system proportions. For the school (Figure 24c) both the shoulder season and peak season ACC savings profiles follow each other quite closely. The similarity in the shoulder and peak season profiles is attributed to the school's unique cooling load distribution (illustrated in Figure 14c) with minimal deviation in cooling demand between shoulder and peak season.

Figure 24: ACC savings potential versus shave factor (α) for buildings (a) mid-rise, (b) high-rise, and (c) school

Figure 24 illustrates a potential economic incentive for the further development of an OGPC strategy. This incentive is realized due to two factors: an assumed increase in auxiliary system efficiency between peak/off-peak operating periods, and an assumed increase in GSHP system efficiency when exposed to a pre-cooled bore-field. The latter of the two aforementioned assumptions introduces the potential for shaving mid-peak/peak power consumption by taking advantage of the improved bore-field thermal characteristic during periods of peak cooling demand. The following Figure 25 and Figure 26 present the predicted annual mid-peak and peak power reduction sensitivity to hybrid system proportions, respectively. The mid-rise (Figure 25a/Figure 26a), high-rise (Figure 25b/Figure 26b), and school (Figure 25c/Figure 26c) all depict consistent trends with regard to mid-peak/peak power reduction potential. In all building cases maximum peak power reduction occurs within a shave factor range of 5% - 40%. As the hybrid system size increases past a shave factor of 40% mid-peak/peak power reduction degrades until a negligible impact results, at a shave factor of approximately 80%.

Figure 25: Mid-peak power reduction potential versus shave factor (α) for buildings (a) mid-rise, (b) high-rise, and (c) school

Figure 26: Peak power reduction potential versus shave factor (α) for buildings (a) mid-rise, (b) high-rise, and (c) school

The school's simulated full season (FS) mid-peak/peak power consumption profiles (illustrated in Figure 25c and Figure 26c) resulted in an approximate linear trend, for a shave factor range of 1% - 40 %. This result is because the off-peak pre-cool has its greatest thermal impact for systems with smaller shave factors. In addition, the schools cooling load profile is consistent throughout the full season time frame (meaning consistent operating cycles), as a result the linearity

assumption used in the performance correlations comes through in the mid-peak/peak power consumption profiles.

The ACC savings profiles presented in Figure 24 greatly depend on the TOU cost structure implemented in the computation. Despite the simulated potential for mid-peak/peak power reduction, illustrated in Figure 25 and Figure 26, financial incentives for bore-field pre-conditioning are not always realized. For this study, the Ontario 2015 TOU cost structure was used, where the off-peak, mid-peak, and peak power cost are 8.0 ¢/kWh, 12.2 ¢/kWh, and 16.1 ¢/kWh, respectively [55]. As the price difference between off-peak/mid-peak and off-peak/peak operating brackets increase, this operating strategy will become more economically appealing.

Figure 27 illustrates the variation in TCDM (total cooling demand met by the GSHP) verses hybrid system size for the mid-rise (Figure 27a), high-rise (Figure 27b), and school (Figure 27c). Figure 27 presents a graphical comparison of the TCDM profiles for the annual operating of a HGSHS system, fitted with the proposed pre-cool operating schedules. For all three builds, the TCDM profiles diverge from the base case profile for small shave factor values, after which the profile trajectory changes and converges with the base case reference within a shave factor range of 40% - 60%. This occurrence is caused by the GSHP system cooling output increasing because of a reduced entering sink temperatures, and cooler ground temperatures resulting from the scheduled bore-field pre-cooling. The school (Figure 27c) shows a more extreme response due to its unique cooling load distribution (Figure 14c), for which the reduced peak season cooling demand makes an undersized system a more appropriately sized design. With the addition of the pre-cool strategy, the school's geo-exchange system is able to meet a larger amount of the cooling requirement than initially designed for, requiring less supplementary assistance from the auxiliary cooling system.

Figure 27: TCDM versus shave factor (α) for buildings (a) mid-rise, (b) high-rise, and (c) school

4.3 Results: Pre-Cool Sensitivity to Geographic Location

Applying the methodology presented in Section 4.1.2, the school was the selected test case to evaluate the proposed OGPC control strategy's sensitivity to geographic location. The scope of this study is on characterizing the impact on the following outputs: annual electricity consumption for space cooling (AECC), annual cost for space cooling (ACC), peak/mid-peak electricity consumption, and total cooling demand met (TCDM) by the geo-exchange system. The school was simulated for a total of three geographic locations; Toronto (Ontario), Houston (Texas), and Phoenix (Arizona).

Figure 28a and Figure 28b present the AECC and ACC savings potential for the proposed building relocation, respectively. From the results presented in Figure 28a and Figure 28b, it can be seen that the bore-field pre-conditioning strategy is extremely sensitive to the geographic location. In Figure 28, it is clearly illustrated that the potential benefit proposed by the OGPC strategy becomes less significant when employed in areas with hotter climates. For this analysis, the simulation parameters that vary with respect to building location are the school's annual hourly heating/cooling demands (due to changes in weather patterns) and ground-loop length (due to weather patterns and higher ground temperatures). For the selected geographic locations, Toronto has the coolest climate (lowest ground temperature) and Phoenix has the hottest climate (highest ground temperature). From the simulation results illustrated in Figure 28, as cooling demand frequency/amplitude increase (for Houston and Phoenix), the bore-field pre-cool produces less thermal impact, as a result of increased heat rejection to the ground. In addition, with higher ground temperatures (for Houston and Phoenix) the ground-loop length requirements increase, necessitating larger bore-field/soil volumes; resulting in slower ground thermal response. Summarizing the trends observed in Figure 28, the AECC and ACC savings potential (due to the

integration of the OGPC strategy) decreases when employed in geographic locations with hot climates and high ground temperatures.

Figure 28: (a) AECC savings potential and (b) ACC savings potential versus geographic location for the school, with a maximum pre-cool duration of 12 hours

Figure 29a and Figure 29b present the mid-peak and peak power reduction potential for the proposed building relocation, respectively. Figure 30 presents the variation in $\Delta TCDM$ (difference in TCDM between the base case and respective pre-cool schedule) of the school building for the proposed relocations. From Figure 29 and Figure 30 it can be seen that the proposed seasonal pre-cool schedules positively impact the school from a peak power shaving and TCDM perspective, for all three geographic locations. Figure 29 and Figure 30 illustrate the same aforementioned trends, where the potential for peak power shaving and increasing TCDM decreases when employed in geographic areas with hot climates and high ground temperatures. For all locations studied, the seasonal pre-cool schedules have a positive impact on reducing mid-peak/peak usage, as a result of the artificially enlarged cooling capacity. Unfortunately, due to the TOU electrical cost structure used in this computation, financial benefits are not always realized, which is clearly illustrated in the case of the school building located in Phoenix (Arizona).

Figure 29: (a) Mid-peak and (b) peak power reduction potential versus geographic location for the school, with a maximum pre-cool duration of 12 hours

Figure 30: $\Delta TCDM$ versus geographic location for the school, with a maximum pre-cool duration of 12 hours

4.4 Chapter Summary

In this chapter, the proposed OGPC operational strategy is further analyzed by the means of two numerical studies. The studies are conducted in the form of sensitivity analyses, in which the potential operational benefit of the three proposed bore-field pre-cool schedules (SS, PS, and FS) were further evaluated. The first study (Section 4.2) evaluated the pre-cool strategy's sensitivity to HGSHP system size (shave factor), which was simulated for three building test cases; a mid-rise multi-residential, high-rise multi-residential, and a school building. All three of the buildings were simulated in Toronto, fitted with an OGPC strategy with a maximum pre-cool duration of 12 hours. The second study (Section 4.3) analyzed the pre-cool strategy's sensitivity to geographic location, which was simulated for a single building case, the school. The school was simulated in three geographic locations, Toronto (Ontario), Houston (Texas), and Phoenix (Arizona). The pre-cool schedules were set to a maximum 12 hour off-peak duration, for a hybrid shave factor of 0.21 (21%). The following is a summary of the significant findings and conclusions from the presented studies:

Pre-Cool Sensitivity to Hybrid System Proportions:

- For all building cases the following hybrid system ranges showed the greatest benefit for: AECC saving (5% - 20%), ACC savings (5% - 40%), mid-peak/peak power reduction (1% - 60%), and TCDM (1% - 50%).
- Of the three proposed seasonal pre-cool schedules, the shoulder season schedule showed the least sensitivity to hybrid system proportions; showing the most potential for producing an economic incentive for all three building cases studied.
- Undersized hybrid systems are most promising for the integration of an OGPC strategy due to smaller ground-loop lengths (meaning smaller bore-filed/soil volumes) and large auxiliary cooling capacity, which translates to faster ground temperature response.

Pre-Cool Sensitivity to Geographic Location:

- The potential benefit proposed by the OGPC strategy degrades when employed in areas with hotter climates and higher ground temperatures.
- As cooling load frequency/amplitude increases, the bore-field pre-cool strategy results in less of an operational benefit.
- For locations with higher ground temperatures, the ground-loop length requirements increase; larger ground-loop lengths result in less potential for operational benefits.
- The financial benefits realized with this operating strategy are sensitive to the TOU utility rate structure. As the price difference between off-peak/mid-peak and off-peak/peak electricity rates increase, this proposition will become more economically appealing.

CHAPTER 5 – Conclusions

The ‘Off-Peak Ground Pre-Cool Control Strategy’ has been presented in this thesis as an alternative operating methodology for control of HGSHP systems. This control paradigm aims to employ proactive decision making in tandem with TOU conscious control in an attempt to shave peak power usage; accomplished by artificially improved bore-field/soil thermal characteristics by a means of off-peak pre-conditioning. This strategy introduces a new prospective on hybrid geo-exchange system operation, targeting the improvement of system economics through a reduction in operating costs while contributing to the balancing of the electrical grid.

The studies performed in this thesis present a preliminary analysis aimed at demonstrating the impact that the proposed OGPC control strategy has on a HGSHP system’s performance. With aid of the numerical performance prediction model, newly developed in this thesis, the presented analyses shed light on the potential trends that would occur for the physical implementation of an OGPC strategy. The analysis presented in Chapter 3 illustrates the impact that the duration of the scheduled off-peak pre-cool has on hybrid geo-exchange system’s performance, for the three building cases. The sensitivity analysis presented in Chapter 4 shows the potential impact the proposed off-peak pre-cool schedules have, for a variation in hybrid system proportions and geographic location. Based on the studies performed, a framework of the OGPC control strategy has been established which merits further research and development, as an alternative operating methodology for the control of HGSHP systems.

5.1 Summary

The following is a summary of the significant findings and conclusions determined from the analyses conducted in Chapter 3 and Chapter 4:

5.1.1 Characterizing the Effect of an Off-Peak Ground Pre-Cool Control Strategy on Hybrid Ground-Source Heat Pump Systems¹

Shoulder Season Operation:

- Both the mid-rise and high-rise buildings showed the most potential from a shoulder season schedule (for a maximum pre-cool duration); with an annual cooling cost savings of just under 4% with a 1% increase in power consumption.
- For all building cases, a significant impact on mid-peak/peak power consumption was observed; resulting in 17% - 25% peak power shaving potential.
- In all cases the total cooling demand met by the geo-exchange system increased; resulting in an additional 12% - 22% of demand met.
- For all three building cases, no potential for emissions savings was realized, resulting in minor increases of just under 1% of CO₂ emitted.

Peak Season Operation:

- The impact the OGPC has on the hybrid geo-exchange system's performance is extremely sensitive to the buildings cooling load characteristics.
- The mid-rise and high-rise simulations showed no financial benefit as a result of the high cooling load density in the selected time frame; reducing the impact the pre-cool strategy has on the soil/bore-field.

¹ This section is adapted from Chapter 3

- Although the mid-rise and high-rise showed no financial benefit, there are still positive trends observed with regard to mid-peak/peak power reduction and the total cooling demand met by the GSHP.
- The school indicated a potential 15% reduction in total CO₂ emitted, due to the combination of reduced cooling demand (lower building occupancy) and a greater difference between the off-peak/peak emissions factors, during peak season.

Full Season Operation:

- Minor operating cost savings for both the mid-rise and high rise were observed; to improve the benefit, peak season operation should be omitted from the pre-cool schedule.
- The school data showed a significant response to a full season pre-cool schedule; predicted cost savings of over 16% for a 1% increase in energy consumption, while reducing mid-peak/peak power consumption by 50% - 59% with the GSHP meeting an additional 44% of the total cooling demand.
- The OGPC strategy significantly impacts the school's operation due to its unique cooling load distribution, caused by to the unique occupancy pattern specific to an educational institution (reduced summer loads).

5.1.2 Pre-Cool Sensitivity to Hybrid System Proportions and Geographic Location²

Pre-Cool Sensitivity to Hybrid System Proportions:

- The following hybrid system ranges showed the greatest benefit for proposed pre-cool schedules: AECC saving (5% - 20%), ACC savings (5% - 40%), mid-peak/peak power reduction (1% - 60%), and TCDM (1% - 50%).

² This section is adapted from Chapter 4

- The shoulder season pre-cool schedule showed the least sensitivity to hybrid system proportions, indicating the greatest potential for an economic benefit for a wider range of hybrid system sizes.
- Undersized hybrid systems are most promising for the integration of an OGPC strategy due to smaller ground-loop lengths (meaning smaller bore-field/soil volumes) and large auxiliary cooling capacity, translating to faster ground temperature response.

Pre-Cool Sensitivity to Geographic Location:

- The OGPC strategy's operating benefit degrades when employed in areas with hotter climates and higher ground temperatures.
- As cooling load frequency/amplitude increases, the bore-field pre-cool strategy has less of a thermal impact on the bore-field/soil, a result of increased heat rejection to the ground (due to amplified cooling load requirements).
- For locations with higher ground temperatures, the ground-loop length requirements increase. Larger ground-loop lengths (bore-field/soil volumes) result in a slower ground thermal response, degrading the operating benefit proposed by the OGPC strategy.
- The financial benefits realized in the proposed control scheme are sensitive to the TOU utility rate structure employed. As the price difference between off-peak/mid-peak and off-peak/peak electricity rates increase, this scheme will become more economically appealing.

5.2 Recommendations

Based on the background research and analyses performed in this thesis, a preliminary understanding of trends that occur for the integration of an OGPC control strategy has been established. However, due to the complexity of the proposed concepts, further research and development is required to fully quantify the benefit behind this methodology. The following are recommendations for further concept development and areas of future research:

Model Development:

The current state of the numerical performance prediction model provides an analysis following the principles of an energy balance. The energy balance method was selected to conduct a preliminary evaluation of the proposed control strategy, to determine if this concept merits further investigation. Despite the promising result generated in this thesis, due to the lack of complexity in the numerical model, there still remains a great deal of work to fully characterize this strategy's benefit. It is recommended that a finite element/computational fluid dynamics component of this model be developed, characterizing the bore-field thermal response (including bore-field configuration) along with the thermal response of the ground-loop working fluid (for both on/off cycles). This component can either be integrated into the numerical platform presented in Appendix C, or can be developed on a separate platform such as COMSOL Multiphysics [67]. Additionally, an algorithm should be incorporated within the model, accounting for the variability in auxiliary system COP as a function of weather data, as opposed to the fixed off-peak/peak COP correction factor currently implemented.

Free-Exchange Operation Coupled to an OGPC Control Strategy:

Implementing an OGPC operating strategy presents the potential for a controlled artificial pre-cool of a geo-exchange system's bore-field. If the pre-cooling technique is able to make a significant reduction to the bore-field's local ground temperatures, the incorporation of a passive cooling (free-exchange) would become feasible. With the addition of this operating mode, added potential for improved system economics would occur [49]. For further discussion of this concept, refer to Appendix A.

Fuzzy Control Algorithms Coupled to an OGPC Control Strategy:

When increasing the complexity of a system's operation, it is important to adopt higher level control techniques, to ensure the maximum benefit is accomplished. From the literature it is clear that fuzzy logic shows promise in HVAC process control [4, 37, 46, 47], with the added benefit of industry acceptance [37]. This control scheme is suggested for future integration into the OGPC operating strategy; where a fuzzy supervisor algorithm should be developed to manage the various operating states of the HGSHF system. Fuzzy control theory was studied further, and a preliminary design of a fuzzy algorithm for local subsystem control has been established, with the framework presented in Appendix B.

Appendix A – Potential for Free-Exchange Operation Coupled with a Bore-field Pre-Cool Strategy

Passive ground-source cooling systems, also referred to as ‘Free-Exchange’ systems, have been implemented in a variety of industrial applications. For example, used for the cooling of electronics in protected sites (i.e., military radar facilities, telecommunications facilities) [20]. For many of these applications, free-exchange systems are designed for dedicated cooling, only using the ground as a thermal sink. As a result, the ground temperatures will increase over time due to the bore-field load imbalance (absence of heat extraction) [68]. To mitigate this thermal imbalance cooling towers are commonly implemented, allowing for bore-field recharging, rejecting excess heat to the outdoor ambient air. A study done by Johansson *et al.* [20], evaluated a hybrid ground-source free-exchange system, paired with an auxiliary dry cooler for increased heat rejection. In [20], a heating season ground recharging strategy was integrated to reduce the load on the bore-field (due to excessive heat injection) and potential for a reduction in ground-loop length requirements were considered (as a result of cooler ground temperatures).

The OGPC strategy proposed in this thesis implements a similar operating principles as presented in [20], but for the integration into dual purpose (heating/cooling capable) heat pump assisted geo-exchange systems, rather than dedicated passive cooling systems. Introducing a controlled bore-field pre-cool will add to the potential feasibility of free-exchange operation in dual purpose HGSHP system. The following sections provide the proposed conceptual framework for the further research of free-exchange operation (paired with the OGPC strategy), along with a preliminary evaluation of the potential benefits associated to this addition.

Integration of Free-Exchange Operation:

For the physical integration of a free-exchange operation, a modified hybrid system configuration is proposed in Figure 31. The addition of two four-way-three-position (4W3P) directional control valves (XV1 and XV2) should be incorporated in the BHE circulation loop as indicated. The addition of the valves XV1 and XV2 allow for the incorporation of a GSHP by-pass (BP) loop, represented by the dashed line in Figure 31. If the BP circulation loop is initiated and the ASHP is decoupled from the BHE loop, free exchange operation would be made feasible. The most probable means of cooling distribution would be a dual coil AHU, where one coil would be dedicated to heat pump assisted cooling (DS loop) and the other would be designated for free-exchange cooling (BP loop) [17].

Figure 31: Modified HGSHP system configuration for free-exchange integration

If a HGSHP was sufficiently instrumented with sensors to accurately monitor bore-field fluid temperature, the passive cooling source could be used to satisfy the respective zone's thermal demand. Due to an absence of the compression affect, indicative to heat pump assisted heating/cooling, free-exchange operation is able to generate COPs in the range of 20 – 30 [69].

The following is list of scenarios where free-exchange could potentially provide significant operating benefits:

- During hours of reduced cooling load, free-exchange mode can be initiated to meet the required demand (assuming fluid temperature is within AHU design specification).
- After a significant geo-exchange system off-cycle, when the bore-field working fluid has had time thermally equilibrate with the surrounding soil, free-exchange cooling could be harnessed more effectively. A significant off-cycle would result in colder working fluid temperatures (depending on the ground temperature and the OGPC schedule used).
- For buildings with large heating demands, seasonal heating operation results in a reduction of bore-field/soil temperatures [70]. As a result, free-exchange could potentially be used to meet a portion of a shoulder season's cooling demand; combined with an OGPC control strategy there is a potential to increase the percentage of load met by free-exchange.
- Immediately after a bore-field pre-conditioning cycle, the ground-loop working fluid would be significantly cooler than the ground (in order to facilitate a reduction in ground cooling). When transitioning into a mid-peak operating period, free-exchange mode should be initiated to make use of the cooling potential in the working fluid, before it is lost to the surrounding soil.

To ensure occupancy comfort, modulation between free-exchange and heat pump assisted operation would be required. While monitoring the BP loop AHU supply temperature, if it exceeds the acceptable temperature threshold (based on the distribution system's design specifications), heat pump assisted operation would be initiated (terminating free-exchange mode).

Potential for Free-Exchange Operation within an OGPC control Strategy:

The HGSHP performance prediction model developed in this thesis currently lacks the computational capabilities to fully quantify the bore-field's thermal response to various operating cases. As a result, a detailed evaluation of the potential for the integrating free-exchange was not possible, as the model is unable to quantify the benefits associated to the various operating scenarios, previously discussed. Although the computational power of the model is limited, with regard to transient operation, it can still shed light on potential for free-exchange for assumed steady-state operation.

A preliminary evaluation of the potential for free-exchange operation was conducted to analyze if the OGPC strategy introduces enough of a thermal impact to reduce steady-state heat pump entering sink temperature below an ideal chill-water threshold. For this analysis, the three buildings presented in Table 1 were used as case studies, with an assumed geographic location of Toronto. Using the HGSHP design methodology proposed in [13], three new hybrid geo-exchange system were generated. All design specifications and capital cost parameters were held constant (presented in Table 2 and Table 3) except for the cooling design entering sink temperature (EST) parameter, which was changed to 14°C (consistent with the arrangement in [20]). Table 13 presents the hybrid system specifications for the modified designs.

Table 13: Summary of modified hybrid system design specification (EST 14°C)

Building	Shave Factor (α)	Ground-Loop Length (m)
Mid-Rise	0.23	3765.6
High-Rise	0.21	6939.3
School	0.21	912.2

Three annual simulations were run for each building case, assessing the impact the proposed OGPC strategy (SS, PS, and FS schedule) has on the modified hybrid design. A modified subroutine was generated to determine the portion of the cooling demand that occurs when the EST is below a threshold of 10 °C. This threshold was selected because a common chill-water supply temperature range used for space cooling/dehumidification is 8 °C – 10 °C [71]. The model assumes a steady-state EST, which is corrected by the temperature response correlations when exposed to a pre-cooled bore-field. For each building case, the simulated results for the employed pre-cool schedules are used as a base case to evaluate the potential for free-exchange operation. It is assumed for an EST below 10 °C, the cooling demand in the corresponding time-step can be met by free-exchange operation, with a COP of 20 [69]. Table 14 presents a summary of result for the preliminary evaluation of free-exchange operation coupled to an OGPC control strategy.

Table 14: Summary of Free-Exchange operation coupled to an OGPC control strategy

Building	Cooling Demand Met, Free-Exchange (MWh)	Portion of TCDM (%)
Mid-Rise	0.9	0.2
High-Rise	12.6	1.1
School	28.1	13.8

From the results presented in Table 14, there is varying potential for the integration of free-exchange into an OGPC schedule. It can be seen that for systems with smaller bore-fields (school), the OGPC strategy has a greater thermal impact, meaning an increased potential for EST suitable for free-exchange operation. For the mid-rise and high-rise building, only a shoulder season pre-cool schedule showed potential for free-exchange operation. On the other hand the school showed potential for free-exchange operation for a full season pre-cool schedule, as a result of the buildings unique cooling load profile (reduction in heat rejection to the bore-field, meaning cooler ground

temperatures). The resulting power/cost reduction potential (for the thermal demand met by free-exchange) was 74.5%, 74.1%, and 68.6% for the mid-rise, high-rise, and school respectively.

Due to the limited computational power of the numerical performance prediction model, the benefits/potential for free-exchange operation coupled to an OGPC control strategy cannot be fully characterized. The future research and development of this proposed partnership is stressed, as there is believed to be significant potential for improved system economics due to a reduction in peak power consumption.

APPENDIX B – Summary of Process Control Logic and Preliminary Framework of FL Algorithms for the Integration into an Off-Peak Ground Pre-Cool Control Strategy

The framework presented in this appendix are intended to serve as a platform for future algorithm developed, for the integration into an OGPC control strategy. This is not intended to serve as an introduction to fuzzy control theory, for background on this control technique refer to [72]. The following outline provides a preliminary design for supervisory control logic for a HGSHp, output actuation logic, and a sample local FL algorithm for motor speed control (i.e. circulation pumps, fans, and compressors).

Supervisor Control Logic:

Table 15 presents a summary of the process logic for the operation of a HGSHp system fitted with an OGPC control strategy, for the various operating states.

Table 15: HGSHp operating logic summary

Loop 1	Cooling Required $\leq$ GSHP Capacity
Loop 2	Cooling Required $>$ GSHP Capacity
Loop 3	(SS OR PS OR FS) AND Off-peak TOU Bracket
Loop 4 (BP)	Cooling Required AND EST $\leq$ 10

Output Actuation Logic:

The control logic in this section pertains to the following output devices: bore-field circulation pump, bore-field directional control valves, GSHP, and ASHP. For the simplification of the control logic, these devices are considered discrete outputs. The bore-field directional control valves (V1 - V2 and XV1 - XV2 in Figure 31) are represented as a cluster of solenoid valves, illustrated in the following Figure 32.

Figure 32: Directional control valve solenoid cluster

The following Table 16 represents the discrete control logic necessary to accomplish the following operating modes for the system presented in Figure 31.

Table 16: Summary of output actuation logic for the HGSHP

	Loop 1	Loop 2	Loop 3	Loop 4 (BP)
P1	1	1	1	1
XV1-S1	0	0	0	1
XV1-S2	0	0	1	1
XV1-S3	0	0	1	0
XV2-S1	0	0	0	1
XV2-S2	0	0	0	0
XV2-S3	0	0	1	1
V1-S1	0	1	1	0
V1-S2	0	1	1	0
V2-S1	0	1	1	0
V2-S2	0	0	0	0
ASHP	0	1	1	0
GSHP	1	1	0	0

Local FL Algorithm:

The following is a preliminary design for a local fuzzy control algorithm that can be interfaced with a variable frequency drive (VFD) module to provide speed regulation of auxiliary equipment, such as: circulation pumps, compressors, and fans. For example, we will assume this design represents a flow control algorithm for a circulation pump. Utilizing the flow set-point reference with feedback from the resulting process state (coming from a flow-metre or a motor shaft encoder), the error (S) and the rate-of-change of the error ($\dot{S}$) are inputs to the control algorithm. The error term is selected as a fuzzy input and the rate-of-change is used as a discrete input. The following Table 17 are the suggested minimum required membership functions for to the fuzzy error input (S).

Table 17: Preliminary membership function selection

	$S < -S_{limit}$	$-S_{limit} < S < 0$	$0 < S < S_{limit}$	$S > S_{limit}$
Positive Lager (PL)	0	0	$\frac{1}{S_{limit}} * S$	1
Positive Small (PS)	-	-	$\frac{-1}{S_{limit}} * S$	0
Negative Large (NL)	1	$\frac{-1}{S_{limit}} * S$	0	0
Negative Small (NS)	0	$\frac{1}{S_{limit}} * S$	-	-

Using the membership functions as proportional weights for the output control signal, the following rule base was generated, and present in Table 18.

Table 18: Control Logic Rule Base

		$\dot{S}$	
		0	1
S	PL	PL	NL
	PS	PS	NS
	NL	NL	PL
	NS	NS	PS

APPENDIX C – HGSHp Numerical Performance Predication Model

The following script is the foundation of the HGSP numerical performance prediction model, newly developed in this thesis. The model syntax pertains to a MATLAB platform, from which is was created.

Code initiated:

```
clear all
close all
clc

%-----System Constants-----
Stage2_PartLoad= 0.67; %dual stage heat pump part load percentage of full
capacity
Stage2_PowerCons= 0.5; %dual stage heat pump energy consumption percentage of
full load operation
COPCT= 10.8/3.412; %coefficient of performance of the cooling tower in
cooling mode (2.2) & ASHP (3.1)
PC_CT_CF= 1.5; %pre-cool cooling tower/ASHP correction factor for nightly
operation of auxiliary system
AuxEffH= 0.78; %thermal efficiency of the aux. heating system
PipeEff= 0.85; %Ground loop pipe thermal efficiency (%)
Temp_Cap= 10; %ground temperature difference limit due to pre-cool (C)
NG_Rate= 0.35; %cost of natural gas in $ per m^3, not including HST
kWh_m3= 10.416;%conversion factor from kWh to m^3 of natural gas

%-----Pre-Cool Mode-----
PC_Peak= 0; %Peak season pre-cool control (Boolean indicator)
PC_Shoulder= 0; %Shoulder season pre-cool control (Boolean indicator)
%-----

filename= 'school original';% demand is the cooling and heating loads (8760
readings, hourly demands on a yearly basis)
alpha= 0.21; %cooling peak shave factor, determined through using the design
approach for optimized hybrid GSHP systems
    % school= 0.21
    % 4elm= 0.23
    % chichester= 0.21
beta= 0.25; %heating peak shave factor, determined through using the design
approach for optimized hybrid GSHP systems
Rad_BH= 127; %Borehole radius in (mm)

%-----Ground Loop Regression Fits-----
%school
GL_ft= (84.4)+((1.24e4)*alpha)+((-2.05e3)*power(alpha,2));

%4elm
%GL_ft= (187)+((3.21e4)*alpha)+((-1.19e4)*power(alpha,2));
```



```

%chichester
%GL_ft= (384)+((6.36e4)*alpha)+((-1.99e4)*power(alpha,2));

GL= GL_ft*0.3048; %Optimal ground loop length (m)
V_BH_SAGSHP= ((pi*power((200/1000),2))/4)*300; %SAGSHP systems bore-field
volume(m^3)
V_HGSHP= ((pi*power((Rad_BH/1000),2))/4)*GL; %HGSHP system bore-field
volume(m^3)
ThermMas_CF= V_BH_SAGSHP/V_HGSHP; %Thermal mass correction factor

%-----Data Loading-----
demand_hr= xlsread(filename);
PreCool_Results = zeros(36,8);

for z= 1:36

    filename1= [ 'TOU' num2str(z) 'TSPC'];
    TOU_hr= xlsread(filename1);

    %-----Load per Hr to Load per Thirds of an Hr-----

    demand_3rd_hr= zeros(26280,2);
    TOU_3rd_hr= TOU_hr;
    count_1=0;

    for i= 1:8760
        for x= 1:3
            demand_3rd_hr(x+count_1,:)= demand_hr(i,:);
        end
        count_1=count_1+3;
    end

    %-----Data Analysis-----
    metric_demand_hr= demand_hr*0.293; %hourly demand in metric units (kW)
    metric_demand_3rd_hr= demand_3rd_hr*0.293; %3rds of and hour demand in
metric units (kW)

    % Metric demand in 3rds, cooling/heating column vectors
    mcl_3= metric_demand_3rd_hr(:,1);
    mhl_3= metric_demand_3rd_hr(:,2);

    % Metric demand in hrs, cooling/heating column vectors
    mcl= metric_demand_hr(:,1);
    mhl= metric_demand_hr(:,2);

    DD= (mcl_3 - mhl_3);% computing the difference between cooling and
heating loads

    MAXCL= max(mcl);
    MAXHL= max(mhl);
    SUMCL= sum(mcl);
    SUMHL= sum(mhl);

```

```

%----Determining Heating and Cooling Demand after internal adjustment----

%Pre-allocation of the two matrixes in the following loop:

DDC= zeros(26280,1);
DDH= zeros(26280,1);

%Loop utilized to fill DDC & DDH:

for i= 1:26280

    if DD(i,1)>0 %more cooling than heating
        DDC(i,1)=DD(i,1);
    end

    if DD(i,1)<0 %more heating than cooling
        DDH(i,1)=abs(DD(i,1));
    end
end

%----- Determine Hybrid System Capacity-----

if MAXCL>MAXHL
    DesCap= alpha*MAXCL; %cooling dominate building system capacity
    cooling=1;
    heating=0;
end

if MAXHL>MAXCL
    DesCap= beta*MAXHL; %heating dominate building system capacity
    cooling=0;
    heating=1;
end

%--- Determining demand met by GSHP and AUX subsystems-----

%Cooling Pre-allocation
GSHP_Sup_C= zeros(26280,1);
GSHP_CL_C= zeros(26280,1);
GSHP_CS_C= zeros(26280,1);

Aux_Sup_C= zeros(26280,2);
Aux_CL_C= zeros(26280,1);
Aux_CS_C= zeros(26280,1);

C_NL= zeros(26280,1);
C_NS= zeros(26280,1);

%Pre-cooling thermal energy balance tracker
PreC_ThermBal= zeros(26280,1);
Therm_Extra= zeros(26280,1);
GTD= zeros(26280,1);
STD= zeros(26280,1);

```

```

Cap_Inc= zeros(26280,1);

%Heating Pre-allocation
GSHP_Sup_H=zeros(26280,1);
GSHP_CL_H= zeros(26280,1);
GSHP_CS_H= zeros(26280,1);

Aux_Sup_H= zeros(26280,1);
Aux_CL_H= zeros(26280,1);
Aux_CS_H= zeros(26280,1);

H_NL= zeros(26280,1);
H_NS= zeros(26280,1);

%Dual Stage Capacity
PartLoadCap= Stage2_PartLoad*DesCap;

%-----Set-point control-----

%*****HEATING*****

% Filling Net Load Column vector:
%Initial condition
H_NL(1,1)= DDH(1,1)/3;

for i= 2:26280
    H_NL(i,1)= (DDH(i,1)/3)+ H_NL(i-1,1);
end

if DDH(1,1)>0 && DDH(1,1)< DesCap && DDH(1,1)> PartLoadCap %heating load
present in the first 20 mins of Jan 1st and is less than the heating capacity
of the hybrid system
    GSHP_Sup_H(1,1)= DesCap; %kW
    GSHP_CL_H(1,1)= DDH(1,1)/3; %kWh
    GSHP_CS_H(1,1)= GSHP_Sup_H(1,1)/3; %kWh
end

if DDH(1,1)>0 && DDH(1,1)< DesCap && DDH(1,1)< PartLoadCap %heating load
present in the first 20 mins of Jan 1st and is less than the heating capacity
of the hybrid system
    GSHP_Sup_H(1,1)= PartLoadCap; %kW
    GSHP_CL_H(1,1)= DDH(1,1)/3; %kWh
    GSHP_CS_H(1,1)= GSHP_Sup_H(1,1)/3; %kWh
end

if DDH(1,1)>0 && DDH(1,1)> DesCap %heating load is present in the first
20 mins of Jan 1st and is greater than the heating capacity of the hybrid
system
    GSHP_Sup_H(1,1)= DesCap; % kW
    Aux_Sup_H(1,1)= MAXHL-DesCap; % kW

    GSHP_CL_H(1,1)= GSHP_Sup_H(1,1)/3; %kWh
    Aux_CL_H(1,1)= (DDH(1,1)-GSHP_Sup_H(1,1))/3; %kWh

```

```

    GSHP_CS_H(1,1)= GSHP_Sup_H(1,1)/3;           %kWh
    Aux_CS_H(1,1)= Aux_Sup_H(1,1)/3;             %kWh

end

% Control Loop to determine operational authority of GSHP and Aux systems
H_NS(1,1)= Aux_CS_H(1,1)+ GSHP_CS_H(1,1);

for i= 2:26280

    % GSHP control logic
    if H_NL(i,1)>H_NS(i-1,1)
        NLmNS= H_NL(i,1) - H_NS(i-1,1);

        if NLmNS> (DesCap/3)
            GSHP_Sup_H(i,1)= DesCap;
        end

        if (NLmNS< (DesCap/3) || NLmNS == (DesCap/3)) && NLmNS>(PartLoadCap/3)
            GSHP_Sup_H(i,1)= DesCap;
        end

        if NLmNS< (DesCap/3) && (NLmNS< (PartLoadCap/3) || NLmNS ==
(PartLoadCap/3))
            GSHP_Sup_H(i,1)= PartLoadCap;
        end
    else

        GSHP_Sup_H(i,1)= 0;
    end

    % GSHP cumulative supplied in heating mode
    GSHP_CS_H(i,1)= (GSHP_Sup_H(i,1)/3) + GSHP_CS_H(i-1,1);

    % GSHP cumulative load in heating mode
    if (DDH(i,1)>DesCap || DDH(i,1)==DesCap)

        GSHP_CL_H(i,1)= (DesCap/3)+ GSHP_CL_H(i-1,1);
    else

        GSHP_CL_H(i,1)= (DDH(i,1)/3)+ GSHP_CL_H(i-1,1);
    end

%~~~~~

    % Aux control logic
    if H_NL(i,1)>H_NS(i-1,1) && (H_NL(i,1)-H_NS(i-1,1))> (DesCap/3)

        Aux_Sup_H(i,1)= MAXHL- DesCap;
    else

        Aux_Sup_H(i,1)= 0;
    end
end

```

```

% Aux cumulative supplied in heating mode
Aux_CS_H(i,1)= (Aux_Sup_H(i,1)/3) + Aux_CS_H(i-1,1);

% Aux cumulative load in heating mode
if DDH(i,1)>DesCap

    Aux_CL_H(i,1)= ((DDH(i,1)-DesCap)/3)+ Aux_CL_H(i-1,1);
else

    Aux_CL_H(i,1)= Aux_CL_H(i-1,1);
end

% Net Supplied in heating mode, both Aux and GSHP system
H_NS(i,1)= Aux_CS_H(i,1)+ GSHP_CS_H(i,1);
end

%*****COOLING*****

%Net Load Column vector:
%Initial condition
C_NL(1,1)= DDC(1,1)/3;

for i= 2:26280
    C_NL(i,1)= (DDC(i,1)/3)+ C_NL(i-1,1);
end

if DDC(1,1)>0 && DDC(1,1)< DesCap && DDH(i,1)> PartLoadCap %cooling load
present in the first 20 mins of Jan 1st and is less than the heating capacity
of the hybrid system
    GSHP_Sup_C(1,1)= DesCap; %kW
    GSHP_CL_C(1,1)= DDC(1,1)/3; %kWh
    GSHP_CS_C(1,1)= GSHP_Sup_C(1,1)/3; %kWh
end

if DDC(1,1)>0 && DDC(1,1)< DesCap && DDC(1,1)< PartLoadCap %heating load
present in the first 20 mins of Jan 1st and is less than the heating capacity
of the hybrid system
    GSHP_Sup_C(1,1)= PartLoadCap; %kW
    GSHP_CL_C(1,1)= DDC(1,1)/3; %kWh
    GSHP_CS_C(1,1)= GSHP_Sup_C(1,1)/3; %kWh
end

if DDC(1,1)>0 && DDC(1,1)> DesCap %cooling load present in the first 20
mins of Jan 1st and is greater than the heating capacity of the hybrid system
    GSHP_Sup_C(1,1)= DesCap; %kW
    Aux_Sup_C(1,1)= MAXCL-DesCap; %kW
    Aux_Sup_C(1,2)= 0; %kW

    GSHP_CL_C(1,1)= GSHP_Sup_C(1,1)/3; %kWh
    Aux_CL_C(1,1)= (DDC(1,1)-GSHP_Sup_C(1,1))/3; %kWh

```



```

end

if (NLmNS< ((DesCap + Cap_Inc(i,1))/3) || NLmNS == ((DesCap +
Cap_Inc(i,1))/3)) && NLmNS>((Stage2_PartLoad*(DesCap + Cap_Inc(i,1))/3)
    GSHP_Sup_C(i,1)= DesCap + Cap_Inc(i,1);
end

if NLmNS< ((DesCap + Cap_Inc(i,1))/3) && (NLmNS<
((Stage2_PartLoad*(DesCap + Cap_Inc(i,1))/3) || NLmNS ==
((Stage2_PartLoad*(DesCap + Cap_Inc(i,1))/3))
    GSHP_Sup_C(i,1)= (Stage2_PartLoad*(DesCap + Cap_Inc(i,1)));
end
else

    GSHP_Sup_C(i,1)= 0;
end

% GSHP control logic
elseif PR==0

    if C_NL(i,1)>C_NS(i-1,1)
        NLmNS= C_NL(i,1) - C_NS(i-1,1);

        if NLmNS> (DesCap/3)
            GSHP_Sup_C(i,1)= DesCap;
        end

        if (NLmNS< ((DesCap/3)) || NLmNS == ((DesCap/3))) &&
NLmNS>((PartLoadCap/3))
            GSHP_Sup_C(i,1)= DesCap;
        end

        if NLmNS< ((DesCap/3)) && (NLmNS< ((PartLoadCap/3)) || NLmNS ==
((PartLoadCap/3)))
            GSHP_Sup_C(i,1)= PartLoadCap;
        end
    else

        GSHP_Sup_C(i,1)= 0;
    end
end
% GSHP cumultive supplied in cooling mode
GSHP_CS_C(i,1)= (GSHP_Sup_C(i,1)/3) + GSHP_CS_C(i-1,1);

% GSHP cumultive load in cooling mode
if DDC(i,1)>DesCap

    GSHP_CL_C(i,1)= (DesCap/3)+ GSHP_CL_C(i-1,1);
else

    GSHP_CL_C(i,1)= (DDC(i,1)/3)+ GSHP_CL_C(i-1,1);
end

```

```

%|||||||||||||||||||||||||||||||||||||||||||||||||||||||||||||||||||||||||||||||||||||

%-----Aux control logic-----
%Primary logic
if PR==1 % Aux operaton with pre-cool benifit
    if C_NL(i,1)>C_NS(i-1,1) && (C_NL(i,1)-C_NS(i-1,1))> ((DesCap +
Cap_Inc(i,1))/3)
        Aux_Sup_C(i,1)= MAXCL - DesCap;
    else
        Aux_Sup_C(i,1)= 0;
    end
elseif PR==0 %Aux operation with no pre-cool benifit
    if C_NL(i,1)>C_NS(i-1,1) && (C_NL(i,1)-C_NS(i-1,1))> ((DesCap/3))
        Aux_Sup_C(i,1)= MAXCL - DesCap;
    else
        Aux_Sup_C(i,1)= 0;
    end
end

    Aux_CS_C(i,1)= (Aux_Sup_C(i,1)/3) + Aux_CS_C(i-1,1);
    % Aux cumulitive supplied in cooling mode

% Aux cumulative load in cooling mode
if DDC(i,1)>DesCap

    Aux_CL_C(i,1)= ((DDC(i,1)-DesCap)/3)+ Aux_CL_C(i-1,1);
else

    Aux_CL_C(i,1)= Aux_CL_C(i-1,1);
end

% Net Supplied in heating mode, both Aux and GSHP system
C_NS(i,1)= Aux_CS_C(i,1)+ GSHP_CS_C(i,1);

%//////////Pre-Cooling logic//////////

if flag==0
    if PC_Shoulder == 1
        if i>6479 && i<10873
            if TOU_3rd_hr(i,2)==1 && TOU_3rd_hr(i,1)<0.1
                Aux_Sup_C(i,2)= MAXCL- DesCap;
            end
        end
    end

    if PC_Peak == 1
        if i>13034 && i<17497
            if TOU_3rd_hr(i,2)==1 && TOU_3rd_hr(i,1)<0.1
                Aux_Sup_C(i,2)= MAXCL- DesCap;
            end
        end
    end
end
end
end

```



```

elseif CCOP_CNT(i,1) == 3
    COPGC(i,1)= COP_CF*(COPGC3 + COP_Inc(i,1));
elseif CCOP_CNT(i,1) == 4
    COPGC(i,1)= COP_CF*(COPGC4 + COP_Inc(i,1));
elseif CCOP_CNT(i,1) > 4
    COPGC(i,1)= COP_CF*(COPGC5 + COP_Inc(i,1));
end
end

if GSHP_Sup_C(i,1)== PartLoadCap
    if CCOP_CNT(i,1) == 1
        COPGC(i,1)= COP_CF*COPGC1;
    elseif CCOP_CNT(i,1) == 2
        COPGC(i,1)= COP_CF*COPGC2;
    elseif CCOP_CNT(i,1) == 3
        COPGC(i,1)= COP_CF*COPGC3;
    elseif CCOP_CNT(i,1) == 4
        COPGC(i,1)= COP_CF*COPGC4;
    elseif CCOP_CNT(i,1) > 4
        COPGC(i,1)= COP_CF*COPGC5;
    end
end

%-----
    if GSHP_Sup_C(i,1)== 0
        COPGC(i,1)=0.01;
    end

end

%*****HEATING*****

% 2nd order polynomial regression for cooling COP data:
%COPGC= (4e-8)*t^2 - (4e-4)*t + 3.8377;

%discrete transient COP's of GSHP in heating mode:
COPGH1= 3.66; %Averaged COP from the first 20 mins range of heating data
COPGH2= 3.12; %Averaged COP from the second 20 mins range of heating data
COPGH3= 2.90; %Averaged COP from the third 20 mins range of heating data
COPGH4= 2.74; %Averaged COP from the fourth 20 mins range of heating data
COPGH5= 2.64; %Averaged COP from the data greater than the fourth 20 mins
range

COPGH = zeros(26280,1);
HCOP_CNT = zeros(26280,1);

%Initial Condition
if GSHP_Sup_H(1,1)> 0 && GSHP_Sup_H(1,1)== DesCap
    COPGH(1,1)= COPGH1;
elseif GSHP_Sup_H(1,1)> 0 && GSHP_Sup_H(1,1)== PartLoadCap
    COPGH(1,1)= COP_CF*COPGH1;
else
    COPGH(1,1)=0.01;
end

```

```

HCOP_CNT(1,1) = 1;

for i= 2:26280

    if GSHP_Sup_H(i,1)> 0 && GSHP_Sup_H(i-1,1)> 0
        HCOP_CNT(i,1)= HCOP_CNT(i-1,1) +1;
    else
        HCOP_CNT(i,1)= HCOP_CNT(i-1,1);
    end
    if GSHP_Sup_H(i,1)==0
        if HCOP_CNT(i-1,1)==1
            HCOP_CNT(i,1)= 1;
        end
        if HCOP_CNT(i-1,1)> 1
            HCOP_CNT(i,1)= HCOP_CNT(i-1,1)-1;
        end
    end

    if GSHP_Sup_H(i,1)== DesCap

        if HCOP_CNT(i,1) == 1
            COPGH(i,1)= COPGH1;
        elseif HCOP_CNT(i,1) == 2
            COPGH(i,1)= COPGH2;
        elseif HCOP_CNT(i,1) == 3
            COPGH(i,1)= COPGH3;
        elseif HCOP_CNT(i,1) == 4
            COPGH(i,1)= COPGH4;
        elseif HCOP_CNT(i,1) > 4
            COPGH(i,1)= COPGH5;
        end
    end

    if GSHP_Sup_H(i,1)== PartLoadCap

        if HCOP_CNT(i,1) == 1
            COPGH(i,1)= COP_CF*COPGH1;
        elseif HCOP_CNT(i,1) == 2
            COPGH(i,1)= COP_CF*COPGH2;
        elseif HCOP_CNT(i,1) == 3
            COPGH(i,1)= COP_CF*COPGH3;
        elseif HCOP_CNT(i,1) == 4
            COPGH(i,1)= COP_CF*COPGH4;
        elseif HCOP_CNT(i,1) > 4
            COPGH(i,1)= COP_CF*COPGH5;
        end
    end

    if GSHP_Sup_H(i,1)== 0
        COPGH(i,1)=0.01;
    end
end

%-----Power Consumption-----

```

```

%Pre-allocation
Con_Elec_Heating=zeros(26280,1);
Con_Elec_Cooling= zeros(26280,1);
GSHP_EleCon_Cooling= zeros(26280,1);
Con_NatGas_Heating= zeros(26280,1);

for i= 1:26280

    %Heating:
    Con_Elec_Heating(i,1)= (GSHP_Sup_H(i,1)/3)/COPGH(i,1);
    Con_NatGas_Heating(i,1)= (((Aux_Sup_H(i,1)/3)/kWh_m3)/AuxEffH);

    %Cooling:
    Con_Elec_Cooling(i,1)=
((GSHP_Sup_C(i,1)/3)/COPGC(i,1))+((Aux_Sup_C(i,1)/3)/COPCT)+
((Aux_Sup_C(i,2)/3)/(PC_CT_CF*COPCT));
    GSHP_EleCon_Cooling(i,1)= ((GSHP_Sup_C(i,1)/3)/COPGC(i,1));
end

TCDM= sum(GSHP_Sup_C)/sum(DDC);
THDM= sum(GSHP_Sup_H)/sum(DDH);
TEDM= (sum(GSHP_Sup_C)+sum(GSHP_Sup_H))/(sum(DDC)+sum(DDH));
Total_Electircal_Consh= sum( Con_Elec_Heating);
Total_Electircal_Consc= sum( Con_Elec_Cooling);
Total_NG_Consh= sum( Con_NatGas_Heating);

%Net usage:
Total_Elec_Con= Con_Elec_Cooling + Con_Elec_Heating;

Total_Electical_Con= sum(Total_Elec_Con);

GSHP_ElecCons_Total_C= sum(GSHP_EleCon_Cooling);
%-----Operating Cost-----
Cost_Elec= zeros(26280,1);
for i= 1:26280
    %Cost of Electricity:
    Cost_Elec(i,1)= Total_Elec_Con(i,1)*TOU_3rd_hr(i,1);
end
%Cost of Natural Gas:
Cost_NG= Con_NatGas_Heating* NG_Rate;
Total_Cost= Cost_Elec + Cost_NG;
Total_Electical_Cost= sum(Cost_Elec);
Total_NG_Cost= sum(Cost_NG);
Total_AnnualCooling_Cost= sum(Total_Cost);
%-----
End

```

Code terminated:

APPENDIX D – Bore-field Thermal Response and Heat Pump Performance Correlations

The following Figure 33, Figure 34, and Figure 35 present a graphical representation of the correlations used in the HGSHP numerical performance prediction model. The correlations were generated from the data presented in [34]. In Figure 33, the linear correlation presented is normalized relative to bore-field volume; volume of the SAGSHP [34] divided by the volume of the HGSHP being simulated. This is done to correct for the variation (with respect to [34]) of bore-field/soil volumes, where bore-field/soil volume is proportional to the amount of energy extraction required to decrease the temperature of the body.

Figure 33: Bore-field/ground thermal response correlation

The EST variation is determined by implementing a ground-loop efficiency factor. The EST correlation is the product of the ground-loop efficiency factor and the normalized correlation presented in Figure 33. The difference in EST is then used to predict the variation in ground-loop heat exchange rate and GSHP system COP, presented in Figure 34 and Figure 35, respectively.

Figure 34: Ground-loop heat exchange rate correlation

Figure 35: GSHP COP correlation

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