PERFORMANCE ANALYSIS OF A RESIDENTIAL GROUND SOURCE HEAT PUMP SYSTEM WITH ANTIFREEZE SOLUTION

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ABSTRACT

If the minimum anticipated fluid temperature in a ground source heat pump system falls near or below 0^oC, an antifreeze mixture must be used to prevent freezing in the heat pump. The antifreeze mixture type and concentration has a number of implications for the design and performance of the system. These include the required ground loop heat exchanger length, the capacity and energy consumption of the heat pump, the circulating pump selection, pumping energy, and the first cost of the system. For example, the required ground loop heat exchanger length and first cost will decrease, due to lower permissible operating temperatures, with increasing antifreeze concentration in heating-dominated climates. On the other hand, the antifreeze also degrades the heat pump performance; operating costs can be expected to increase with increasing antifreeze concentration, and a larger capacity heat pump may be needed.

The complex interaction between all of the design variables makes it difficult to choose an optimal design, and it is desirable to have a simulation and life cycle cost analysis that can be used to evaluate all of the variable interactions, to be used as the basis for an optimal design procedure. This paper reports on a simulation procedure implemented in HVACSIM+ and a life cycle cost analysis and gives example result for a typical Canadian residential building. Four different antifreeze mixtures are considered; methyl alcohol, ethyl alcohol, propylene glycol and ethylene glycol. The life cycle cost analysis was based on the electricity costs for the heat pump and circulating pump and first costs for the heat pump, circulating pump, grout, borehole drilling, U-tube, and antifreeze.

INTRODUCTION

Ground source heat pump (GSHP) systems have been shown to be an environmentally-friendly, efficient alternative to traditional heating and cooling systems in both residential and commercial applications. However, these benefits usually come at the expense of additional first cost. Because of the significant additional first cost, a premium may be placed on optimal design. Although some work related to optimal design of GSHP systems for commercial buildings has been done, relatively little has been reported on optimal design of GSHP systems for residential systems. While most residential GSHP systems are designed by contractors using rules-of-thumb or design tools, the focus of this paper is investigating simulation-based design.

The residential GSHP system generally consists of one or more heat pumps, ground loop heat exchangers (GLHE), and circulating pumps. The GLHE is typically constructed by placing high-density polyethylene tubes in a vertical borehole or a horizontal trench. Vertical boreholes are usually used when there is insufficient surface area for a horizontal system or when trenching conditions preclude installation of a horizontal system. Although drilling a vertical borehole is usually more expensive, circulating pump power consumption is usually lower than that of a horizontal system. A schematic of a GSHP system with vertical GLHE is shown in Figure 1.

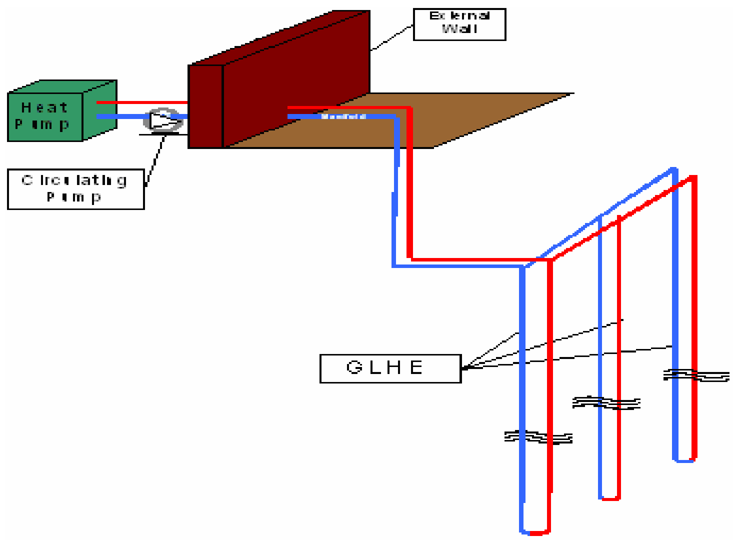

Figure 1 Schematic of the ground source heat pump system

The design problem of the GSHP system with vertical GLHE might be summarized as finding a combination of GLHE length, heat pump capacity, circulating pump, working fluid, borehole diameter, grout conductivity and U-tube diameter that allows the heating and cooling loads to be met for many years, avoids freezing of the working fluid, and minimizes life cycle cost. A typical design procedure would involve first selecting equipment and then choosing minimum and maximum heat pump entering fluid temperatures (EFT) which allow the loads to be met. In parallel, the antifreeze concentration would be chosen. The GLHE would then be sized (and other parameters – U-tube size, grout type and borehole diameters chosen) to meet the minimum and maximum heat pump EFT. Finally, a circulating pump would be chosen. All of these parameters, to some degree, trade off against each other. For example, increasing the antifreeze concentration allows lower EFT, which in turn allows shorter and less expensive GLHE. However, operating costs for heating will increase as EFT decreases. Increasing the antifreeze concentration also decreases the heat pump capacity, so a larger unit might be required.

Generally, the typical sequential design procedure leads to economical working designs. However, whether or not significantly more optimal designs might be found is unknown.

The focus of this study is the development of a computer simulation aimed at residential GSHP systems, which can account for all of the interacting design parameters. The paper describes the simulation methodology and a demonstration for a Canadian house. Several design alternatives are evaluated and discussed. Incorporation of the simulation into an optimization procedure is the focus of current work, to be reported in the near future.

BUILDING DESCRIPTION

The heating/cooling loads were calculated using the ESP-r program (Purdy 2004) for a typical Canadian residential building. The example residential building is one of the two similar test houses at the Canadian center for housing technology (CCHT), which was built to the R-2000 energy efficiency standard for research purposes. The CCHT house is composed of two above-grade floors and a fully conditioned basement. Its wood-framed construction is built upon a cast-in-place concrete foundation. It has 240 m² of conditioned floor area excluding the basement, which is typical of a modern Canadian suburban house. The nominal U-value of the above-grade walls is 0.24W/m²K, ceiling 0.34W/m²K and the windows have a U-value of 1.9 W/m²K. The basement walls are covered with RSI 2.72. The air-tightness rating of the house is 1.5 ach at 50 Pa depressurization. The house was modeled so that the living space and basement zones were conditioned by the house's HVAC system while the attic and garage were "free floating". The basement, attic space, stairwell, attached garage and two stories of living space were represented as thermal zones.

An important consideration for GSHP systems is the annual balance between heat rejection and extraction. The closer the annual balance is to zero, the smaller the GLHE that will suffice. For this house in Ottawa, the annual load is heating-dominated. However, if the basement load (which is heating only) is served by another heating source, the remaining loads, as shown in Figure 2, result in roughly balanced heat rejection and extraction over the year, For purposes of this study, we assumed that the basement load was met by an electric resistance heater.

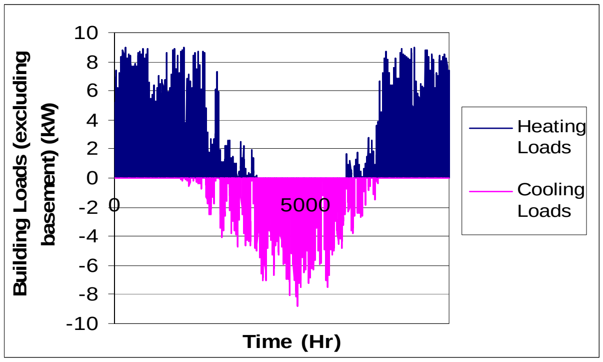

Figure 2 Annual hourly building loads excluding the heating loads on the basement

LIFE CYCLE COST ANALYSIS METHODOLOGY

Life cycle cost analysis of the system was done on a present value basis with an assumed life of 20 years and an annual interest rate of 6%. First costs and operating costs were determined based on the unit costs shown in Table 1. Annual electricity consumption for the heat pump and circulating pump were used to determine the annual operating cost.

SIMULATION METHODOLOGY

The GSHP system simulation was done using the HVACSIM+ Building Systems and Equipment Simulation Program (Clark 1985). HVACSIM+ is a non-proprietary modular simulation package developed at the National Institute of Standards and Technology (NIST), Gaithersburg, Maryland, U.S.A. A graphical user interface (Varanasi 2002) aided the development of the simulation.

The component models used to simulate the system are described in the Model Description section below. In terms of overall organization, it is of interest in this problem to simultaneously model the thermal performance and fluid flow, partly because temperature-induced changes in viscosity have the possibility of resulting in moderate changes in flow rate, depending on which antifreeze is used.

The mass flow rate at the operating point was calculated by methodology explained in detail in previous work (Khan, et al., 2003). Each component has a thermal model and a fluid flow model. From a fluid flow standpoint, the heat pump is represented as a pipe fitting and the GLHE is represented as a pipe. Both models take the mass flow rate as input and give the pressure drop as output. The circulating pump model takes the total pressure drop as input and the flow rate as output. The fluid flow models take flow temperatures as inputs from the thermal models, and with the antifreeze type and concentration set as parameters, viscosity may be determined within the fluid flow models. All models are solved simultaneously within one superblock.

Model Description

The water-to-air heat pump model is a parameter estimation-based steady state simulation model (Jin and Spitler 2002). This deterministic model is built up from models of individual components. Various unspecified parameters for individual components are estimated from manufacturer's catalog data using a multivariable optimization procedure. This model can also predict the performance variation when an antifreeze mixture is used as circulating fluid (Jin and Spitler 2003). A degradation factor (equation 1) is calculated which is multiplied by the fluid-side heat transfer coefficient (originally estimated for water.) In turn, the heat pump performance with antifreeze can be modeled.

$$DF = \frac{h_{antifreeze}}{h_{water}} = \left(\frac{\mu_{antifreeze}}{\mu_{water}}\right)^{-0.47}\left(\frac{\rho_{antifreeze}}{\rho_{water}}\right)^{0.8}\left(\frac{C_{p_{antifreeze}}}{C_{p_{water}}}\right)^{0.33}\left(\frac{k_{antifreeze}}{k_{water}}\right)^{0.67} \qquad \text{Eq. 1}$$

For this study, an hourly time step was used. Accordingly, some approximation is necessary for cycling effects. The runtime was estimated based on the building load and heat pump capacity for a given entering fluid temperature. Then the heat pump and circulating pump runtimes were used to reduce the hourly energy consumption and hourly heat transfer accordingly. No adjustment was made for transient cycling effects.

The ground loop heat exchanger model is an updated version of that described by Yavuzturk and Spitler (1999), which is an extension of the long-time step temperature response factor model of Eskilson (1987). It is based on dimensionless, time-dependent temperature response factors known as "g-functions", which are unique for various borehole field geometries. The g-function for the geometry specified can be calculated using GLHEPRO software (Spitler 2000). The model at present does not account for varying convection coefficient. The convection coefficient is calculated based on the thermophysical properties of the antifreeze mixture for the average operating temperature. It is desirable to be able to calculate the convection coefficient for each time step, particularly for any cases where the fluid might transition to laminar. For this study, the circulating pump was sized to maintain turbulent flow in the U-tubes over the entire range of temperatures.

The circulating pump model is a non-dimensional equation fit model based on similarity laws (Clark 1985). The performance of the pump is characterized in terms of mass flow rate and the shaft power requirements at the pressure drop of the system. The dimensionless performance curves that relate mass flow and efficiency to the pressure drop are estimated as 4^{th} order polynomial functions from catalog data.

The pipe model gives the pressure drop across a length of pipe for a certain diameter and roughness ratio and mass flow rate input. The fitting model, applied to the heat pump by estimating an equivalent K value, takes the mass flow rate as input and gives the pressure drop.

As all models depend at least partly on the working fluid properties, and those properties vary with concentration and temperature, thermophysical property routines were developed based on experimental data in the literature. Models suggested in the literature were either used directly or modified to improve the accuracy.

RESULTS

The ultimate goal of this work is to be able to determine an optimal design of the GSHP system that minimizes life cycle cost with antifreeze type, antifreeze concentration, heat pump capacity, circulating pump size, GLHE depth, U-tube diameter, and grout type as independent variables.

In this work we present some sample results, beginning with the choice of a base case. For our base case, we assumed:

- A nominal 3.5 ton water-to-air heat pump
- 1" nominal diameter SDR-11 HDPE pipe forming a single U-tube in the 114 mm (4.5") diameter borehole.
- Three boreholes spaced 4.6 m apart.
- Standard bentonite grout with a thermal conductivity of 0.8 W/mK
- An aqueous mixture of propylene glycol was used as the antifreeze.

To determine the antifreeze concentration and GLHE depth, the Hooke-Jeeves algorithm in Gen-Opt (Wetter 2001) was used to find the combination that gave the minimum life cycle cost. A penalty function was applied for both freezing and unmet loads. (Unmet loads result when the capacity of the equipment, which changes with the flow rate and entering fluid temperature, falls below the hourly building heating or cooling load. The penalty function forces the optimization to find a solution which neither freezes nor has unmet loads.) The optimization results in a borehole depth of 67 m or a total borehole length, for all three boreholes, of 201 m and an antifreeze concentration of 12% propylene glycol by weight.

To put this in perspective, Figure 3 shows a plot of the amount of antifreeze mixture required to prevent freezing as a function of total GLHE length. As expected, longer total GLHE lengths allow higher minimum fluid temperatures and hence permit lower concentrations of antifreeze. It is important to note that total GLHE lengths below 201 m cause unmet loads. This is because the lower entering fluid temperatures to the heat pump result in lower heat pump capacities.

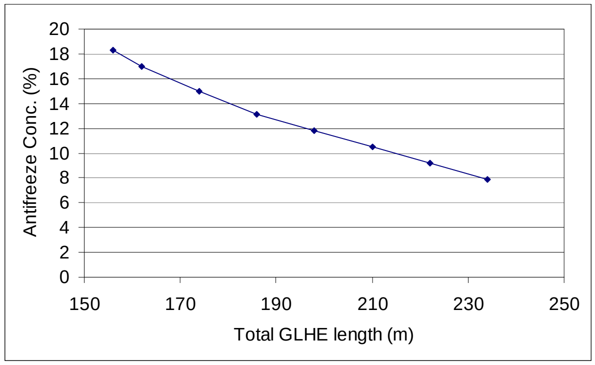

Figure 3 Amount of antifreeze mixture required to prevent freezing for a GLHE length.

Figure 4 shows life cycle costs for a range of systems with different combinations of antifreeze and GLHE length. For antifreeze concentrations below 12%, with corresponding GLHE lengths above 201 m, these combinations represent the minimum GLHE length required to prevent freezing at a given concentration. At concentrations above 12%, the GLHE length remains at 201 m to prevent unmet loads.

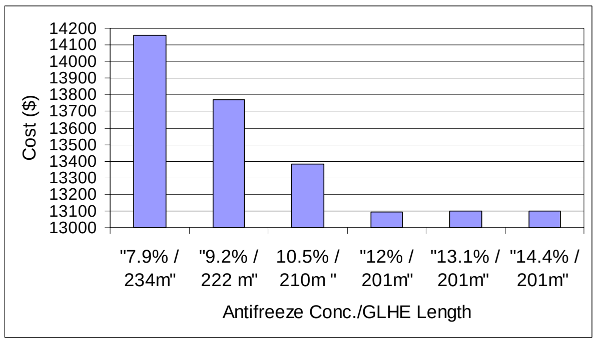

Figure 4 Life cycle cost as a function of propylene glycol concentration and GLHE length.

As described above, the basement heating loads were met by electrical resistance heater operating at an efficiency of 100%. The total annual power consumption of the heater was calculated as 5268 kWh.

The average annual electrical energy consumption for the heat pump and the circulating pump was calculated as 4427 kWh and 131 kWh respectively. The heat pump power includes the fan power consumed. A breakdown of life cycle costs for the system is shown in figure 5. The energy costs for the heat pump and the circulating pump shown are based on the present value approach and economic assumptions described above. The net present value of the system is equal to $13,095.

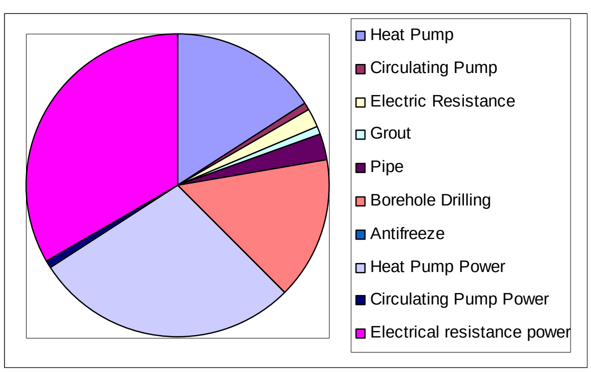

Figure 5 Base case life cycle cost breakup of the GSHP system

Grout Conductivity

Having determined a partly-optimal base case, it may be interesting to look at variations in a few other parameters. Thermally-enhanced grout contains additives (often quartz sand) that increase the thermal conductivity of standard bentonite grout. The increased conductivity results in lower borehole resistance (the thermal resistance between the fluid and the borehole wall.) This in turn allows shorter GLHE with lower drilling costs. If the GLHE is not shortened, slightly lower heat pump operating costs might be expected due to more favorable operating temperatures. However, the unit grout cost and grout installation costs are higher for thermally-enhanced grout.

To look at this, the grout conductivity was increased from 0.8 W/(mK) (bentonite grout) in the base case to 2.4 W/(mK) (thermally enhanced grout). From Table 1, the unit cost of the grout increases by a factor of 6. For the first alternative, shown in the column "Thermally Enhanced Grout" in Table 2, only the grout conductivity is changed from the base case. In this case, there is approximately a 3% reduction in the average annual heat pump energy consumption due to more favorable operating temperatures. However, the life cycle cost is increased because of the extra first cost of the thermally-enhanced grout.

A more likely scenario is that the thermally-enhanced grout is used to reduce the GLHE length. As shown in the next column in Table 2, "Thermally Enhanced Grout with decreased GLHE length", it is possible to reduce the GLHE length from 201 m to 165 m by using thermally-enhanced grout. At the same time, heat pump power is slightly reduced. However, life cycle cost is still about $100 higher than the base case, indicating that the thermally-enhanced grout option does not pay for itself.

Another option is that some blend of grouting materials can be used that gives a grout with an intermediate thermal conductivity. If we assume that the cost of the grout and its thermal conductivity can be approximated with linear interpolation between the pure bentonite grout and the thermally-enhanced grout, it is possible to find an optimal combination of the grout mix and GLHE length. Again, the Hooke-Jeeves algorithm in GenOpt was used to find an optimum combination. The resulting mixture has a thermal conductivity of 1.1 W/mK and a cost of 0.14 $/Liter. The corresponding GLHE length is 186 m and the total life cycle cost is about $20 less than the base case. While these savings are not substantial, it remains to be seen what might be achieved by simultaneous optimization of all independent parameters.

U-tube Diameter

Another parameter of interest is the U-tube diameter. In practice, the U-tube diameter could be traded off against pump size, mass flow rates, pumping power, etc. In order to look at the sensitivity, the U-tube diameter was changed to the next smaller size (nominal ¾") and no other changes were made to the system. In practice, a smaller U-tube diameter might have also allowed a smaller borehole diameter with additional savings in grout costs and drilling costs. Reducing the U-tube diameter while changing no other parameters resulted in lower mass flow rates, which in turn increased the heat pump power consumption. A negligible reduction in pumping power also occurred, though this depends on the pump curve and change in the operating point. However, the overall life cycle cost decreased from the base case because the savings in U-tube cost and antifreeze cost outweighed the increases in grout cost and heat pump power. This is shown in Table 2, in the column labeled "Decreased U-tube Diameter."

Antifreeze Mixture

The antifreeze mixture used in the system has a number of effects on the economics. These include the cost of the antifreeze, the change in the borehole resistance and heat pump performance, and the change in pumping requirements and pumping power. A few previous studies related to the use of antifreeze mixtures with GSHP systems are briefly described below.

Stewart and Stolfus (1993) made an analysis for a single operating point, for several antifreeze types. They concluded that methanol provided the best combination of good heat transfer properties with low pressure losses. Ethanol was recommended as a viable alternative to methanol solutions because of only slightly lower performance than methanol, in conjunction with lower toxicity and perceived risk.

Heinonen, et al. (1997) modeled a residential GSHP system with six different antifreeze solutions in order to estimate relative energy use and the life-cycle cost. The six different antifreeze solutions studied were methanol, ethanol, propylene glycol, potassium acetate, calcium magnesium acetate (CMA) and urea. A series of risk analyses (fire, corrosion, leakage, health, environmental impact) were also described. Regarding operating costs, ethanol had the lowest and propylene glycol the highest. For total life cycle cost, ethanol was the lowest and potassium acetate was the highest. The antifreeze mixtures were modeled only for specific concentrations and no attempt at optimization of the concentration or other parameters was made.

Again, the ultimate goal of this work is simultaneous optimization of all parameters. For this paper, we have also compared life cycle costs for several antifreeze mixtures, with concentrations that would provide the same freeze protection as 12% by weight of propylene glycol would provide. These include (by weight) 15.7% ethylene glycol, 7.6% methanol, and 11.5% ethanol. In addition, a system with pure water is compared. For the antifreeze systems, the GLHE length was kept the same for each case. The circulating pump was the same for all cases; with different viscosities, flow rates varied slightly between the cases.

For the pure water system, it was necessary to increase the GLHE size significantly. A five borehole system with boreholes 74 m deep, spaced 4.6 m apart, was sufficient to prevent freezing of the water. In addtion, a larger pump was used to keep the flow rate in each borehole approximately the same. Therefore, the overall flow rate was approximately 2/3 higher than the antifreeze cases.

The life cycle cost analysis is shown in Table 3 for the base case and each of the alternatives. Just comparing the antifreeze cases, there are relatively small differences in the energy consumption for the heat pump and circulating pump. Ethanol and methanol do give the best performance, but it is not dramatically different than the propylene glycol or ethylene glycol cases. This is likely due to the relatively low concentrations needed to prevent freezing when the loads are relatively balanced.

As expected, with much higher flow rate, more favorable operating temperatures, and the "best" heat transfer fluid, the water system shows significantly lower heat pump energy consumption and somewhat higher circulating pump energy consumption. However, the increased GLHE size dominates the life cycle cost, which is significantly higher than any of the antifreeze systems.

CONCLUSION

A system simulation of residential GSHP systems has been presented. This simulation is capable of predicting the interactions between a number of parameters, including antifreeze type, antifreeze concentration, heat pump capacity, circulating pump size, GLHE depth, U-tube diameter, and grout type. In this paper, sensitivity analyses were presented for several different variables.

For this specific case, with relatively balanced heating and cooling loads, and low antifreeze concentrations, a sensitivity analysis of the grout thermal conductivity showed that the higher cost of thermally-enhanced grout was not balanced by a large enough cost savings in the GLHE to justify its use. However, a slightly enhanced grout did show minimal savings in the life cycle cost.

When different antifreeze types with the same freeze point were used, with all other parameters held the same, the life cycle cost varied over a very small range. This is presumably partly due to the low concentrations needed for this application.

The ultimate goal of this work is to be able to determine an optimal design of the GSHP system that minimizes life cycle cost with all design parameters being treated as independent variables. To date, no large differences have been found for systems with antifreeze, so the potential usefulness of the optimization procedure remains to be seen. It is hoped that the next stage of work will show conclusively the potential usefulness or lack thereof.

In the next stage of work, systems for which heating and cooling loads are not balanced will be examined. For heating-dominated cases, it is expected that significantly higher antifreeze concentrations will be required, and therefore any differences between the antifreeze fluids may become more obvious.

Also, to date, the convective resistance between the U-tube and the fluid has been assumed constant over the simulation period. In the case with methanol as circulating fluid, the flow is always turbulent, and the minor changes in convective resistance will have little effect on the results. However, in other cases the flow rate falls into the laminar region, and it would be interesting to be able to at least roughly model the effects of this phenomena. (Given the uncertainties in transition, "roughly" may be the best that can be done.) A new model is under development which will allow time-varying convection coefficients.

ACKNOWLEDGMENTS

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NOMENCLATURE

ρ= density (kg/m3)

μ= Viscosity (Pa.s)

C_p= Specific Heat (kJ/kg-K)

κ= Thermal Conductivity (W/m K)

DF = Degradation Factor

$ = United States Dollar

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Table 1 Cost Of components of residential GSHP system

Component	Cost ($)
Pipe / m	
SDR 11 ¾ " (22mm Internal Diameter)	**0.66**
SDR 11 1" (27mm Internal Diameter)	**0.92**
Antifreeze / Liter	
Propylene Glycol	**1.06**
Ethylene Glycol	**0.71**
Methanol	**0.18**
Ethanol	**0.38**
Grout / Liter	
Bentonite Grout	**0.06**
Thermally Enhanced Grout	**0.38**
Heat Pump (Unit Cost)	
Florida Heat Pump Model (GT042)	**2060**
Electrical Resistance Heater (Unit Cost)	
Base Board Heater	**90**
Heavy Duty Heater	**180**
Circulating Pump (Unit Cost)	**120**
Drilling / m	
0.057m Borehole Diameter	**9.8**
0.076m Borehole Diameter	**10.2**
Power / kWh	**0.0725**

Table 2 Life cycle cost and power consumption of system with grout conductivity and U-tube diameter varied

	Base Case	**Thermally Enhanced Grout**	**Thermally Enhanced Grout with decreased GLHE length**	**Optimized Grout Conductivity and decreased GLHE length**	**Decreased U-tube Diameter**
U-tube Diameter (m)	0.027	0.027	0.027	0.027	0.022
Grout Conductivity (W/mK)	0.8	2.4	2.4	1.1	0.8
Borehole Radius (m)	0.057	0.057	0.057	0.057	0.057
Total GLHE length (m)	201	201	165	186	201
Heat Pump Annual Energy consumption (kWh)	4427	4304	4393	4461	4500
Circulating Pump Annual Energy consumption (kWh)	131	131	131	131	131
Electrical Heater Energy consumption (kWh)	5268	5268	5268	5268	5268
Total Annual Operating Cost ($)	712	703	710	715	718
20 years Net Present Value of Operating ($)	8,171	8,069	8,143	8,199	8,232
First Cost of the System ($)	4,924	5,629	5,059	4874	4,824
***Total Net Present Value of the System* ($)**	13,095	13,698	13,202	13,073	13,055

Table 3 Life cycle cost and power consumption of system with different circulating fluids

	Base Case (Propylene Glycol)	**Ethylene Glycol**	**Methanol**	**Ethanol**	**Water**
Antifreeze Concentration (Wt %)	12	16	8	12	0
Total Borehole Depth (m)	201	201	201	201	370
Heat Pump Annual Energy Consumption (kWh)	4439	4441	4429	4439	4117
Circulating Pump Annual Energy Consumption (kWh)	131	135	124	123	207
Electrical Heater Annual Energy Consumption (kWh)	5268	5268	5268	5268	5268
Total Annual Operating Cost ($)	713	714	712	713	695
20 years Net Present Value of Operating ($)	8,181	8,186	8,167	8,174	7,976
First Cost of the System ($)	4,924	4,921	4,903	4,908	6,962
***Total Net Present Value of the System* ($)**	13,105	13,107	13,070	13,082	14,939