APPLICATION MANUAL

Improving Geothermal Heat Pump Air Conditioning Efficiency with Wintertime Cooling using Seasonal Thermal Energy Storage (STES)

ESTCP Project EW-201013

NOVEMBER 2016

Ronald W. Falta and Fred Molz
Clemson University

Charles Newell
GSI Environmental, Inc.

Distribution Statement A
This document has been cleared for public release

REPORT DOCUMENTATION PAGE	*Form Approved* *OMB No. 0704-0188*

1. REPORT DATE *(DD-MM-YYYY)*	**2. REPORT TYPE**	**3. DATES COVERED** *(From - To)*
01/11/2016	Manual	July 2010 - November 2016

4. TITLE AND SUBTITLE
Improving Geothermal Heat Pump Air Conditioning Efficiency with Wintertime Cooling using Seasonal Thermal Energy Storage (STES)

5a. CONTRACT NUMBER
W912HQ-10-C-0027

5b. GRANT NUMBER

5c. PROGRAM ELEMENT NUMBER

6. AUTHOR(S)
Ronald W. Falta and Fred Molz
Clemson University

Charles Newell
GSI Environmental, Inc.

5d. PROJECT NUMBER
EW-201013

5e. TASK NUMBER

5f. WORK UNIT NUMBER

7. PERFORMING ORGANIZATION NAME(S) AND ADDRESS(ES)
Clemson University
P.O. Box 340919, Brackett Hall, Room 340C
Clemson, SC 29634-0919

8. PERFORMING ORGANIZATION REPORT NUMBER

9. SPONSORING/MONITORING AGENCY NAME(S) AND ADDRESS(ES)
Environmental Security Technology Certification Program Program Office
4800 Mark Center Drive
Suite 17D03
Alexandria, VA 22350-3605

10. SPONSOR/MONITOR'S ACRONYM(S)
ESTCP

11. SPONSOR/MONITOR'S REPORT NUMBER(S)

12. DISTRIBUTION/AVAILABILITY STATEMENT
Approved for public release; distribution is unlimited.

13. SUPPLEMENTARY NOTES
N/A

14. ABSTRACT
The purpose of this manual is to describe the use of the Seasonal Thermal Energy Storage (STES) technology, particularly through the employment of wintertime cooling. The technology is a means to solve excessive temperature increase in ground source heat pump loops and subsurface media. It gives an example of a recently demonstrated application and provides Energy Managers (EM) and Facility managers (FM) the background to quickly prioritize and document application projects to increase energy efficiency and occupant comfort.

15. SUBJECT TERMS
Seasonal Thermal Energy Storage (STES) technology, energy efficiency, geothermal heat pump systems.

16. SECURITY CLASSIFICATION OF: a. REPORT	b. ABSTRACT	c. THIS PAGE	**17. LIMITATION OF ABSTRACT**	**18. NUMBER OF PAGES**	**19a. NAME OF RESPONSIBLE PERSON** / **19b. TELEPHONE NUMBER** *(Include area code)*
Unclassified	Unclassified	UU	UL	56	Ronald Falta / 864-656-0125

Standard Form 298 (Rev. 8/98)
Prescribed by ANSI Std. Z39.18

Page Intentionally Left Blank

Page Intentionally Left Blank

Table of Contents

List of Tables

List of Figures

Acronyms

ASHRAE	American Society for Heating, Refrigeration and Air-conditioning Engineers
BTU	British Thermal Unit
EER	energy efficiency ratio
GSI	GSI Environmental, Inc.
HDPE	high density polyethylene
HVAC	heating, ventilation and air conditioning
IGHSPA	International Ground Source Heat Pump Association
kBTU	thousand BTU
kW	kilowatt
kWhr	kilowatt hour
MCAS	Marine Corps Air Station
VFD	variable frequency drive

1.0 INTRODUCTION

1.1 Purpose

The purpose of this manual is to describe the use of the Seasonal Thermal Energy Storage (STES) technology, particularly through the employment of wintertime cooling. The technology is a means to solve excessive temperature increase in ground source heat pump loops and subsurface media. It gives an example of a recently demonstrated application and provides Energy Managers (EM) and Facility managers (FM) the background to quickly prioritize and document application projects to increase energy efficiency and occupant comfort. It suggests industry tools that will assist the EM/FM in doing needed calculations to justify the acquisition of equipment to assist in optimizing ground source heat pumps.

1.2 Geothermal Heat Pump Systems

Geothermal heat pump systems use the ground as a heat source and heat sink to heat and cool buildings. These systems, also known as ground source heat pump systems, use reversible heat pumps (Figure 1.1) to either extract or reject heat into a water loop (the ground loop) that runs through the building and interacts with the subsurface through trenches, wells, or boreholes.

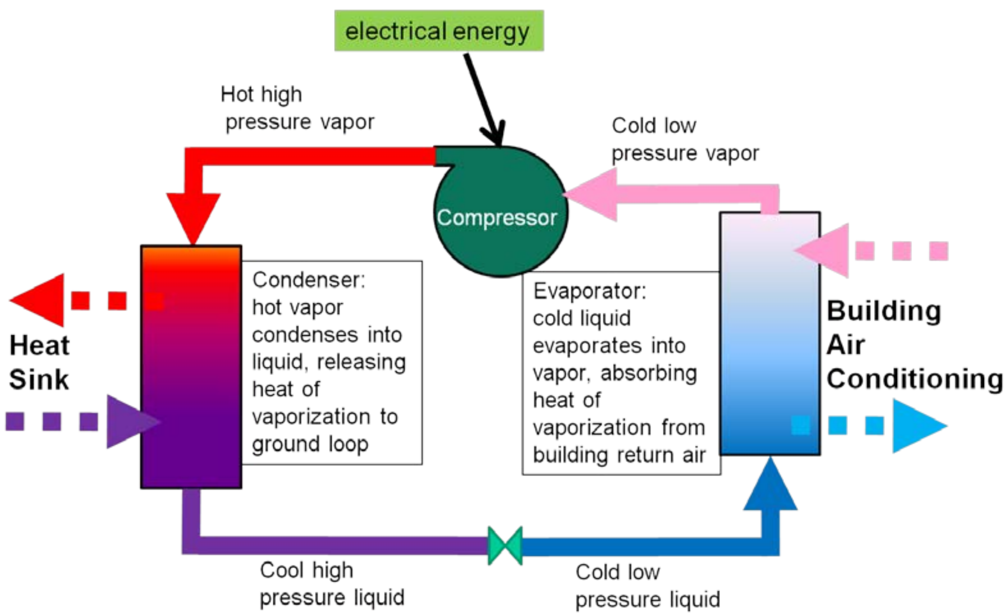

Figure 1.1. Schematic diagram of a heat pump during cooling operation.

When the building is being cooled, heat from the building is absorbed by a liquid refrigerant in the evaporator, converting the liquid into a vapor. The cool, low-pressure vapor is compressed in an electrically driven compressor to convert it into a hot, high-pressure vapor. A heat sink (the ground loop) is used to remove heat from the hot vapor, causing it to condense back into a liquid. The liquid is then routed back to the evaporator to complete the cycle. The basic principle of operation is that the building heat is transferred using the latent heat of refrigerant vaporization from the heat source to the heat sink. This is an extremely efficient method of heat transfer, because several units of heat energy can be transferred per unit of electricity consumed by the compressor.

When the building is being heated, the refrigeration cycle is reversed, and heat is extracted from the heat source (the ground loop) to evaporate the liquid refrigerant. The refrigerant vapor condenses in a coil inside the building, releasing heat to warm up the building.

There are three main types of ground loops: open loops using wells, closed loops using trenches, and closed loops that use boreholes. With an open loop, groundwater is pumped from a well, through the heat pump system, and back into the ground through another well. This type of system can be very effective, but it requires access to a productive aquifer with associated permitting and water chemistry considerations.

Closed loop systems use sealed piping to move a mixture of water and antifreeze through the ground. Small household geothermal systems often use shallow trenches for these closed loops, but trenches become impractical for larger buildings, where the necessary length of the ground loop may be thousands of feet. The most common ground loop configuration for larger buildings consists of an array of vertical boreholes extending up to several hundred feet deep into the ground with a horizontal spacing of 20 feet or more (Figure 1.2).

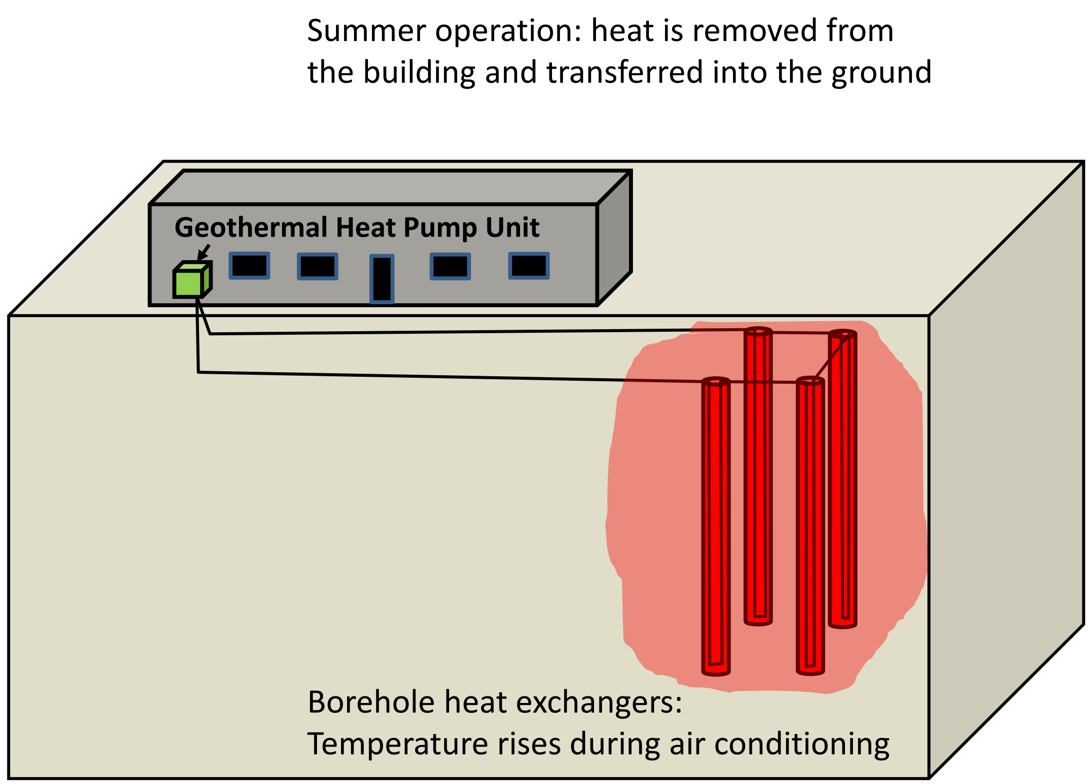

Figure 1.2. A closed loop geothermal heat pump system with borehole heat exchangers operating in air conditioning mode. Heat is rejected from the building into the ground.

These vertical boreholes are typically constructed by drilling a 6-inch diameter borehole. A high-density polyethylene (HDPE) U-tube is installed in the borehole and is grouted into place using a thermally conductive grout. This design isolates the ground loop fluid from the groundwater system, and heat transfer between the ground loop and the subsurface occurs by thermal conduction.

When the heat pump system is in air conditioning mode, the ground loop rejects the building heat into the ground loop, resulting in an increase in temperature in the subsurface (Figure 1.2). When the heat pump is in heating mode, heat is extracted from the ground loop and delivered to the building, causing the ground to cool (Figure 1.3).

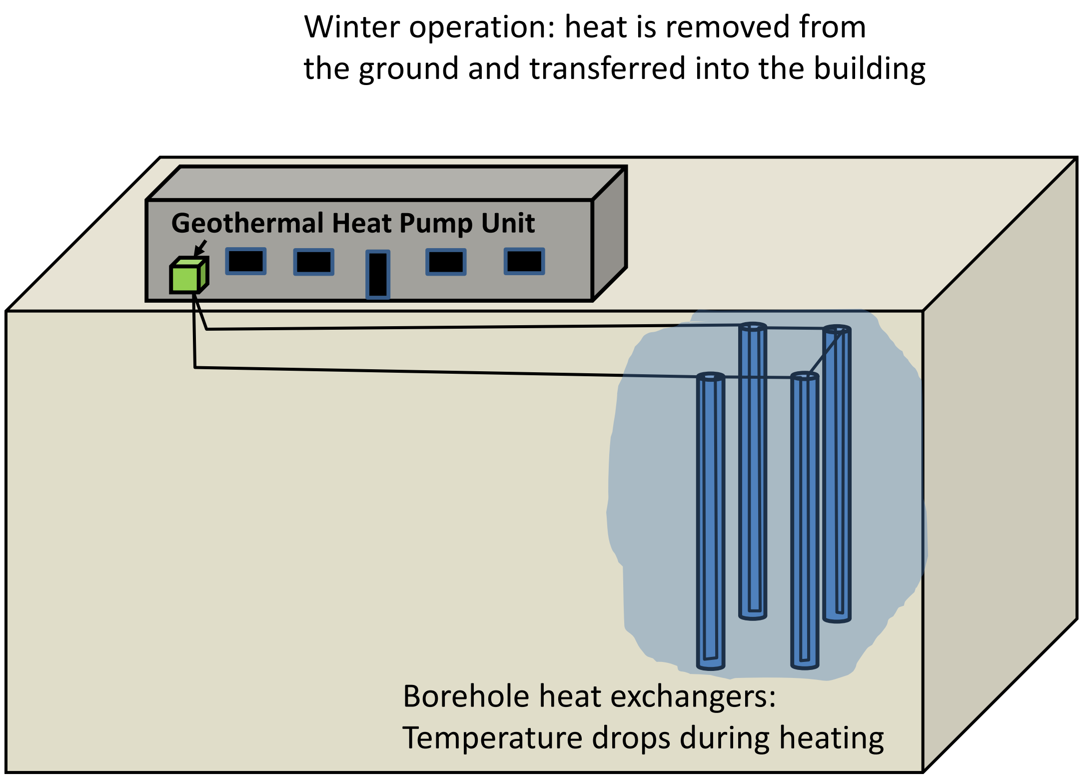

Figure 1.3. A closed loop geothermal heat pump system with borehole heat exchangers operating in heating mode. Heat extracted from the ground and delivered to the building.

When the building heating and cooling loads (including the waste heat from the heat pumps) are exactly balanced, then the heating of the subsurface during the summer air conditioning period is offset by the cooling of the subsurface during the winter heating period. The case of balanced heating and cooling loads is an ideal condition for use of geothermal heat pumps. Although the ground temperature rises some in the summer, the average ground loop temperature remains well below the outside air temperature, making it an efficient heat sink for the heat pump. Similarly, in the winter, the ground temperature drops some, but it remains higher than the outside air temperature, making it an efficient heat source for the heat pumps.

It appears, however, that many medium to large buildings in the United States do not have balanced heating and cooling loads, and they tend to be strongly cooling dominated. This imbalance may lead to substantial heating of geothermal ground loops, which causes the heat pump performance and efficiency to degrade over time.

1.3 Cooling-Dominated Buildings and MCAS Beaufort Case Study

Commercial and institutional buildings in the United States commonly have unbalanced heating and cooling loads that are dominated by cooling. The load imbalance arises from waste heat caused by lighting and other appliances, industrial machinery, communications and computing equipment, and people. Medium to large buildings have interior rooms that may never require heating, and the ratio of building surface area to volume decreases as the building size increases. Geothermal heat pumps themselves generate substantial waste heat, equal to about 20-25% of the building cooling load. This heat must also be removed from a building to cool it.

For example: At the Marine Corps Air Station in Beaufort, South Carolina, most of the major buildings are heated and cooled using geothermal heat pump systems. As part of the ESTCP EW-201013 demonstration, three of these building ground loop systems have been instrumented to collect data at 15-minute intervals over the past several years. These buildings include the Base Headquarters, Building 601; the Structural Fire Station, Building 2085; and the Military Police station, Building 584. These buildings range in size from 12,500 to 24,000 sq. ft. and have geothermal ground loops that consist of 24 to 39 vertical boreholes, each about 300 ft. deep.

The heating and cooling loads in these buildings can be measured by comparing the geothermal loop temperature entering and leaving the building. During heat pump air conditioning, water from the borehole heat exchangers enters the building and absorbs the heat rejected by the heat pumps inside the building. The heated water exits the building and is routed back into the boreholes. Similarly, during building heating, heat is extracted from the loop, and the loop temperature exiting the building is lower than the entering temperature.

Multiplying the loop entering and exiting temperature difference by the loop flowrate (and correcting the units) gives the building heating or cooling load at that point in time. Those data can be integrated with respect to time to get monthly and yearly values for the building heating and cooling loads.

The three buildings monitored as part of this project (Buildings 601, 2085, and 584) were found to be extremely cooling dominated on an annual basis. As evidenced by the 15-minute data, these buildings are only heated on very cold winter nights and early mornings. At other times, the buildings are undergoing cooling, even in January.

Figure 1.4 shows the 12,550-sq. ft. Military Police building (Building 584). This building was converted to a geothermal heat pump system in 2004, with twenty-four 300-foot-deep borehole heat exchangers located beneath the parking lot.

The measured ground loop temperatures from Building 584 during the latter half of January 2013 are shown in Figure 1.5. The blue line shows the ground loop water temperature entering the building, and the red line shows the ground loop temperature leaving the building. The green line shows the outside air temperature. Except for a few brief periods during cold nights, the exiting loop temperature is well above the entering loop temperature, indicating that the building is being cooled rather than heated.

The August 2012 ground loop temperatures for Building 584 are shown in Figure 1.6. During this period, the building was continuously air conditioned, with peak cooling loads occurring in the late afternoons.

Figure 1.4. Building 584, MCAS, Beaufort, SC.

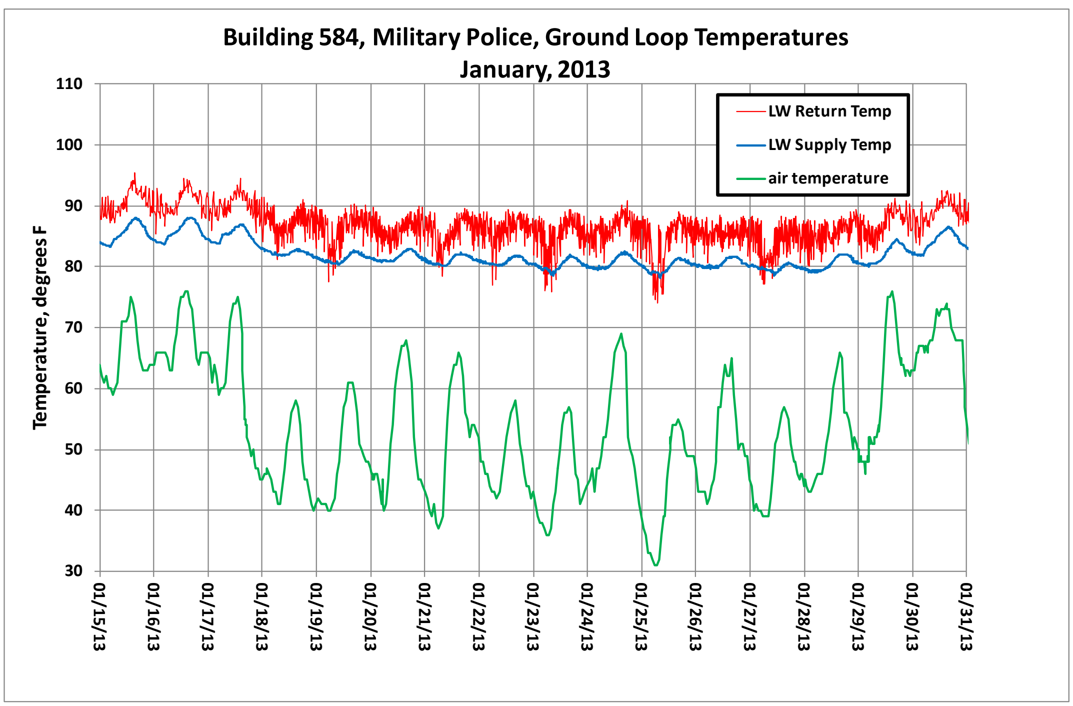

Figure 1.5. Ground loop and outside air temperatures at Building 584 in January 2013.

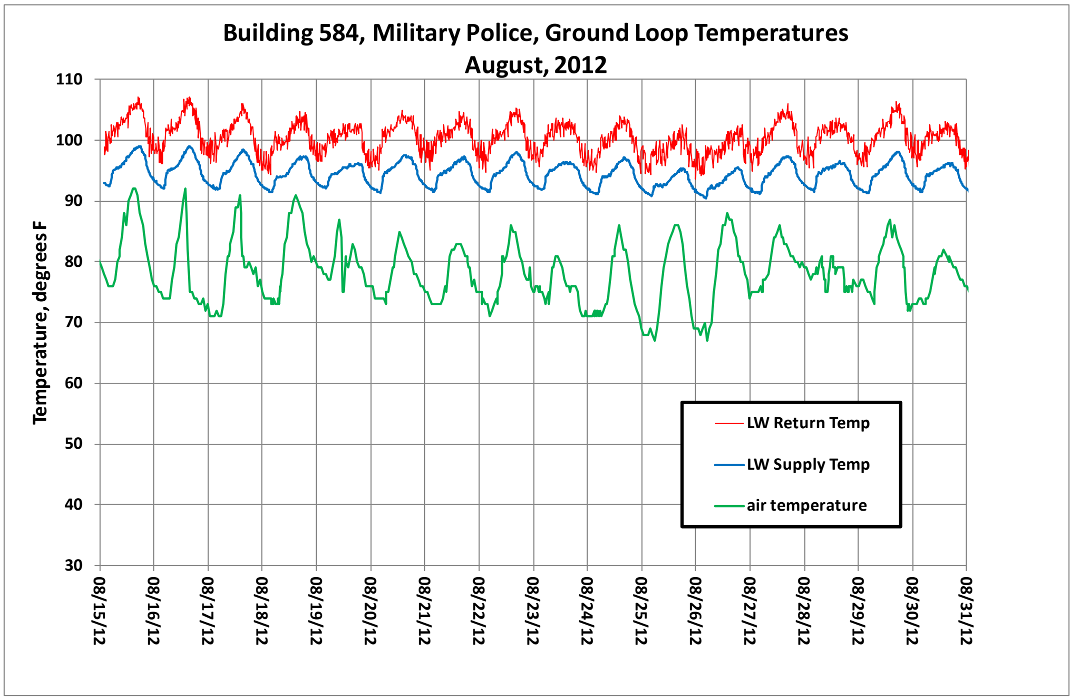

Figure 1.6. Ground loop and outside air temperatures at Building 584 in August 2012.

The loop temperature and loop flow rate data from Building 584 were used to calculate building heating and cooling loads over the course of several years. The average heating and cooling loads (kBTU per month) are listed in Table 1.1. Over the course of the year, 99.5% of the building load is for cooling. Buildings 601 and 2085 show a similar behavior and are also extremely cooling dominated.

Table 1.1. Average observed heating and cooling loads (kBTU) for Building 584 at the MCAS, Beaufort, SC.

.

	heating	cooling
January	1844.00	21400.00
February	2100.00	21623.00
March	180.00	48880.00
April	0.00	65600.00
May	0.00	90241.00
June	0.00	120208.00
July	0.00	133906.00
August	0.00	129393.00
September	0.00	111301.00
October	0.00	80040.00
November	256.00	34747.00
December	105.00	32647.00

The continuous heat rejection from Building 584 and the other buildings at the MCAS, Beaufort has led to increases in the ground loop temperatures over time. The undisturbed ground temperature below the depth of seasonal variation (a few tens of feet) is usually close to the average yearly air temperature. Using the model developed by Xing (2014), as implemented in the GLHEPro program (IGSHPA, 2016), the ground temperature at Beaufort, SC is estimated to be 67 ^{o}F.

It is clear from Figures 1.5 and 1.6 that the geothermal ground loop temperatures for Building 584 are far above the background value. The late-summer 2012 loop temperature entering the building was in the mid-90s, while the temperature leaving the building often exceeded 100 ^{o}F. These temperatures are far above the natural ground temperature and are also well above the outside air temperature. These high ground loop temperatures degrade the performance of the heat pumps during air conditioning. Table 1.2 shows heat pump cooling efficiency for a water source heat pump. (The use of brand names does not constitute an endorsement and is used only for illustrative purposes.)

Table 1 2. Heat pump cooling efficiency as a function of loop temperature. Data are from the GLHEPro (IGSHPA, 2016) standard library of heat pumps.

Trane WPHF021 heat pump performance ratings	
Ground loop temperature	**Cooling** energy efficiency ratio (EER), kBTUh/kW
45 ^{o}F	28.0 (+133%)
60 ^{o}F	21.5 (+79%)
70 ^{o}F	18.5 (+54%)
80 ^{o}F	16.0 (+33%)
90 ^{o}F	13.8 (+15%)
100 ^{o}F	12.0

The cooling efficiency is expressed in units of kBTU/hr of cooling per kW of electrical power. The cooling efficiency drops by about 1-2% per degree F increase. Once the loop temperature exceeds about 110 ^{o}F, it may be necessary to shut down the heat pumps. During heating, the increased loop temperature improves the heat pump efficiency, but since heating only represents one-half of one percent of the total load, this is of no real benefit at Beaufort.

The heating of the subsurface by the borehole heat exchangers represents a long-term damage to the natural system. Once a large volume of the subsurface associated with a building geothermal heat pump system has been excessively heated, it is no longer as suitable for use as a heat sink. This means that performance of the geothermal heat pump system is permanently degraded unless some of the excess heat is removed from the subsurface. It also limits the applicability of future applications of geothermal heat pump systems at the location.

The occurrence of unbalanced, cooling-dominated buildings is not unique to the MCAS Beaufort site. It is a widespread characteristic of medium to large commercial and institutional buildings across most of the country. To illustrate this point, building heating and cooling loads were calculated for a variety of locations using the eQUEST (Hirsch & Associates, 2016) building energy simulation tool (http://www.doe2.com/equest/). The eQUEST tool uses the DOE-2 energy modeling program that was developed by the U.S. Department of Energy Lawrence Berkeley National Laboratory. The eQUEST program uses a detailed description of the building layout, construction, equipment, usage, and local weather conditions to perform hourly simulations of heating and cooling loads and energy consumption over the course of a year.

For individual projects, the building simulation program contains templates for dozens of typical buildings (office buildings, schools, multi-family apartments, restaurants, retail buildings, motels, hospitals, etc.). Each of these building templates is populated with realistic default values for the important building layout, construction, and usage details. These default values can be entered manually in the 43 pages of building characteristics that are used in each simulation. The program also allows the user to select the location, and it uses long-term average weather files for each location (with about 300 locations available in North America).

Building load simulations were performed for a hypothetical 25,000-sq. ft., two-story office building that is primarily occupied during normal weekday working hours. This building has a square footprint, with metal frame construction, and insulated exterior walls and roof surfaces. The building details used in these simulations are the default values for the building type “Office Bldg, Two Story” in eQUEST. The eQUEST program computes the building loads every hour for one year. Those results can be processed in a separate program called the Peak Load Analysis Tool that is provided with the GLHEPro geothermal heat pump simulation model (IGSHPA, 2016). The Peak Load Analysis Tool is used to covert the hourly eQUEST results into monthly building heating and cooling loads (which is the input format for the GLHEPro program).

Table 1.3 shows the computed building loads for a 25,000-sq. ft., two-story office building in Beaufort, SC. These simulated monthly loads are similar to what we observed at Military Police Building 584 at the MCAS, Beaufort (Table 1.1). The simulated heating and cooling loads show an extreme imbalance, with 98.6% of the total load going to cool the building. This simulated building cooling/heating load imbalance is very close to the observed value of 99.5% for Building 584.

Using the same methodology, the eQUEST program was used to compute the building heating and cooling loads for a two-story office building located in 33 cities across the United States (Figure 1.7). The bar graphs associated with each city show the annual heating loads (in red) and the cooling loads (in blue) in units of million BTUs. Out of the 33 cities, only one – Anchorage, AK – is heating dominated, and only two – Helena, MT and Mineapolis, MN – have balanced heating and cooling loads. The remaining 30 cities are moderately to very strongly cooling dominated.

Table 1.3. Simulated monthly heating and cooling loads (kBTU) for a hypothetical 25,000-sq. ft., two-story office building in Beaufort, SC.

	Heating	Cooling
January	6653	7027
February	2942	11875
March	611	37932
April	2	67894
May	0	90897
June	0	120145
July	0	129384
August	0	133305
September	0	114993
October	0	72249
November	45	37997
December	2126	14217

Locations in the Southeast, Florida, Gulf Coast, the Southwest, and Hawaii were extremely cooling dominated, with very low heating loads (<5% of the total). In the Northeast, the buildings were still strongly cooling dominated, but with larger heating loads (~25% of the total). In the Upper Midwest and Northwest, the buildings also tended to be cooling dominated except for Anchorage, AK; Minneapolis, MN; and Helena, MT. The other locations in those regions had cooling loads that were two to four times larger than the heating loads.

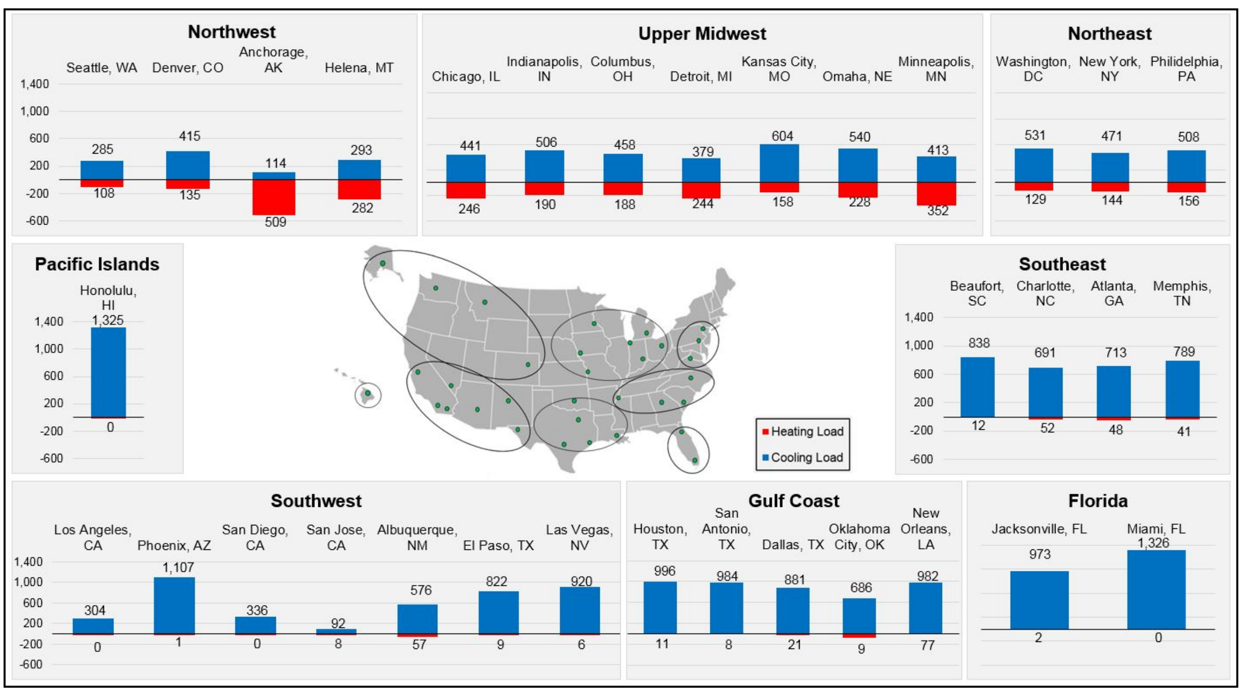

Figure 1.7. Simulated annual heating (red) and cooling (blue) loads in million BTUs for a 25,000-sq. ft., two-story office building in different U.S. cities.

1.4 Potential Approach for Mitigating Heat Buildup - Hybrid Geothermal Heat Pump Systems

There are two primary methods for reducing the build-up of heat in borehole heat exchangers subject to cooling-dominated loads. The most common method is to increase the size of the ground loop by installing more boreholes or by making them deeper. Geothermal heat exchanger simulation programs such as GLHEPro (IGSHPA, 2016) are used to predict future ground loop temperatures given the building loads, location, heat pump characteristics, subsurface thermal properties, and the geometry and properties of the borehole heat exchangers. With this design process, the size of the ground loop is increased until the predicted future temperature rise falls within an acceptable range.

The major disadvantage of increasing the ground loop size is that it may greatly increase the system capital cost. The cost of drilling and installing the borehole heat exchangers is a major part of the overall system expense, with drilling costs ranging from about $10 per foot to $20 per foot or more. Moreover, space may not be available for increasing the size of the ground loop beyond a certain point. Finally, while increasing the size of the ground loop decreases the rate of subsurface temperature increase, it does not eliminate the problem.

A more cost effective option for reducing ground loop heat build-up involves the use of a supplemental cooling device, such as a cooling tower or dry fluid cooler (IGSHPA, 2016). These systems are known as hybrid geothermal heat pump systems. With a hybrid system, the cooling device is used mainly during the peak cooling months to reduce the excess heat rejection into the ground loop. If the cooling device capacity is high enough, it is possible to remove enough heat to balance the heating and cooling load delivered to the ground loop, thus eliminating long-term ground loop heating without increasing the size of the ground loop. This heat removal comes at the cost of added electricity use, water use (for cooling towers), and maintenance.

Several simulation-based optimization methods have been proposed for hybrid geothermal heat pump system design (Cullin and Spitler, 2010; Cullin, 2008; IGSHPA, 2016; Kavanaugh, 1998; Xu, 2007; Hackel et al., 2009; Chaisson and Yavuzturk, 2009). Given permissible ground loop temperature limits, capital costs for the ground loop and cooling device, and the cooling device operating costs, these methods optimize the size of the ground loop and cooling device to minimize the system costs.

Hybrid systems most commonly use cooling towers, with operation mainly in the peak cooling months of summer. Cooling towers remove heat primarily through water evaporation, although some cooling also occurs due to sensible heat transfer. Cooling towers are very effective heat transfer devices, but they consume significant amounts of water, and they have relatively high maintenance requirements associated with the process water. Summertime operation of cooling towers also adds to electrical demand during peak electricity use periods.

Dry fluid coolers are an alternative to cooling towers that have the advantage of not requiring any process water or associated maintenance. Dry fluid coolers remove heat through sensible heat transfer, and the rate of heat rejection depends on the temperature difference between the water entering the cooler and the outside air temperature. Dry fluid coolers are less efficient than cooling towers during summertime operation, but they can be operated in the wintertime with very high efficiency and during periods of low electricity demand.

1.5 Another Approach - Wintertime Ground Loop Cooling with Dry Fluid Coolers

An alternative approach to managing ground loop temperatures involves the use of dry fluid coolers operated mainly in the wintertime. Dry fluid coolers are similar in operation to automobile radiators. The ground loop fluid (a mixture of water and antifreeze) is pumped through copper coils that are attached to aluminum fins. Outside air is pulled across the coils using fans (Figure 1.8) to extract heat from the loop water.

Dry fluid coolers have a nominal heat rejection rating that corresponds to a specific set of operating conditions. Commercial dry fluid coolers range in rated capacity from about 24 kBTU/hr (a 2-ton cooler) up to about 1200 kBTU/hr (a 100-ton cooler) and have anywhere from one to eight fans (typically one horsepower each). Each fluid cooler is designed to operate within a certain water flow rate range, and flow rates that are below the design range tend to result in a linear decrease in performance.

Dry fluid cooler manufacturers publish tables and graphs that show the cooler performance under different conditions (water flow rate, outside air temperature, and entering water temperature). For a particular dry fluid cooler with a specified flow rate, the heat rejection is a nearly linear function of the temperature difference between the incoming fluid and the air temperature. For example, a Technical Systems FC Series model FC-48-597A operating at a fluid flow rate of 80 gpm with a 40% glycol solution has a heat rejection rating of 20.2 kBTU/hr/ΔT (Technical Systems, 2016). If the incoming fluid temperature is 90 ^{o}F and the air temperature is 70 ^{o}F, the heat rejection by the cooler is 404 kBTU/hr or about 34 tons.

Figure 1.8. 96-ton dry fluid cooler installed at Building 584.

Dry fluid coolers can be ordered with variable frequency drive (VFD) motors that allow for lower fan speed operation when 100% capacity is not needed. Use of VFD fans can result in major improvements in electrical efficiency due to the nature of fan power. Heat rejection from a dry fluid cooler is approximately a linear function of fan speed. Fan power, however, varies with the cube of the fan speed. Therefore, if a fan is operated at 50% of maximum speed, the cooler can reject about half the heat that it would at 100% fan speed while using only one-eighth of the power. In other words, the cooler has a heat rejection efficiency (kBTU/hr/kW) that is about four times larger, with a fan speed of 50% compared to a fan speed of 100%. (Actual efficiency may be lower depending on fluid pumping costs.)

Table 1.4 shows the calculated energy efficiency ratio (EER, kBTU/hr/kW) of a Technical Systems FC Series model FC-48-597A operating at a fluid flow rate of 80 gpm over a range of fan speeds and temperature differences. The efficiency includes pumping costs associated with the fluid pressure drop (10 ft. of heat) through the cooler. The fluid pumping power becomes a significant part of the cooler power consumption at fan speeds below 30% in this case.

Table 1.4. Calculated heat rejection energy efficiency ratio (kBTU/hr/kW) for a 48-ton dry fluid cooler at various fan speeds.

			Fan speed						
delta T (deg F)	100	90	80	70	60	50	40	30	20
5	32	39	48	60	75	95	116	128	112
10	64	78	96	119	151	190	232	256	225
15	96	116	143	179	226	285	348	384	337
20	128	155	191	239	302	380	464	512	449
25	160	194	239	298	377	476	581	639	561
30	192	233	287	358	452	571	697	767	674
35	224	272	334	418	528	666	813	895	786
40	256	310	382	477	603	761	929	1023	898

The dry fluid cooler control system can be designed to maximize the efficiency by using a fan speed ramp that is proportional to the temperature drop. For example, using the cooler described above, the fan speed could be set to vary linearly from a minimum of 30% with a temperature difference of five degrees up to a maximum of 80% with a temperature difference of 40 degrees. That control scheme would result in an energy efficiency ratio greater than 200 for temperature differences larger than 10 degrees (Figure 1.9).

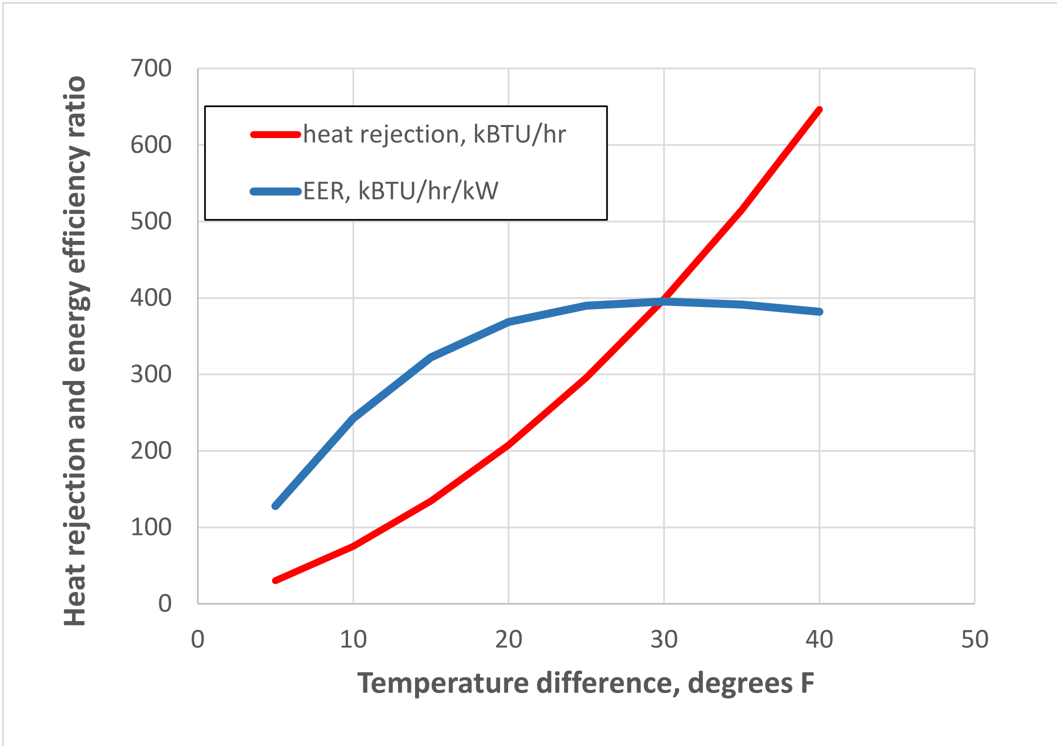

Figure 1.9. Heat rejection and energy efficiency ratio for a 48-ton dry fluid cooler with fan speed controlled by the temperature difference.

2.0 EXAMPLE APPLICATION OF TECHNOLOGY

The following describes an example application of the technology, as part of the ESTCP ER-201013 demonstration. This demonstration involved retrofitting three building geothermal heat pump systems at the MCAS Beaufort with dry fluid coolers to reduce ground loop temperatures. The case study presented here focuses on the Military Police building, Building 584. The other building systems are similar and are described in the Final Report.

2.1 Geothermal Heat Pump System

Building 584 is a one-story, 12,550-sq. ft. building that includes the original main building and an annex. The main part of the building (9,700 sq. ft.) was constructed in 1959 with concrete block and insulating concrete with rigid foam. An annex (2,850 sq. ft.) was added in the early 2000's. The building is at least partially occupied at all times and it has a large number of interior rooms.

Prior to installation of the geothermal heat pump system, the building was cooled using a 39-ton air-cooled chiller with ice storage capacity. An analysis by Trane found that that system had a cooling energy efficiency ratio of only 4.8 kBTU/kW. The building was heated using high-temperature hot water from the central heat plant at the base.

The Building 584 HVAC system was converted to geothermal heat pumps in 2004 using twenty-four 300-foot-deep boreholes that are located beneath the parking lot in a U shape (Figure 2.1). The average horizontal spacing between these wells is about 20 ft., but the two long rows of boreholes are separated by about 60 ft.

Water exiting the building is split into three parallel streams that each feed the inlet side of 8 borehole heat exchanger U-tubes connected in parallel. The loop water exiting the borehole U-tubes returns in three parallel lines that connect to a single line that is routed back into the building and through the heat pumps inside the building. The ground loop flow is driven by a redundant pump system using a VFD. The flow rate is allowed to drop when the HVAC demand is low, and it increases to a maximum of about 80 gpm when the building loads are high. This variable pumping rate is used to reduce pump energy costs. The initial geothermal heat pump system used a single 35-ton Florida Heat Pump water-to-water heat pump. That single heat pump was replaced in 2009 with 18 smaller Trane GEH series water source heat pumps that are distributed throughout the building.

Continuous monitoring of the Building 584 ground loop temperature entering and leaving the building began in August 2012, with readings taken every 15 minutes. These data were stored and accessed using the base-wide web control system that is in place at the MCAS Beaufort. This system allows for conditions at any of the buildings to be monitored remotely, and selected data can be archived on the web control server.

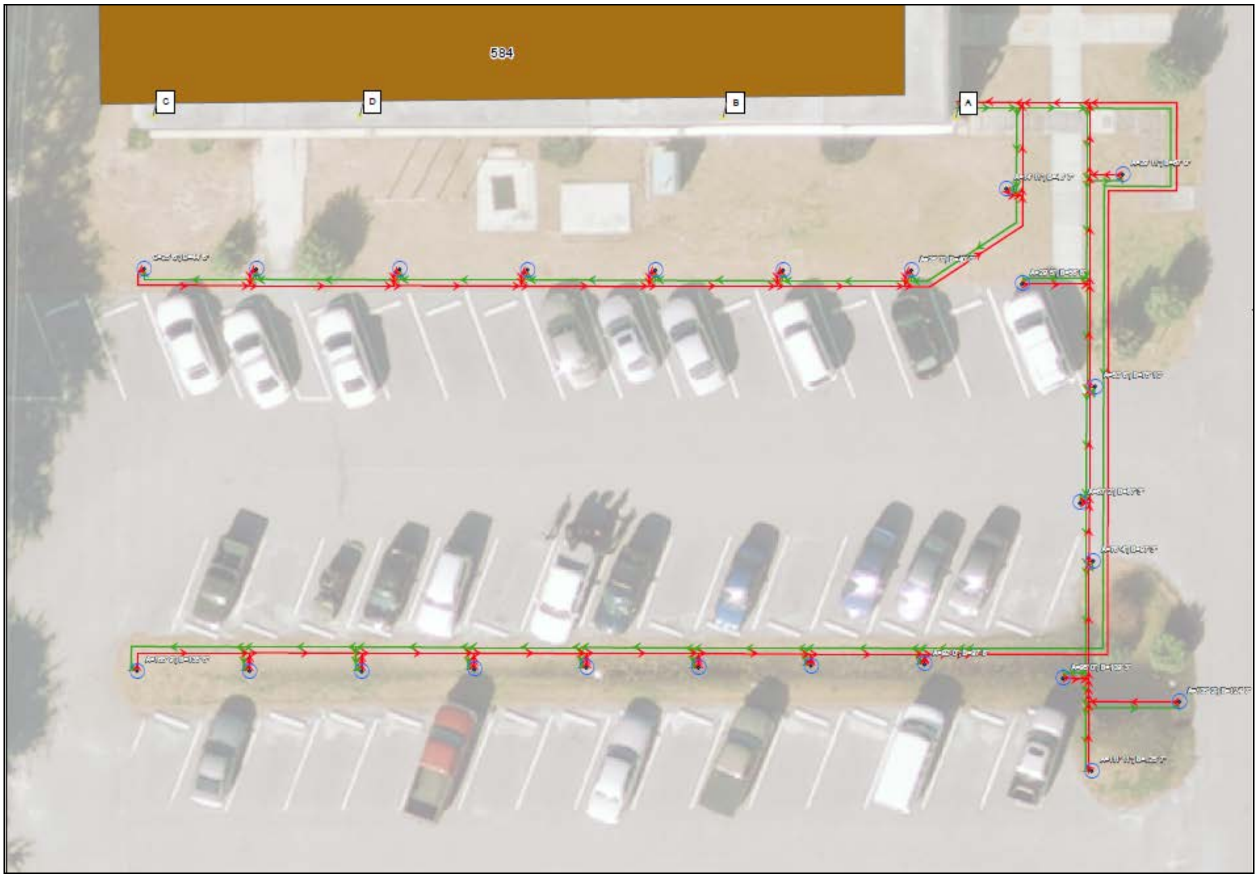

Figure 2.1. Borehole heat exchanger locations for the Building 584 geothermal heat pump ground loop.

As was discussed earlier, Building 584 is extremely cooling dominated (see Table 1.1), and about 99.5% of the total HVAC load is for cooling the building. This has resulted in excessive ground loop temperatures in the summertime (Figure 1.6), with warm temperatures persisting through the year into the winter (Figure 1.5). By the summer of 2012, the ground loop temperature entering the building was consistently exceeding 95 ^{o}F, and the loop temperature leaving the building was exceeding 105 ^{o}F.

2.2 Dry Fluid Cooler Installation and Operation

The geothermal heat pump system at Building 584 was modified in late October 2013 to include a 96-ton Technical Systems FC-96-1195 dry fluid cooler (Figure 1.8). This eight-fan cooler is the largest size built by the manufacturer, and it consists of two rows of 4 VFD fans with separate controls for each row. The cooler was installed adjacent to the building, close to the location where the ground loop enters and exits the building (Figure 2.1). A small wooden shadow fence was installed around part of the cooler for aesthetic reasons. Additional system monitoring was added at this time, including the local outside air temperature, ground loop flow rate, temperatures entering and exiting the dry fluid cooler, and the dry fluid cooler fan speeds for each row of fans. All of these data are accessed and stored using the base web control system. Continuous data have been collected from this system since December 2013.

Ground loop water exiting the building enters the dry fluid cooler before continuing on to the borehole heat exchangers. The fan speeds for the two stages of fans are controlled based on the temperature difference between the outside air and the entering loop temperature. The fans begin operating when the temperature difference exceeds 5 °F, and they ramp linearly to 100% fan speed when the temperature difference reaches 20 °F. This program allows for year-round operation, but fan speeds are higher in the wintertime when the difference between the air temperature and the water temperature entering the cooler is larger.

Figure 2.2 shows typical wintertime operation of the dry fluid cooler – geothermal heat pump system. This graph shows 5 days of operation in December 2014, with the cooler fan speed (black line), the ground loop water temperature entering the building (blue line), the ground loop water temperature leaving the building (purple line), the water temperature leaving the dry fluid cooler (dark red line), and the outside air temperature. The cooler fan speed is controlled by the difference in the outside air temperature (green line) and the loop temperature leaving the building (purple line).

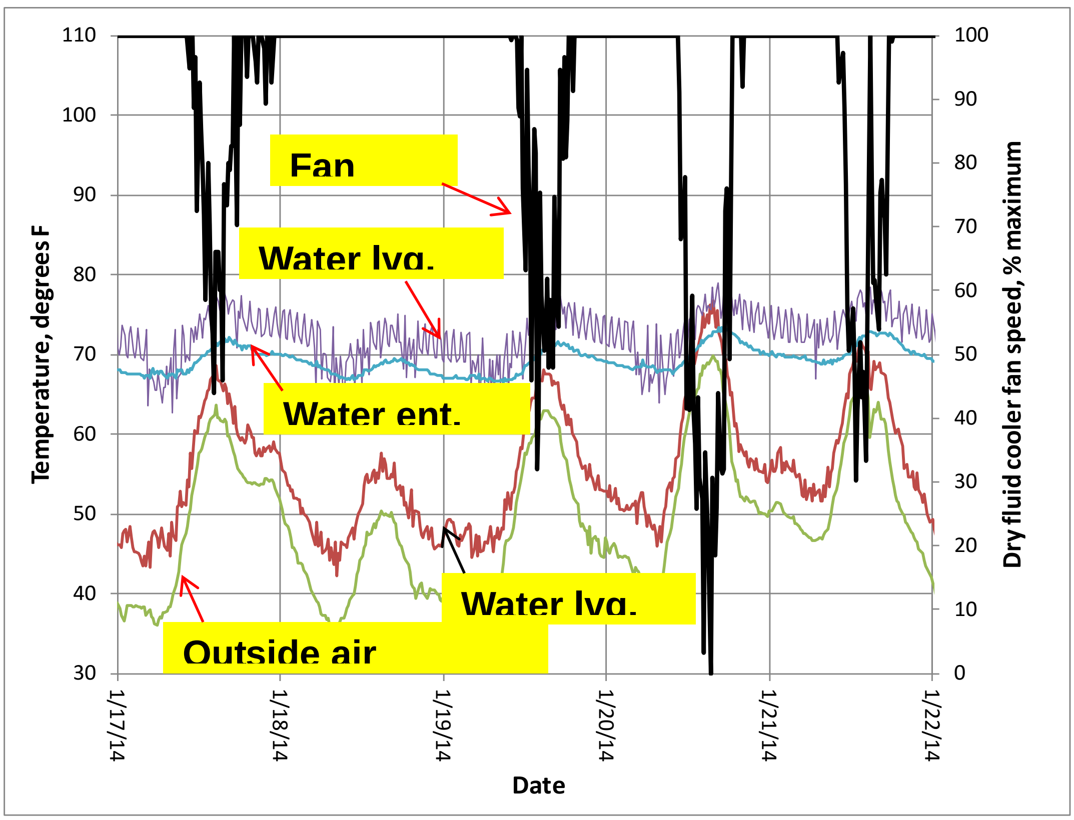

Figure 2.2. Typical wintertime operation of the dry fluid cooler and geothermal heat pump system at Building 584.

At night, when the air temperature is cold, the loop temperature leaving the building remains relatively warm and the fans run at 100%. This results in a substantial amount of heat rejection by the fluid cooler, and the temperature leaving the cooler is low (dark red line), often as cool as

45 ^{o}F. That cold water flows through the borehole heat exchangers, extracting heat from the ground before re-entering the building (blue line).

On days with warm afternoons, the outside air temperature started to approach the loop temperature entering the dry fluid cooler and the fan speed ramped down. As this happens, the rate of heat rejection from the cooler decreases and the temperature leaving the cooler (dark red line) is closer to the temperature leaving the building.

The summertime operation of the cooler is similar, but the outside air temperature tends to be closer to (or above) the loop temperature leaving the building, so the fans are often not running or are running at low speeds. They only very rarely reach 100% fan speed in the summer with the control program.

2.3 Ground Loop Temperatures after Adding Dry Fluid Cooler

The dry fluid cooler was installed in late October 2013, and data collection began in early December 2013. The cooler has been in continuous operation since then. Data collection has also been continuous, but data from the cooler was lost on three occasions due to malfunctions of the data collection or web control server. As of June 2016, there were 18 months of high quality data collected from the system.

The dry fluid cooler had a rapid effect on the ground loop temperature. The loop temperatures in January 2014 are shown in Figure 2.3. Compared to the previous January, the ground loop temperature was about 13 degrees lower, with an average loop entering temperature of 69 ^{o}F. During the first summer of operation, the ground loop temperatures were reduced by about 8 ^{o}F compared to the previous summer. The summer of 2014 was unusually warm, with record or near-record temperatures. During the latter half of August, the outside air temperature recorded at the dry fluid cooler exceed 100 ^{o}F on five days (Figure 2.4). The average ground loop entering temperature during this period was 87.1 ^{o}F, compared to 94.8 ^{o}F the previous August.

After the first year of operation, the ground loop temperatures appeared to stabilize with some additional cooling. In the second half of August 2015 (Figure 2.5), the average ground loop supply temperature was 84.4 ^{o}F, which is more than 10 degrees cooler than the late August loop temperature prior to adding the cooler (see Figure 1.6).

The average summer loop temperature over the past several years was computed by averaging the 15-minute readings for the months of June, July, August, and September. In 2013, before adding the dry fluid cooler, the entering loop temperature averaged 94.1 ^{o}F. In 2014, after one season of cooler operation, the entering loop temperature averaged 85.6 ^{o}F, and in 2015 it averaged 84.8 ^{o}F.

As will be discussed in the next section, it should be possible to reduce the ground loop temperatures further with some adjustment to the ground loop pumping system and dry fluid cooler control.

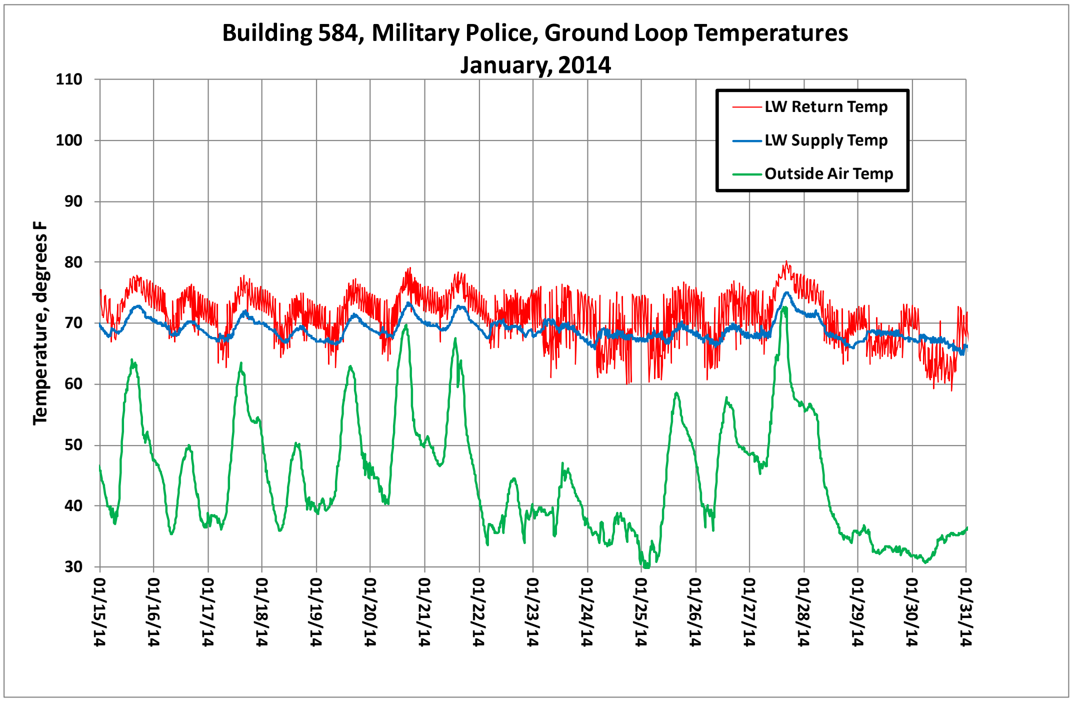

Figure 2.3. Ground loop temperatures at Building 584 in January 2014.

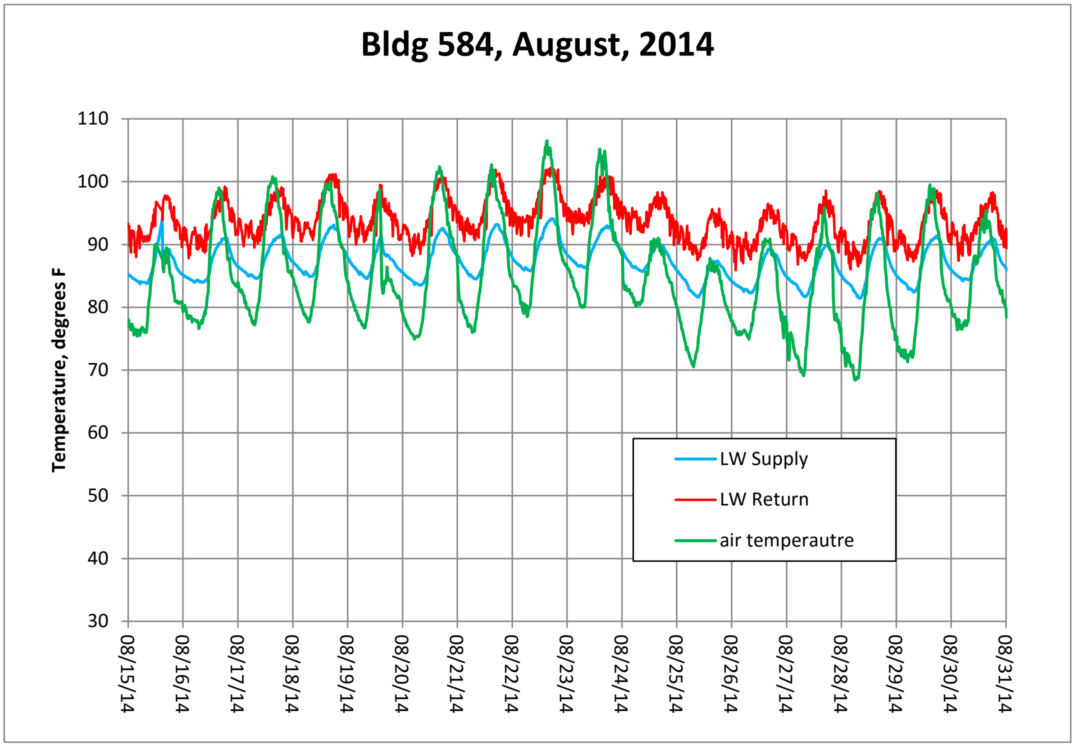

Figure 2.4. Ground loop temperatures at Building 584 in August, 2014.

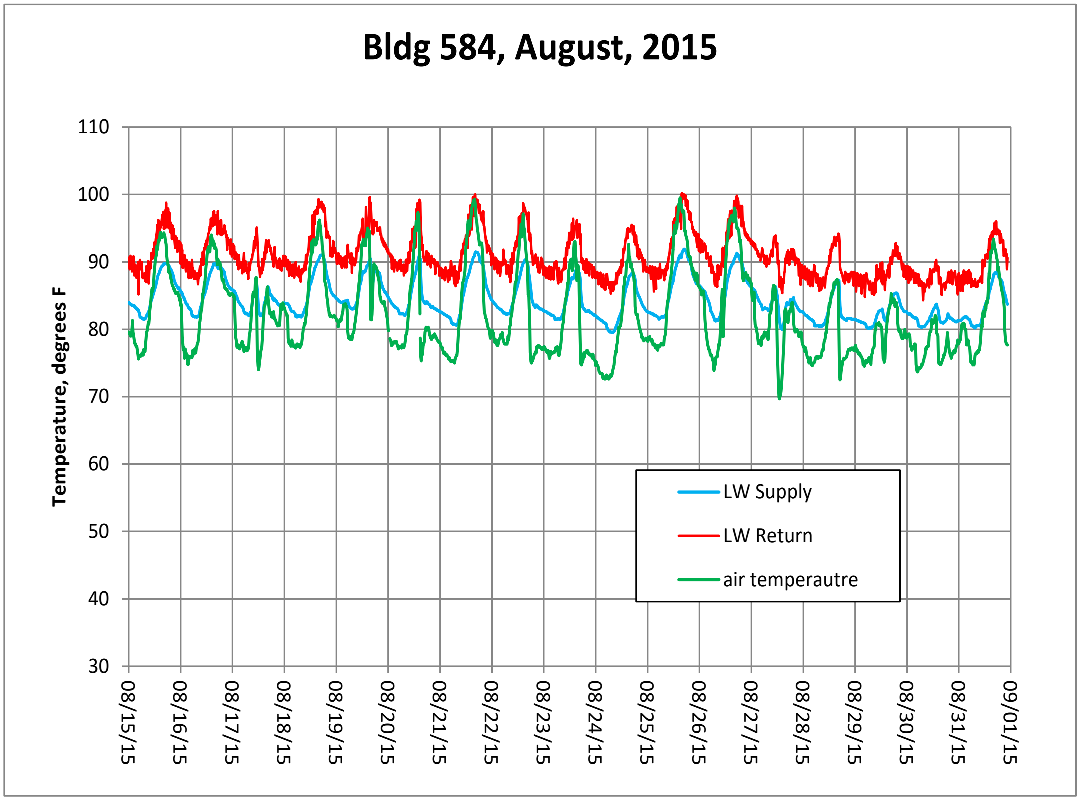

Figure 2.5. Ground loop temperatures at Building 584 in August, 2015.

2.4 Analysis of Dry Fluid Cooler Heat Rejection

The dry fluid cooler heat rejection was calculated for each 15-minute measurement interval using the loop flow rate and the temperature difference between the water entering and leaving the cooler. These data were integrated to get monthly heat rejection values over the 3-year measuring period (Table 2.1). These results show that the dry fluid cooler heat rejection was relatively uniform during the year. (The result for November is probably not reliable due to problems with the data collection system during that month.)

Over the course of a year, the dry fluid cooler has been rejecting about 1.09 million kBTUs from the ground loop. The amount of heat rejected from the building into the ground loop includes the normal building cooling load and the waste heat from the heat pumps, minus the building heating load. These data were shown previously in Table 1.1 for Building 584. Over a year, about 886,000 kBTUs were rejected into the ground loop. Prior to the installation of the dry fluid cooler, this load imbalance had resulted in dramatic heating of the borehole heat exchangers. However, this trend has been reversed following installation of the dry fluid cooler, because now there is a net heat removal from the ground loop of about 204,000 kBTUs per year. This heat removal has allowed the ground loop to cool by about 10 degrees over the last couple of years. In other words, the ground loop/borehole heat exchanger system at Building 584 is currently being rehabilitated following years of excessive heat rejection.

Table 2.1. Average monthly heat rejection by the Building 584 dry fluid cooler, kBTU.

Month	heat rejection
January	97058.62
February	100276.00
March	79951.79
April	80517.20
May	75150.23
June	82211.34
July	92744.25
August	98950.29
September	103201.42
October	103448.14
November	75978.03
December	98823.06

It would be expected that the rate of heat rejection by the cooler should have been substantially larger in the wintertime due to the larger average temperature difference between the entering water temperature and the outside air temperature. This was not the case, however, due to the variable flow rate ground loop water pumps. When operated at 100%, the loop pumping rate is about 80 gpm, but this rate was reduced to levels as low as 10 gpm during periods when the building load was small. These low-load periods mainly occur in the wintertime, and the resulting low flowrate greatly reduces the efficiency of the dry fluid cooler.

During the initial operation of the dry fluid cooler, before the control system was fully implemented, there were periods where the fan ran at 100% regardless of the water/air temperature difference. These data were normalized by computing the heat rejection rate (from the loop flowrate and the temperature drop entering and exiting the cooler) and dividing it by the temperature difference between the incoming water and the outside air. This ratio, in units of kBTU/hr/ΔT, gives the relative cooler performance. The observed dry fluid cooler performance is plotted as a function of the loop flowrate in Figure 2.6. The cooler performance is essentially a linear function of the loop flowrate over the range of observed values.

At low loop flowrates, the rate of heat rejection per degree of temperature difference is very low, only a few kBTU/hr/ΔT. This rate increases to a maximum of about 22 kBTU/hr/ΔT at the maximum flowrate of about 80 gpm. Manufacturers data for the FC-96 cooler are consistent with our measurements and show approximately linear performance up to a flow rate of about 120 gpm. From 120 gpm to about 250 gpm, the cooler performance becomes only a weak function of temperature, with heat rejection rates varying from about 31 to 38 kBTU/hr/ΔT.

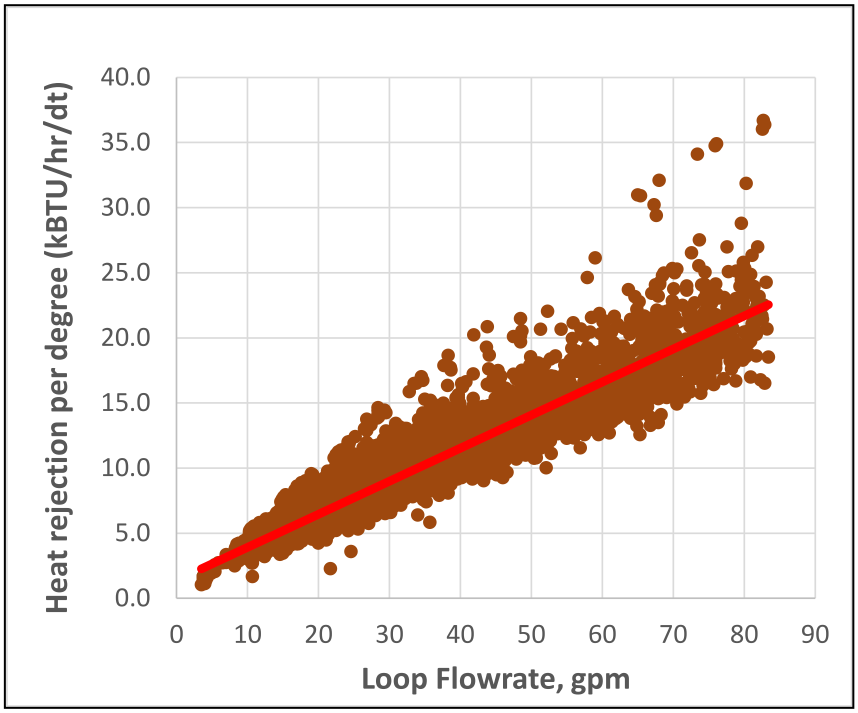

Figure 2.6. Observed dry fluid cooler normalized heat rejection (kBTU/hr/ΔT) as a function of the ground loop flowrate with cooler fans operating at 100%.

While the VFD controlled loop water pump at Building 584 reduces, water pumping costs, it has a negative effect on the performance of the dry fluid cooler, which would operate much more efficiently at a constant flowrate of 80 gpm or more. It is also apparent that similar heat rejection rates could be achieved using a smaller dry fluid cooler (such as a four-fan FC-48) that is designed for lower flowrates than the FC-96. From the manufacturer's data, an FC-48-597A operating at 80 gpm and 100% fan speed has a heat rejection rating of about 20.4 kBTU/hr/ΔT, which is similar to the performance observed with the larger fluid cooler (Figure 2.6).

The electrical energy used by the dry fluid cooler was calculated using the fan speed data that were collected every 15 minutes. The dry fluid cooler at Building 584 has 8 one-horsepower electric fans. When these fans operate at 100% fan speed, the electrical power is 6 kW. As the fan speed is reduced, the power drops by the cube of the fan speed, so at 50% fan speed, the electrical power use is only 0.75 kW. The fan's electrical energy use was integrated over time to get monthly values. An additional power cost for using the dry fluid cooler is due to pumping costs to overcome the water pressure drop in the cooler. At the low flowrates used in this application these pumping costs are very small, but they become significant as the flowrate increases relative to the size of the cooler.

The average monthly dry fluid cooler heat rejection, electrical energy use, energy efficiency ratio, loop flowrate, fan speed, and temperature difference between the entering water and outside air are shown in Table 2.2. (As in Table 2.1, the November results for heat rejection and EER are probably not reliable.)

Table 2.2. Monthly average dry fluid cooler data from Building 584.

	heat reject.	electricity	EER	flowrate	fan speed	delta T
Month	kBTU	kWhr	kBTU/hr/kW	gpm	% max	degree F
January	97058.62	3053.74	31.78	17.49	74.52	26.19
February	100276.00	2473.01	40.55	32.78	70.56	18.53
March	79951.79	1460.04	54.76	43.54	49.53	12.88
April	80517.20	1626.77	49.50	40.91	53.58	12.81
May	75150.23	1945.50	38.63	49.10	56.46	10.43
June	82211.34	555.52	147.99	52.21	32.01	9.52
July	92744.25	601.59	154.16	53.34	32.81	9.96
August	98950.29	618.03	160.11	51.88	35.17	10.26
September	103201.42	1033.49	99.86	50.79	48.90	12.32
October	103448.14	2028.20	51.00	42.10	63.64	15.33
November	75978.03	3231.51	23.51	25.60	83.53	23.28
December	98823.06	2820.87	35.03	26.90	81.23	21.58

The thermal energy efficiency ratio (EER) shown in the fourth column is calculated by dividing the heat rejection by the electrical energy use. Surprisingly, the summer months of June, July, and August have consistently shown the most efficient heat rejection from the dry fluid cooler. This counter intuitive result is due to two factors: the low loop flowrates in the winter and the higher fan speeds in the winter. The average loop flowrates in the winter months (column 5) were only about half of the average rate in the summer. As was discussed earlier, the low flowrates severely reduce the amount of heat rejection.

The higher average winter fan speeds result from the fan control schedule, which increases the fan speeds as the water/air temperature difference increases over the range of 5 to 20 degrees. Because the temperature difference was greater in the wintertime, the fan speeds were higher then, increasing the electricity consumption. The higher fan speeds would normally have increased heat rejection, but the low loop flowrates prevented this.

The yearly average heat rejection EER of the dry fluid cooler was 50.7 kBTU/hr/kW. Using the same equipment, this performance could likely be improved to an EER of between 200 and 300 kBTU/hr/kW by operating the ground loop at a constant flowrate of 80 gpm and by adjusting the dry fluid cooler fan control so that the maximum fan speed is reached when the temperature difference is 30 degrees or more. These adjustments would be expected to produce efficiency results similar to those shown in Table 1.4 and Figure 1.9.

3.0 ANALYSIS OF HEATING/COOLING LOADS AND DRY COOLER SIZING

This section uses a commercial geothermal heat pump simulation model (GLHEPro) along with spreadsheet models to simulate the example site consisting of the Building 584 system, and to explore alternative strategies for controlling the ground loop temperature. These lead to a system optimization and economic analysis for the combined geothermal heat pump/dry fluid cooler combination using wintertime heating.

3.1 Simulation of Ground Loop System Without Supplemental Cooling

The GLHEPro ground source heat pump simulation program was used to simulate the building ground loop performance prior to the addition of the dry fluid cooler. GLHEPro (IGSHPA, 2016) is a commercial program used for the design and analysis of geothermal heat pump ground loops. The code uses semi-analytical and numerical solutions for three-dimensional heat conduction from various geometry ground loop configurations. For the analysis described here, only vertical closed-loop boreholes with single U-tubes were considered, but the model can simulate other vertical borehole designs, as well as horizontal trenches.

The basic inputs to GLHEPro are the monthly building heating and cooling loads on the heat pumps, the local ground temperature, the borehole heat exchanger system geometry (number, location, and depth), construction details of the heat exchangers, subsurface thermal properties, loop flowrate, and the heat pump properties. The model output consists primarily of the predicted monthly ground loop entering and exiting temperatures and the electrical energy consumed by the heat pumps. The simulation period is controlled by the user and may extend for decades.

Normally the building heating and cooling loads would be simulated using a building simulation program such as eQUEST. For the simulation of Building 584, the building loads are known from the ground loop data (Table 1.1). The ground loop load data shown in Table 1.1 include the waste heat from the heat pumps. Because GLHEPro uses the building load on the heat pumps, it was necessary to remove the waste heat component from the cooling loads in Table 1.1. The heat pump waste heat was estimated from heat pump performance data included in the GLHEPro heat pump library. The building cooling loads in Table 1.1 were multiplied by 0.8 to correct for the waste heat. The simulation was started in January 2005 and run for a period of 30 years. The simulated ground loop temperature entering the building is compared to the observed monthly temperatures in 2012 and 2013 prior to the installation of the dry fluid cooler in Figure 3.1. The GLHEPro simulation tool appears to do a very good job of reproducing the observed temperatures after 8-9 years of operation, showing the same level of upward yearly temperature drift in the ground loop.

This comparison is shown again in Figure 3.2, with the full 30-year GLHEPro prediction. The model predicts that in the absence of supplemental cooling or other adjustments, the loop temperature will continue to increase over time, with average monthly inlet temperatures exceeding 100 ^{o}F. The simulation shows that the ground loop heating occurs rapidly, and substantial heating of the loop (~11 degrees) occurs in the first year. After 10 years, the loop yearly average temperature is about 21 degrees warmer than background, and by 30 years, it is 27 degrees warmer.

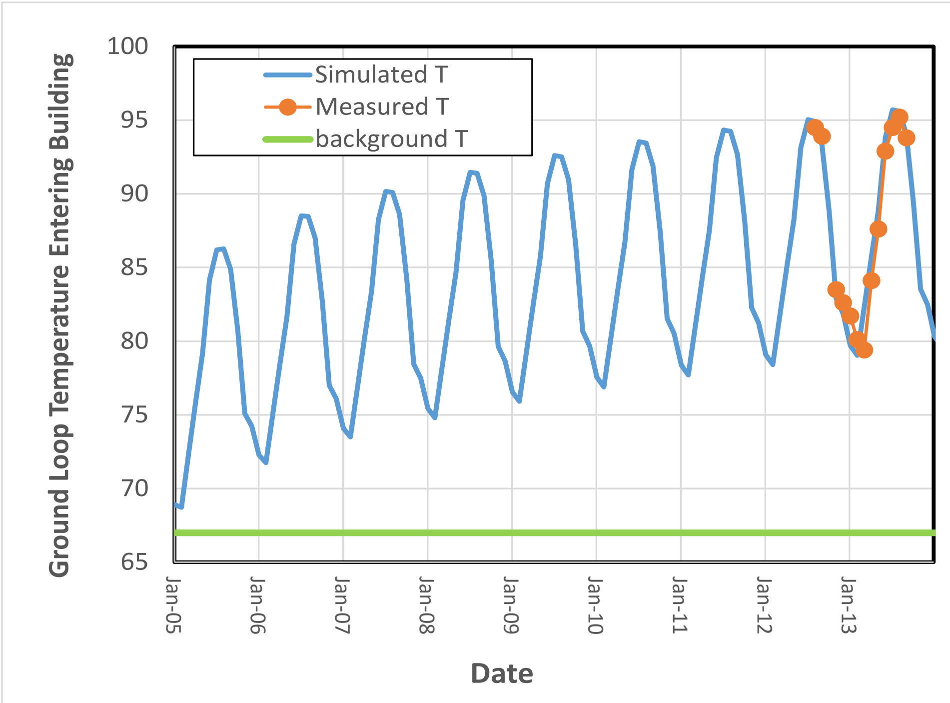

Figure 3.1. Results of GLHEPro simulation of Building 584 ground loop prior to the installation of the dry fluid cooler.

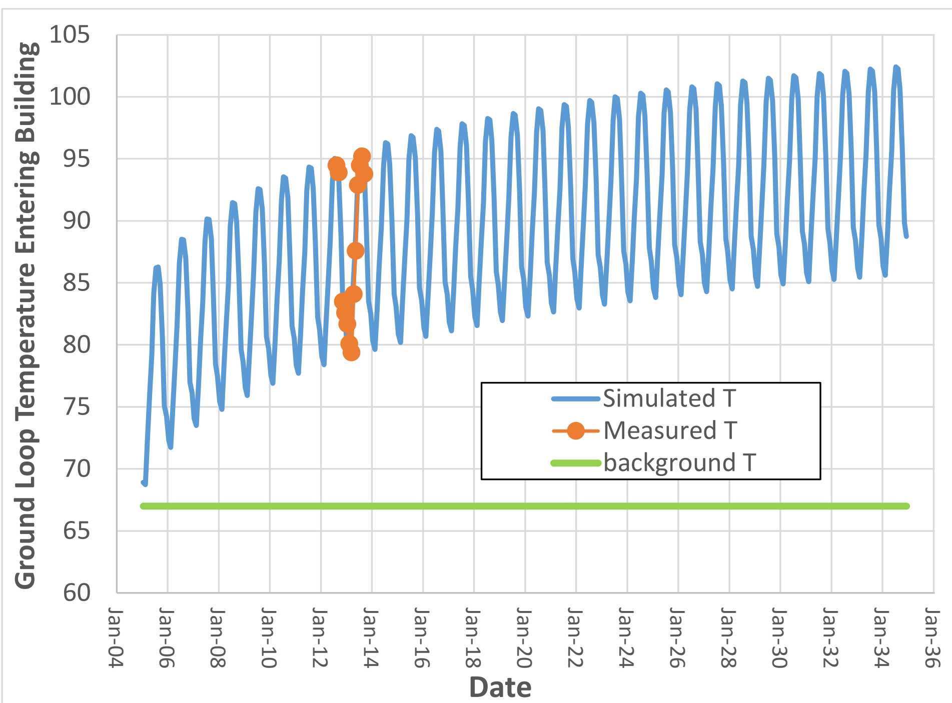

Figure 3.2. GLHEPro simulation showing predicted temperatures for 30 years of operation.

Figure 3.3 shows the simulation results along with ground loop temperature data collected after the installation of the dry fluid cooler (which is not accounted for in this simulation). It is apparent that use of the cooler has stopped the trend of increasing ground loop temperature.

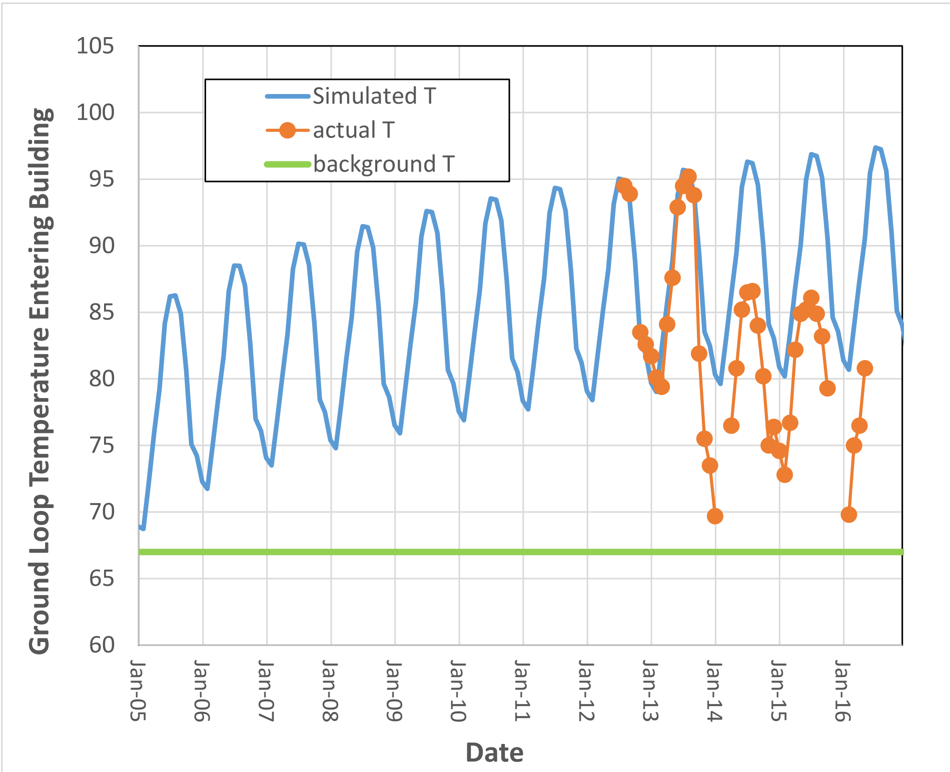

Figure 3.3. Simulated and observed ground loop temperatures. The dry fluid cooler began operation in November 2013.

3.2 Calculating the Ground Loop Size Necessary to Stabilize Loop Temperature

A series of GLHEPro simulations were performed to determine the number of borehole heat exchangers that would have been needed to avoid an excessive temperature build up in the ground loop. These simulations assumed that the larger well field was installed back in 2004, at the time of the initial geothermal heat pump installation.

The borehole depth (300 ft.), construction characteristics, and spacing (20 ft.) were held constant, while the number of boreholes was increased. It was found that as the size of the ground loop was increased, the magnitude of the yearly temperature fluctuations decreased, but the overall average temperature still drifted upward. The ground loop temperature is predicted to increase with time, even when the number of borehole heat exchangers is tripled. Figure 3.4 shows the GLHEPro simulation results using a borehole array with three rows of 25 boreholes each (75 total). This configuration gives a maximum monthly average summer temperature of 81.4 ^{o}F at year 30 of the simulation, with a yearly average temperature of 79.6 ^{o}F. That still represents a 12.6-degree temperature increase over the natural background temperature. With a typical cost for installing borehole heat exchangers of about $17/ft., these additional boreholes would add about $260,000 to the system capital cost.

This large capital expenditure would be difficult to justify for a building of this size, and it would likely be difficult to find the room to install the additional boreholes. For these reasons, enlarging the ground loop does not seem like it would be a practical solution for dealing with highly unbalanced systems.

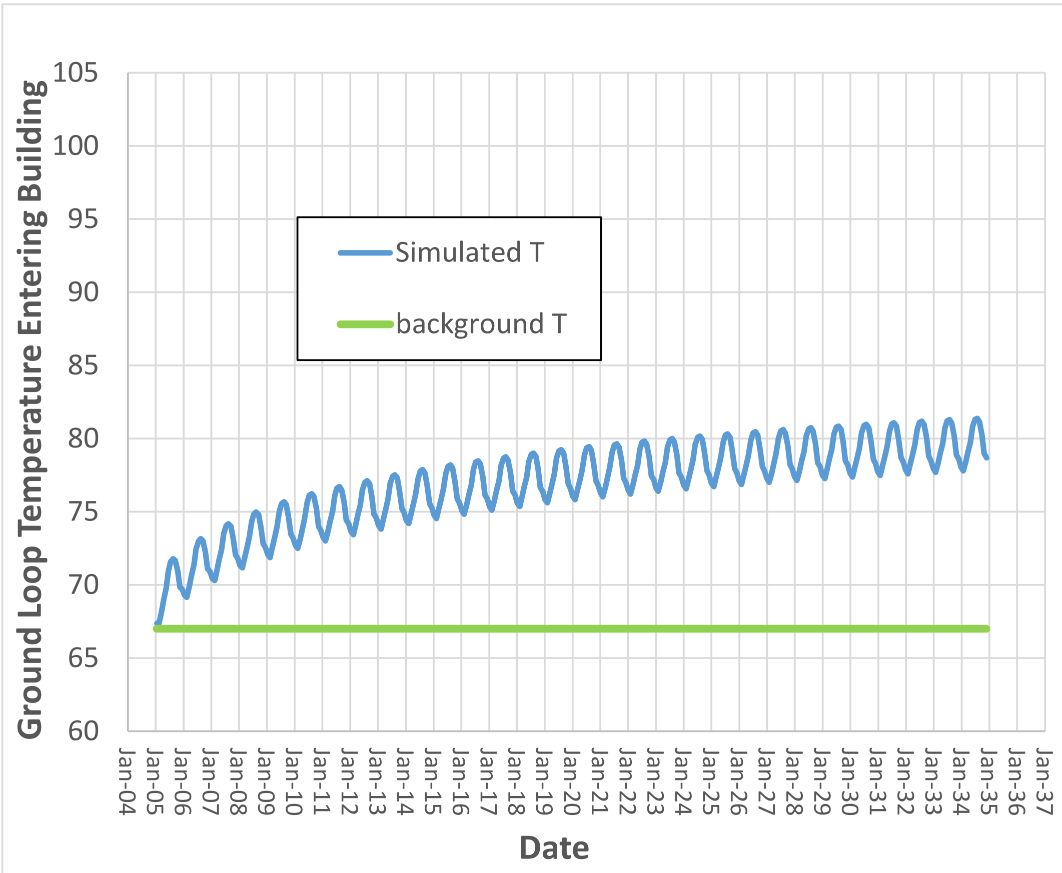

Figure 3.4. GLHEPro simulation of Building 584 using seventy-five 300-foot-deep borehole heat exchangers.

3.3 Optimal Sizing and Operation of Dry Fluid Cooler with Existing Ground Loop

A simulation is performed to identify a design and operation scheme that can economically stabilize the ground loop temperatures over time using the existing array of 24 borehole heat exchangers. The simulation assumes that the dry fluid cooler was added at the same time that the system began operation, at the start of 2005.

The dry fluid cooler heat rejection is included in the GLHEPro simulation by adding in an equivalent monthly heating load to the ground loop. The removal of heat from the ground loop by the cooler drops the temperature in the loop, which has the effect of reducing the efficiency of the fluid cooler due to the lower temperature difference. This negative feedback between the ground loop temperature and the dry fluid cooler efficiency is not accounted for in GLHEPro, which uses constant monthly values for the building loads and dry fluid cooler heat rejection. Therefore, it was necessary to iterate between a separate model of the dry fluid cooler performance and GLHEPro until the loop temperature stabilized for a particular design.

The monthly dry fluid cooler performance was calculated using a spreadsheet-based model that contains measured hourly temperatures from the MCAS Beaufort over a one-year period in 2010 to 2011. Those temperature data were compiled by month, and then sorted into 5-degree temperature "bins". This allows for calculation of the number of hours in each temperature range during each month (Table 3.1).

Table 3.1. Sorted hourly temperatures 2010-2011 at the MCAS Beaufort, SC.

temperature bins													
T range	ave T	Jan hours	Feb hours	Mar hours	Apr hours	May hours	June hour	July hours	Aug hours	Sept hour	Oct hours	Nov hours	Dec hours
21-25	23	12	0	0	0	0	0	0	0	0	0	0	12
26-30	28	44	0	0	0	0	0	0	0	0	0	0	60
31-35	33	83	8	1	0	0	0	0	0	0	0	0	120
36-40	38	117	49	7	0	0	0	0	0	0	0	38	134
41-45	43	101	82	32	0	0	0	0	0	0	0	69	113
46-50	48	160	119	96	12	0	0	0	0	0	9	89	96
51-55	53	129	116	85	72	1	0	0	0	0	46	79	110
56-60	58	58	92	160	93	7	0	0	0	0	93	131	52
61-65	63	33	95	154	135	26	0	2	0	13	143	109	29
66-70	68	11	49	103	173	119	8	17	5	128	121	134	13
71-75	73	5	36	55	122	239	66	53	26	188	140	53	0
76-80	78	0	23	25	91	173	227	212	251	151	122	17	0
81-85	83	0	2	20	18	132	184	203	251	111	69	0	0
86-90	88	0	0	5	1	42	139	143	150	113	1	0	0
91-95	93	0	0	0	0	4	76	75	92	16	0	0	0
96-100	98	0	0	0	0	0	17	20	2	0	0	0	0
101-105	103	0	0	0	0	0	1	2	0	0	0	0	0

Given the ground loop temperature exiting the building, the dry fluid cooler performance rating (kBTU/hr/ΔT), and the fan speed, it is possible to calculate the heat rejection and energy consumption for each of these temperature ranges. The monthly total heat rejection and energy cost is then found using the number of hours associated with each temperature range for the month. At each temperature range, the fan speed is calculated based on the temperature difference with the loop water, using a linear variation over a specified temperature range (as in Table 1.4).

The simulation process started with an initial estimate of dry fluid cooler performance, using an assumed yearly ground loop temperature profile. The cooler fan speed was set so that the fan speed starts at 40% when the temperature difference is 5 degrees, and then increases to 100% when the difference reaches 25 degrees. The dry fluid cooler selected for this design is a Technical Systems FC-48-597a, operated with a constant flowrate of 80 gpm. This four-fan unit is exactly one-half the size of the installed cooler at Building 584, but it has a similar heat rejection rating at the design flowrate of 80 gpm. At 100% fan speed, this unit is rated at 20.4 kBTU/hr/ΔT.

The calculated monthly heat rejection results from the spreadsheet model are used as inputs in the initial GLHEPro simulation. The predicted GLHEPro monthly loop temperatures resulting from the building loads and the dry fluid cooler operation were then used to refine the cooler spreadsheet model calculations. By iterating between the spreadsheet cooler performance model and GLHEPro, a workable design was obtained where the ground loop temperature is nearly stable over time. The monthly dry fluid cooler heat rejection, power consumption (including loop pumping costs), and ground loop temperature are shown in Table 3.2

Table 3.2. Simulated dry fluid cooler performance.

Month	**Heat Rejection, kBTU**	**Energy Use, kW-hr**	**Ground Loop Temp.**
January	101490	636	53
February	63848	431	59
March	58767	403	64
April	40889	309	69
May	19036	202	77
June	19561	204	83
July	27138	239	85
August	21688	212	85
September	38796	292	81
October	50343	359	72
November	76582	511	63
December	124858	764	53
Total	**642994**	**4562**	**average = 70**

The yearly heat rejection by the cooler of 643,000 kBTU is about 72% of the cooling load delivered from the building and heat pumps to the loop. (See Table 1.1; it is about 90% of the building load.) This heat is removed using 4600 kW-hr of electricity, of which about one-quarter is used to pump the water through the cooler. The average energy efficiency ratio (EER) of this operation would be about 141 kBTU/hr/kW.

The simulated ground loop temperature entering the building is shown in Figure 3.5. The loop temperature has been nearly stabilized by the dry fluid cooler operation, using a cooler that is one-half the size of the one currently installed at Building 584. The yearly loop average temperatures now fall well within the range needed for efficient heat pump operation.

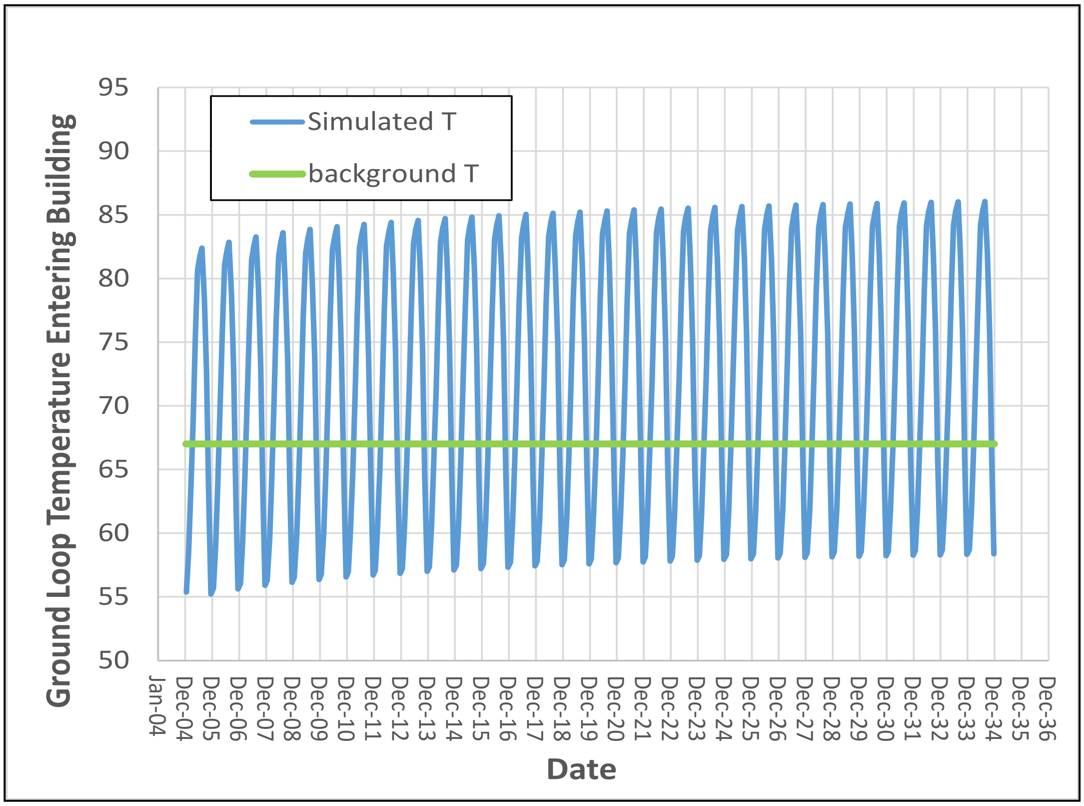

Figure 3.5. Simulated Building 584 ground loop temperatures with optimized dry fluid cooler heat rejection. The dry fluid cooler begins operation immediately after the system is installed.

The purchase cost of the 96-ton, eight-fan dry fluid cooler installed at Building 584 was about $29,000 delivered to the site. We would expect a smaller 48-ton, four-fan unit to be about half the cost, or about $15,000. A reasonable estimate for the installation cost would be $5,000 for a total cost of about $20,000. This is far lower than the capital cost for increasing the size of the wellfield, but the dry fluid cooler uses additional electrical energy to power the fans and pump.

3.4 Comparison of Heat Pump System with and without Dry Fluid Cooler

Based on the observed loop temperatures and the GLHEPro simulation results, it does not appear that continued operation of the heat pump system without supplemental cooling would be desirable. After only about eight years of operation, the late summer loop temperatures entering the building were in the mid- to upper 90s (Figure 1.6). The GLHEPro simulation of the heat pump system predicts that the ground loop temperature would have continued to increase over the next 20 years, with average loop temperatures entering the building well over 100 ^{o}F (Figure 3.2). The simulation also shows that peak loop temperatures on hot summer afternoons could reach almost 120 ^{o}F in the later years of the system operation. Loop temperatures that high would probably be beyond the safe operating range for the heat pumps and might result in temporary system shutdowns or damage to the system.

The heating of the ground that would occur with the conventional system, while not permanent, would take a long time to reverse and would likely discourage continued use of geothermal heat pumps at the site in the future.

The unbalanced geothermal system without supplemental cooling becomes significantly less efficient with time as the loop temperature increases. This is in contrast to a system in which much of the excess cooling load is rejected using a dry flood cooler. Table 3.3 compares the simulated monthly electrical energy use for the geothermal heat pump system with and without the addition of the dry fluid cooler. The base case referred to here is continued Building 584 operation without supplemental cooling, as described in Section 3.1, while dry fluid cooler case uses the optimized design described in Section 3.3, and it includes the electrical energy used by the cooler.

Table 3.3. Simulated Building 584 electrical energy use (kW-hr) for the geothermal heat pump system with and without a dry fluid cooler.

Month	**Year 10 Base Case**	**Year 10 Dry Fluid Cooler**	**Year 30 Base Case**	**Year 30 Dry Fluid Cooler**
January	1108	1499	1195	1509
February	951	1224	1019	1232
March	2201	2094	2404	2129
April	3610	3180	3943	3240
May	5203	4568	5679	4659
June	7446	6484	8119	6614
July	8522	7352	9286	7498
August	8221	7162	8954	7304
September	6910	5916	7524	6031
October	4651	4047	5065	4122
November	1865	1899	2029	1927
December	1625	1853	1768	1875
Total	**52311**	**47277**	**56984**	**48139**

Comparing the yearly totals, during the tenth year of operation, the system with the dry fluid cooler is about 10% more efficient than the base case. In the 30th year of operation, the system with the dry fluid cooler is about 16% more efficient than the base case.

The monthly electricity use during year 30 is plotted in Figure 3.6. The energy savings occur mainly during the summer months of June, July, August, and September, when building cooling loads are at their highest. Energy use in the cooler months of November, December, January, February, and March are similar for the two systems. The energy savings during these months that result from the cooler ground loop temperatures are offset by the cost to operate the dry fluid cooler.

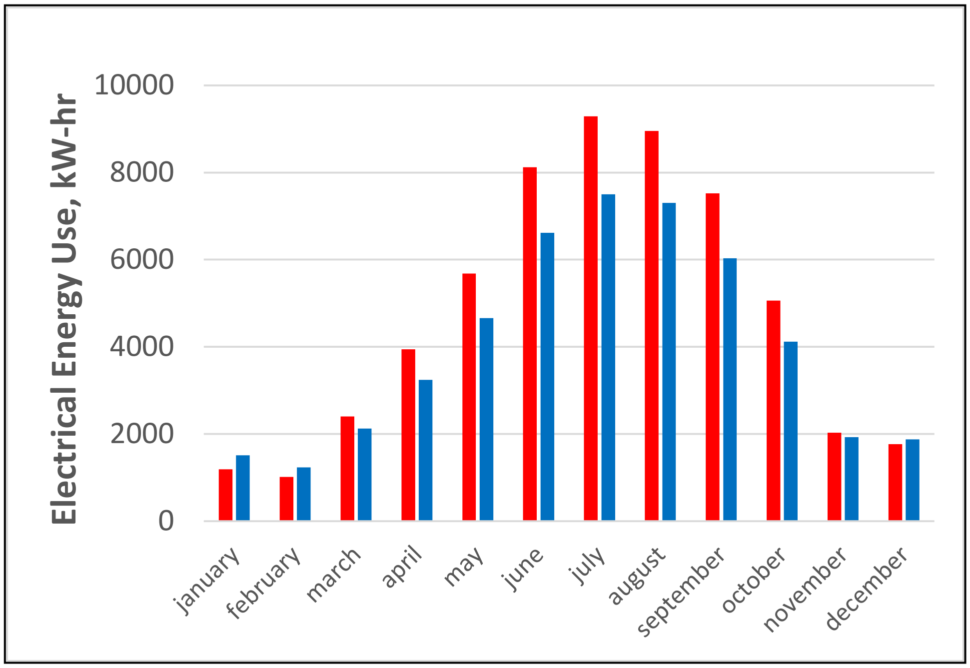

Figure 3.6. Simulated monthly energy use for Building 584 in year 30. The red bars represent the base case and the blue bars include the dry fluid cooler.

An economic comparison of these systems requires assumptions about the costs of electricity over time relative to the general inflation rate. One possible model could consider an initial non-peak energy cost ($/kW-hr), a peak energy cost ($/kW-hr) that would apply to summertime weekday afternoons and evenings, an annual energy cost inflation rate, and a general inflation rate. With this model, the energy costs would be escalated at the energy cost inflation rate. Over time, the calculated monthly energy costs can be converted to present dollars using the general inflation rate. Table 3.4 shows the projected energy costs assuming that one-half of the June, July, August, and September cooling costs occur during peak hours. The initial non-peak energy cost was $0.08/kW-hr; the initial peak energy cost was $0.15/kW-hr; the energy inflation rate was 5%; and the overall inflation rate was 2% in this example.

The energy costs (in current dollars) in this model increase substantially over time, due to the differential between the assumed energy cost inflation rate and the overall inflation rate, and due to the declining efficiency of the heat pumps. We acknowledge that if electrical energy prices remain low over the next 30 years, this projection would overestimate the future energy costs. However, it seems likely that electrical energy costs will increase substantially in the future, and these increases amplify the costs of heating and cooling system inefficiencies.

Table 3.4. Comparison of monthly electricity costs (in current dollars) after 10 and 30 years of system operation.

Month	**Year 10 Base Case**	**Year 10 Dry Fluid Cooler**	**Year 30 Base Case**	**Year 30 Dry Fluid Cooler**
January	$115	$156	$222	$281
February	$99	$128	$190	$230
March	$230	$219	$449	$398
April	$378	$333	$738	$607
May	$547	$480	$1,066	$874
June	$1,128	$982	$2,196	$1,789
July	$1,294	$1,116	$2,517	$2,033
August	$1,251	$1,090	$2,433	$1,985
September	$1,054	$903	$2,050	$1,643
October	$495	$431	$962	$783
November	$199	$202	$386	$367
December	$174	$198	$337	$358
Total	**$6,964**	**$6,238**	**$13,547**	**$11,346**

The cost savings from the system with the dry fluid cooler increase over time due to the fact that the system does not lose efficiency compared to the base system, and also due to the assumption that energy costs rise faster than the general rate of inflation. The cumulative energy costs for the two systems over a 30-year period is shown in Figure 3.7. In the early years, before the base case system ground loop has heated up, the two systems have similar efficiencies, and therefore, similar costs. However, as the base case system loses efficiency, and as the relative cost of electrical energy rises, the cumulative costs for the base system begin to rise more rapidly. The difference in the two curves at any time is the net energy cost savings in current dollars. Therefore, given the capital costs for adding the dry fluid cooler, the payback period can be calculated. The 48-ton unit that we added to the existing system in this example costs about $20,000. From the figure, it can be seen that the payback period would be about 23 years for this case. This time would be reduced if the cost of energy was higher than assumed in the calculations, or if the cooling unit could be acquired at a lower cost.

It should be noted that there are often substantial electrical energy demand costs that are charged on the basis of the maximum peak and non-peak energy usage rates (recorded over the past month); these electrical energy demand costs can be as large as the real energy costs. The dry fluid cooler system has the advantage of reducing electrical energy consumption during periods when the energy usage is likely to reach a maximum value, especially during the summertime peak hours. Therefore, the dry fluid cooler system can likely reduce the energy demand costs, improving the economics of the system.

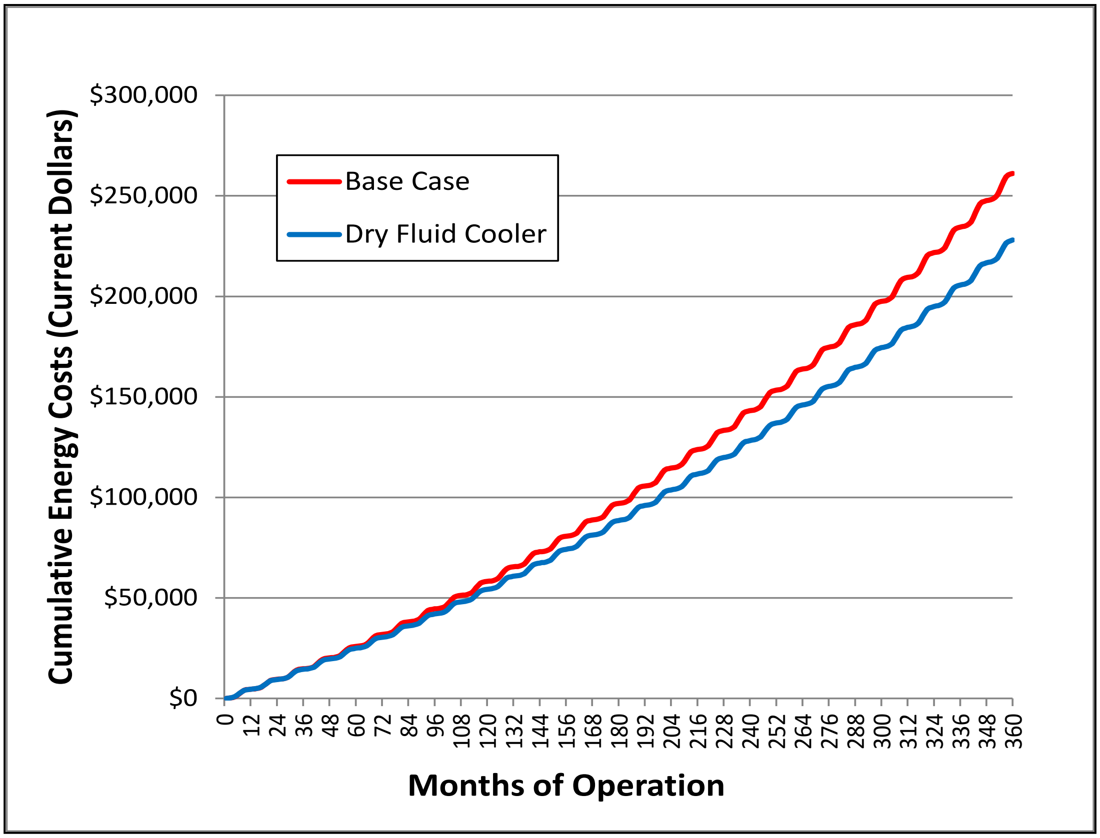

Figure 3.7. Cumulative electrical energy costs for the base case and the case with a dry fluid cooler.

4.0 EXAMPLE DESIGNS FOR DIFFERENT LOCATIONS IN THE UNITED STATES

A series of GLHEPro simulations including dry fluid coolers were performed for conditions in six cities around the US: Minneapolis, MN; Chicago, IL; Philadelphia, PA; Oklahoma City, OK; Jacksonville, FL; and Phoenix, AZ. These cities have a range of climate conditions, with average ground temperatures ranging from a low of 48.4 ^{o}F in Minneapolis to a high of 78.2 ^{o}F in Phoenix. As a result of these different climates, these locations have much different building load profiles.

4.1 Methodology

The GLHEPro simulations are based on heating and cooling loads for a hypothetical 25,000-sq. ft., two-story office building. The building loads were calculated for each city using the eQUEST building simulation tool and are shown in Figure 1.7. The hourly building loads calculated in eQUEST were converted to monthly loads for use in GLHEPro using the Peak Load Analysis Tool. This tool is distributed with the GLHEPro program. The average ground temperature for each city was calculated in GLHEPro using the internal database. The building loads for these locations range from nearly balanced (Minneapolis), to extremely cooling dominated (Oklahoma City, Jacksonville, and Phoenix).

Two simulations were performed for each location: a base case simulation with no dry fluid cooler, and a simulation using the same ground loop with a supplemental dry fluid cooler. Different ground loop configurations were used for the different cities due to the large variation in building loads. A ground loop consisting of 30 three-hundred-foot-deep boreholes with 20-ft spacing arranged in a 3x10 rectangle with a flow rate of 90 gpm was used for the cities of Minneapolis, Chicago, Philadelphia, and Oklahoma City. A larger loop with 42 boreholes in a 3x14 rectangle with a flow rate of 120 gpm was used for Jacksonville, while a loop with 60 boreholes (3x20) and a flow rate of 160 gpm was used for Phoenix. The larger ground loops were needed in Jacksonville and Phoenix to avoid excessively high temperatures.

The simulations with a dry fluid cooler assume cooler operation similar to what was described in Section 3.3. The cooler was assumed to operate year-round, but at only a fraction of its rated capacity, typically about 10-15%. It is assumed that the dry fluid cooler uses a variable speed fan drive with a control system that increases fan speed as the temperature difference between the entering water and the air increases. This results in more cooling in the winter months, and the distribution of heat rejection was assumed to be the same as it was for the Beaufort MCAS example (Table 4.1).

Table 4.1. Monthly distribution of dry fluid cooler heat rejection assumed in the GLHEPro simulations.

Month	Fraction of Yearly Heat Rejection by Dry Fluid Cooler
January	16%
February	10%
March	9%
April	6%
May	3%
June	3%
July	4%
August	3%
September	6%
October	8%
November	12%
December	19%

The dry fluid cooler was sized according to the building load and the loop flowrate. No cooler was required for the Minneapolis and Chicago buildings, which had more balanced cooling loads and low background temperatures. For the Philadelphia building, a 36-ton, three-fan unit would be appropriate. The Oklahoma City example would likely require a 48-ton, four-fan unit, while the Jacksonville and Phoenix cases use a 72-ton, six-fan and a 96-ton, eight-fan unit, respectively, due to their higher cooling loads.

The dry fluid cooler simulations used the cooler to reject either 90% (Philadelphia and Oklahoma City) or 100% (Jacksonville and Phoenix) of the building net cooling load. It should be remembered that the building cooling load only represents about 80% of the load received by the ground loop due to the waste heat produced by the heat pumps, so the systems are still not completely balanced.

4.2 Balanced Heating/Cooling Load Base Case

The eQUEST simulated building loads for a two-story office building in Minneapolis are shown in Table 4.2. The heating and cooling loads are nearly balanced (not including the heat pump waste heat). This balance, combined with the low ground temperature of 48.4 ^{o}F, results in almost ideal conditions for operation of a conventional ground source geothermal heat pump system.

Table 4. 2. Simulated building heating and cooling loads for Minneapolis, MN example.

Month	Heating Loads, kBTU	Cooling Loads, kBTU
January	104,532	0
February	74,250	235
March	42,283	992
April	4,112	9,900
May	3	43,043
June	0	88,714
July	1	100,308
August	1	98,967
September	0	56,680
October	2,547	12,508
November	38,002	1,696
December	86,070	6
Total	**351,802**	**413,049**

The GLHEPro simulation results for this case are shown in Figure 4.1.

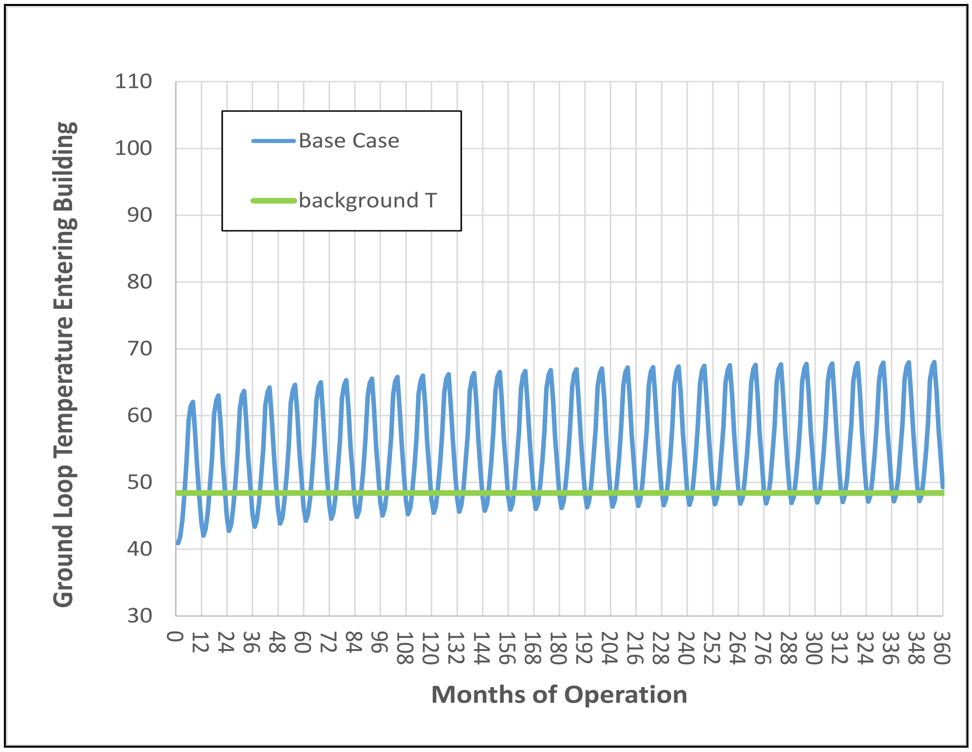

Figure 4.1. Simulated ground loop temperatures for a two-story office building in Minneapolis, MN.

Over the 30-year simulation period, the ground loop temperature increases slightly, but not enough to significantly decrease the air conditioning efficiency. In fact, the heat pump system becomes slightly more efficient over time due to improved heating efficiency with the higher loop temperatures. It appears that no supplemental cooling would be needed for this building.

4.3 Cooling-Dominated Examples

Chicago. The simulated loads for an office building in Chicago are shown in Table 4.3. The heating and cooling loads are somewhat unbalanced but not strongly so (not including the heat pump waste heat). The local ground temperature of 51.9 ^{o}F combined with the relatively weak load imbalance results in a ground loop system that does not undergo an excessive temperature increase over time.

The GLHEPro simulation results for Chicago are shown in Figure 4.2. Over the 30-year simulation period, the ground loop temperature increases slightly but not enough to substantially decrease the air conditioning efficiency, and this loss is mostly offset by the increase in heating efficiency. Over the course of the 30-year simulation, the system efficiency is nearly constant. It appears that no supplemental cooling would be needed for this building.

Table 4.3. Simulated building heating and cooling loads for Chicago, IL example.

Month	Heating Loads, kBTU	Cooling Loads, kBTU
January	71,139	41
February	53,902	164
March	36,867	771
April	4,326	8,455
May	0	43,546
June	1	91,747
July	0	108,460
August	2	100,789
September	0	66,050
October	1,029	16,721
November	20,329	3,776
December	58,584	29
Total	**246,180**	**440,550**

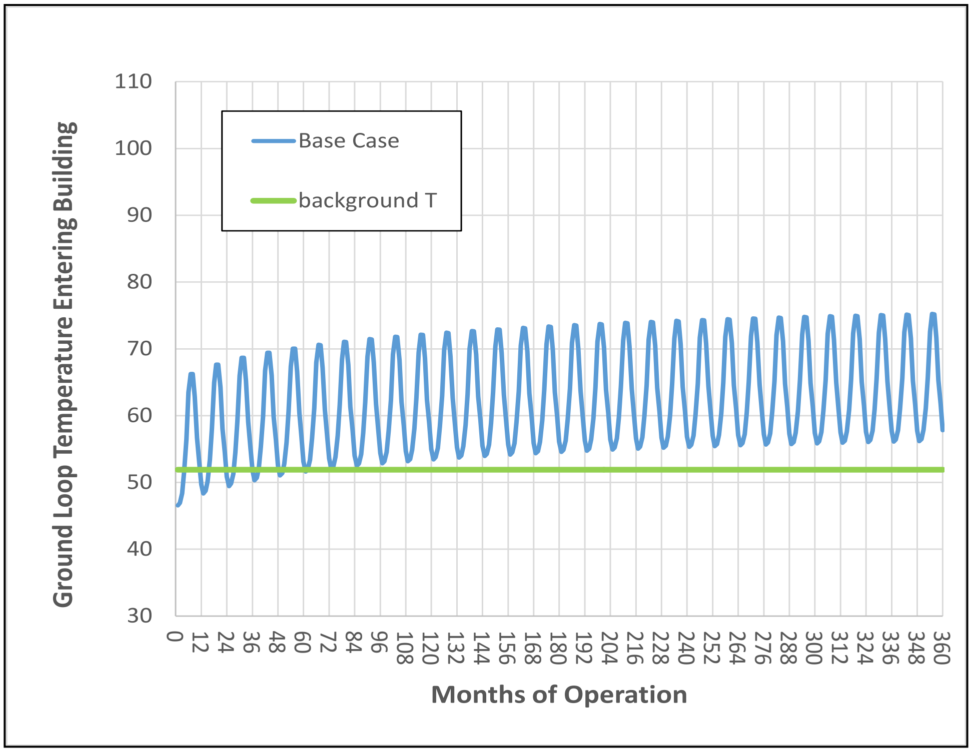

Figure 4.2. Simulated ground loop temperatures for a two-story office building in Chicago, IL.

Philadelphia. The simulated loads for an office building in Philadelphia are shown in Table 4.4. The heating and cooling loads are more strongly imbalanced, although there are relatively large heating loads in the winter. The local ground temperature is about 57.1 °F.

Table 4.4. Simulated building heating and cooling loads for Philadelphia, PA example.

Month	**Heating Loads, kBTU**	**Cooling Loads, kBTU**
January	52,833	337
February	39,656	323
March	15,480	3,132
April	1,586	16,089
May	0	48,735
June	2	93,182
July	0	113,367
August	0	107,768
September	0	81,508
October	557	33,880
November	8,645	9,191
December	36,969	408
Total	**155,727**	**507,920**

The GLHEPro simulation results for Philadelphia are shown in Figure 4.3. Over the 30-year simulation period, without a dry fluid cooler, the ground loop temperature increases by almost 20 degrees. While these loop temperatures are not excessively high compared to the more southern examples, the increased temperature reduces the air conditioning efficiency. This reduction of cooling efficiency is not offset by the increase in heating efficiency due to the lower heating loads. Over the 30-year simulation period, the heat pump system would become about 12% less efficient.

The dry fluid cooler simulation removes 90% of the cooling load imbalance (317,000 kBTU) over the course of a year, with most of the cooling occurring in the wintertime (Table 4.1). This amount of heat rejection does not completely balance the load to the ground loop, but it is enough to stabilize the temperature over time, with a rise in temperature of only a few degrees (Figure 4.3). This system would have a nearly constant efficiency over time. A 36-ton, three-fan dry fluid cooler would be an appropriate size for this building.

Given the fact that the degree of ground loop heating in this case is not severe, the argument for adding a dry fluid cooler would not be as strong as it would be for a building with a larger cooling load (or a smaller ground loop).

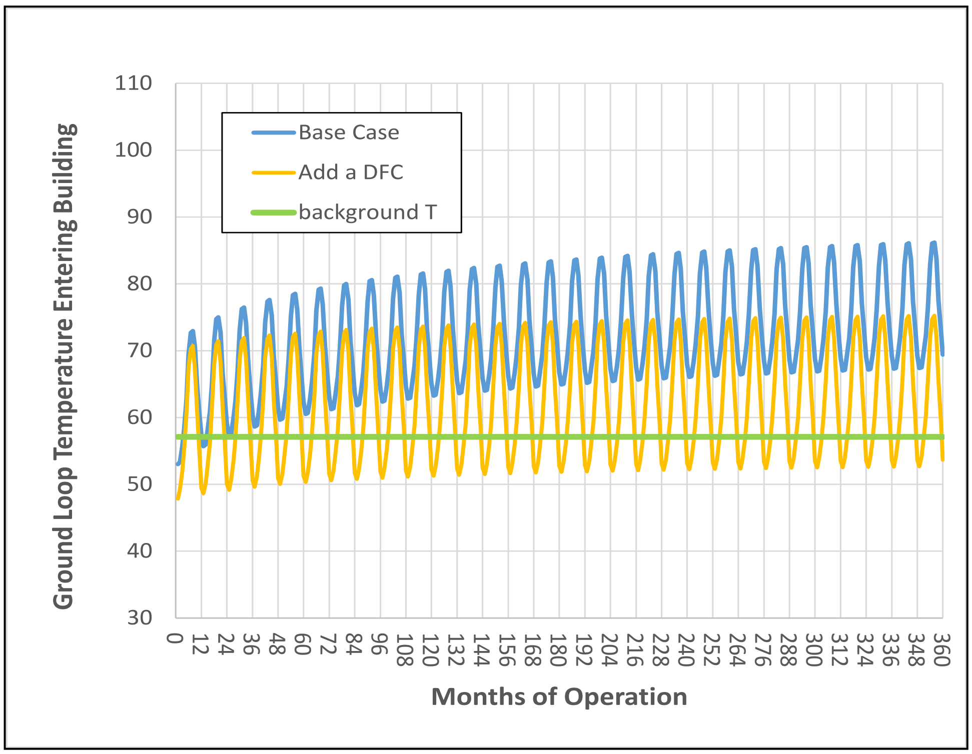

Figure 4.3. Simulated ground loop temperatures for a two-story office building in Philadelphia, PA.

Oklahoma City. The simulated loads for an office building in Oklahoma City are shown in Table 4.5. The heating and cooling loads are strongly imbalanced, and there are very high cooling loads in the summer months. The local ground temperature is about 62.4 degrees. This building would be a good candidate for the addition of a dry fluid cooler. As in the previous three simulations, a 30-borehole heat exchanger system is used for the simulations, with a flowrate of 90 gpm.

The GLHEPro simulation results for Oklahoma City are shown in Figure 4.4. Over the 30-year simulation period, without a dry fluid cooler, the ground loop temperature increases by more than 30 degrees, reaching a level that would likely start to be beyond the heat pump operating range. A large degree of heating is seen in the first year, and by the end of the simulation, the heat pump efficiency has dropped by about 30% compared to first-year operation.

Table 4.5. Simulated building heating and cooling loads for Oklahoma City, OK example.

Month	**Heating Loads, kBTU**	**Cooling Loads, kBTU**
January	42,813	1,479
February	16,090	3,007
March	3,330	14,909
April	188	41,951
May	3	70,891
June	1	114,429
July	2	127,299
August	0	142,298
September	2	93,904
October	0	52,583
November	911	15,970
December	14,087	6,822
Total	**77,428**	**685,541**

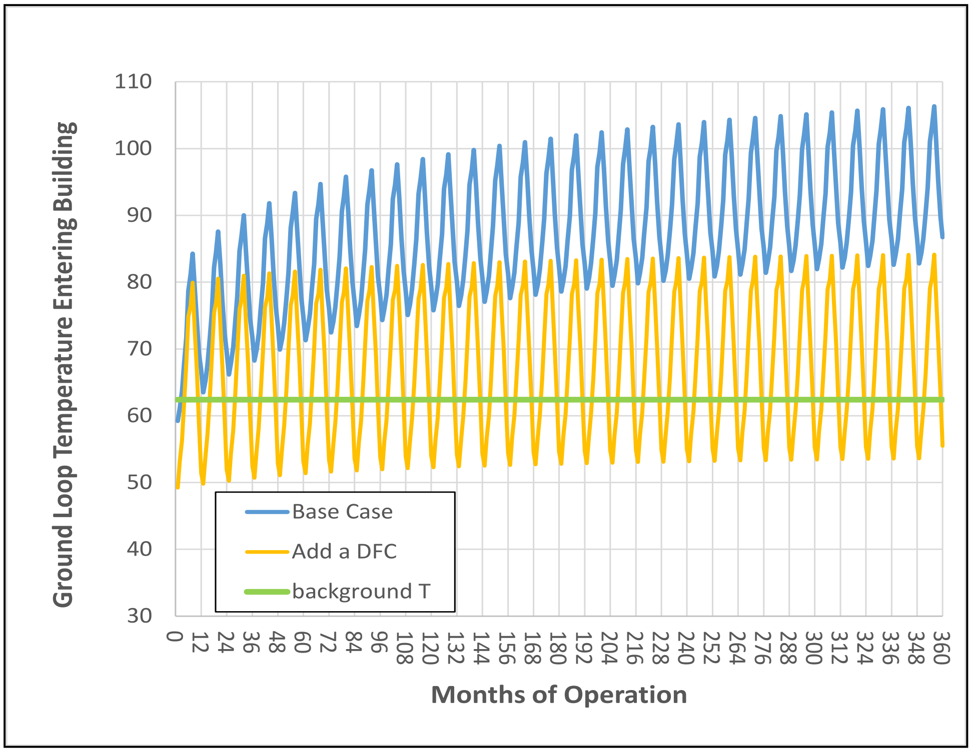

Figure 4.4. Simulated ground loop temperatures for a two-story office building in Oklahoma City, OK.

The dry fluid cooler simulation removes 90% of the cooling load imbalance (609,300 kBTU) over the course of a year, with most of the cooling occurring in the wintertime (Table 4.1). This amount of heat rejection does not completely balance the load to the ground loop, but it is enough to stabilize the temperature over time, with a rise in temperature of only a few degrees. (Figure 4.4). This system would have a nearly constant efficiency over time. A 48-ton, four-fan dry fluid cooler would be an appropriate size for this building.

Jacksonville. The simulated loads for an office building in Jacksonville are shown in Table 4.6. The building load is virtually all cooling, with high loads in the summer and significant loads year-round. The local ground temperature is about 69.3 ^{o}F. It was found necessary to increase the size of the ground loop for this example to prevent excessively high loop temperatures. These simulations increase the ground loop by 12 boreholes, to a total of 42, with a loop flowrate of 120 gpm. With its very high cooling load, this building is an excellent candidate for the addition of a dry fluid cooler.

The GLHEPro simulation results for Jacksonville are shown in Figure 4.5. Over the 30-year simulation period, without a dry fluid cooler the ground loop temperature increases by more than 30 degrees, reaching a level that would likely start to be beyond the heat pump operating range. A large degree of heating is seen in the first year, and by the end of the simulation, the heat pump efficiency has dropped by nearly 50% compared to first-year operation.

The dry fluid cooler simulation removes 100% of the cooling load imbalance (971,000 kBTU) over the course of a year, with most of the cooling occurring in the wintertime (Table 4.1). This amount of heat rejection does not completely balance the load to the ground loop (due to the additional waste heat from the heat pumps), but it is enough to stabilize the temperature over time, with a rise in temperature of only a few degrees (Figure 4.5). This system would have a nearly constant efficiency over time. A 72-ton, six-fan dry fluid cooler would be an appropriate size for this building.

Table 4.6. Simulated building heating and cooling loads for Jacksonville, FL example.

Month	**Heating Loads, kBTU**	**Cooling Loads, kBTU**
January	1,705	13,989
February	222	22,896
March	62	55,816
April	4	86,323
May	3	98,058
June	0	129,072
July	0	134,101
August	0	139,598
September	1	125,691
October	0	87,551
November	1	53,894
December	147	26,061
Total	**2,145**	**973,050**

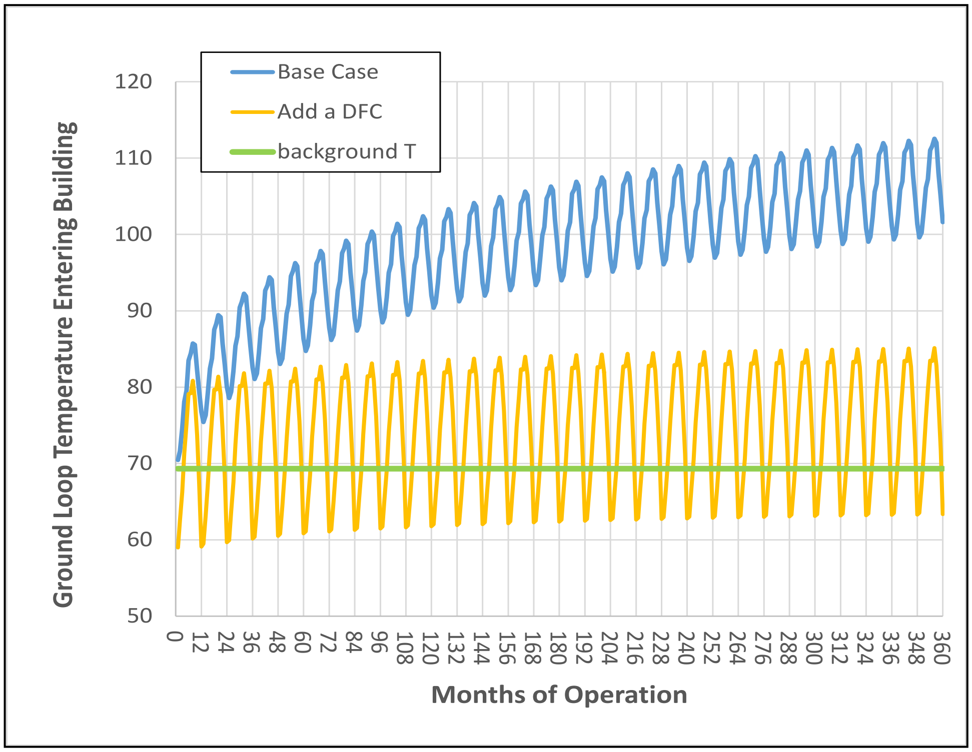

Figure 4.5. Simulated ground loop temperatures for a two-story office building in Jacksonville, FL.

Phoenix. The simulated loads for an office building in Phoenix are shown in Table 4.7. The building load is virtually all cooling, with extremely high loads in the summer and significant loads year-round. The local ground temperature is about 78.2 °F, which is far warmer than the previous examples. It was found necessary to again increase the size of the ground loop for this example to prevent excessively high loop temperatures. These simulations increase the original ground loop by 30 boreholes, to a total of 60, with a loop flowrate of 160 gpm. With its extremely high cooling load, this building is an excellent candidate for the addition of a dry fluid cooler.

The GLHEPro simulation results for Phoenix are shown in Figure 4.6. Over the 30-year simulation period, without a dry fluid cooler the ground loop temperature increases by about 30 degrees, reaching a level that would likely start to be beyond the heat pump operating range. A large degree of heating is seen in the first year, and by the end of the simulation, the heat pump efficiency has dropped by nearly 40% compared to first-year operation.

Table 4.7. Simulated building heating and cooling loads for Phoenix, AZ example.

Month	Heating Loads, kBTU	Cooling Loads, kBTU
January	457	24,775
February	29	30,333
March	4	69,503
April	3	87,853
May	3	100,331
June	0	149,613
July	0	159,107
August	0	160,018
September	0	141,656
October	10	94,121
November	7	62,093
December	916	27,814
Total	**1,429**	**1,107,216**

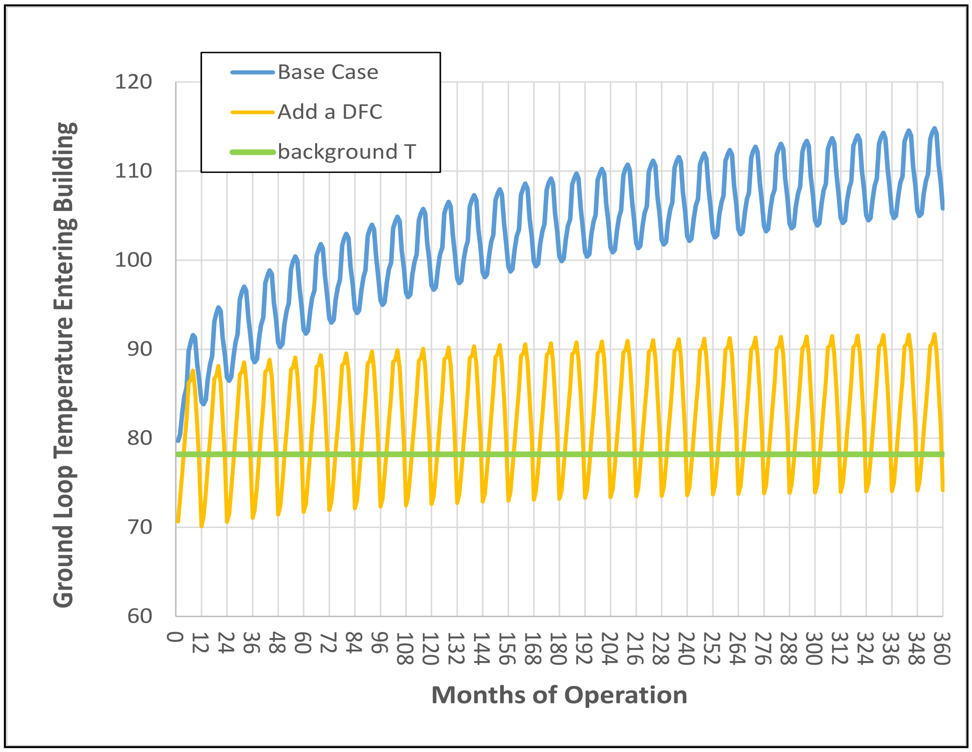

Figure 4.6. Simulated ground loop temperatures for a two-story office building in Phoenix, AZ.

The dry fluid cooler simulation removes 100% of the cooling load imbalance (1,106,000 kBTU) over the course of a year, with most of the cooling occurring in the wintertime (Table 4.1). This amount of heat rejection is enough to stabilize the temperature over time, with a rise in temperature of only a few degrees (Figure 4.6). This system would have a nearly constant efficiency over time. A 96-ton, eight-fan dry fluid cooler would be an appropriate size for this building.

It was possible to stabilize the ground loop temperature using moderately sized loops in each of these examples by using a dry fluid cooler to reject an amount of heat approximately equal to the building cooling load imbalance. Most of this heat can be rejected in the winter time, when it is most efficient to do so. This lowering of the ground loop temperature each winter pays dividends the following summer as loop temperatures remain lower.

5.0 DESIGN GUIDELINES

Geothermal heat pump systems that experience cooling-dominated building loads tend to heat up over time, reducing system efficiency. In severe cases, the ground loop temperature increase may cause loop temperatures to exceed safe operating ranges of the heat pumps. Once the ground has been excessively heated, the subsurface resource (a geothermal heat sink) has been degraded, limiting future applicability of geothermal heat pump technology at the site.

Excessive ground loop heating from cooling-dominated loads can be avoided by adding a supplemental cooling device, such as a cooling tower or a dry fluid cooler. Cooling towers are typically operated during hot summer months and rely mainly on water evaporation for cooling. Dry fluid coolers cool using heat transfer from outside air and can be operated very efficiently in the wintertime.

The following guidelines are provided for application of wintertime cooling using dry fluid coolers:

1) Calculate building heating and cooling loads using a building simulation tool such as eQUEST (Hirsch & Associates, 2016).

2) Simulate a conventional geothermal heat pump system using a ground loop simulation tool such as GLHEPro (IGSHPA, 2016).

3) Using reasonable ground loop sizes, assess the degree to which the ground loop temperature will increase over the expected life of the system. The projected loop temperature increase will depend on the building cooling and heating loads, the ambient ground temperature, and the size of the ground loop. ***If the average loop temperature increases by more than about 15 ºF, a dry fluid cooler would be beneficial.***

4) The dry fluid cooler should be sized to match the ground loop flowrate; that is, the loop flowrate should fall within the dry cooler design range. Operating below this range will greatly reduce the efficiency of the cooler, while operating above this range may lead to excessive pressure drop across the cooler.

5) The dry fluid cooler should be sized and operated so that it can reject an amount of heat equal to the yearly cooling load minus the yearly heating load. Using variable frequency drive fan motors, the fan speeds should be controlled by the temperature difference between the water entering the cooler and the outside air, reaching maximum fan speeds when the temperature difference is large (~20 degrees or more). Most of the loop heat will be rejected in the winter using this type of operation, and the heat rejection energy efficiency can be very high.

   On a yearly average basis, the dry fluid cooler should be sized so that the heat rejection is on the order of 10-20% of the cooler-rated capacity. For example, if the desired yearly cooler heat rejection is 600,000 kBTU, this is equivalent to an average rate of about 5.7 tons. This would be about 12% of the rated capacity of a 48-ton, four-fan dry fluid cooler. During the peak cooling months of December and January, the cooler is likely to be operating at 25-35% of the rated capacity (averaged over the month).

6) The ground loop system should include antifreeze to prevent damage to the dry fluid cooler during freezing temperatures.

6.0 REFERENCES

Chiasson, A.D., and C. Yavuzturk, 2009, "A design tool for hybrid geothermal heat pump systems in cooling-dominated buildings", *ASHRAE Transactions* 115(2), 74-87.

Cullin, 2008, "Improvements in Design Procedures for Ground Source and Hybrid Ground Source Heat Pump Systems", MS Thesis, Oklahoma State University.

Cullin, J.R., and J.D. Spitler, 2010, "Comparison of simulation-based design procedures for hybrid ground source heat pump systems", *Proceedings of the 8th International Conference on System Simulation in Buildings*, Liege, Belgium, December 13-15, 2010.

Hackel, S., G. Nellis, and S. Klein, 2009, "Optimizations of cooling-dominated hybrid ground-coupled heat pump systems", *ASHRAE Transactions* 115(1), 565-580.

Hirsch & Associates, 2016, eQUEST Quick Energy Simulation Tool, http://www.doe2.com/equest/, accessed 9/12/16.

IGSHPA, 2016, GLHEPro 5.0 For Windows User's Guide, Oklahoma State University, distributed by the International Ground Source Heat Pump Association, 150 p.

Kavanaugh, S.P., 1998, "A design method for hybrid ground-source heat pumps", *ASHRAE Transactions*, 104(2): 691-698.

Technical Systems, 2016, FC Series Fluid Coolers, http://www.raecorp.com/Upload/TSI/Series%20FC%20Fluid%20Coolers.pdf accessed 9/12/16.

Xing, L., 2014, "Estimations of Undisturbed Ground Temperatures Using Numerical and Analytical Modeling". PhD Thesis, Oklahoma State University, Stillwater, OK.

Xu, X., 2007, "Simulation and Optimal Control of Hybrid Ground Source Heat Pump Systems", PhD Thesis, Oklahoma State University.