Spitler, J.D. and C. Underwood. 2003. UK Application of Direct Cooling Ground Source Heat Pump Systems. Proceedings of ASHRAE-CIBSE Conference, Edinburgh, Scotland, September 24-26, 2003.

UK Application of Direct Cooling Ground Source Heat Pump Systems

J.D. Spitler[1] C.P. Underwood[2]

[1]Oklahoma State University, 218 Engineering North, Stillwater, OK 74078, USA, spitler@okstate.edu, [2]University of Northumbria, Ellison Building, Newcastle Upon Tyne, NE18ST, chris.underwood@unn.ac.uk

Summary

One form of ground-source heat pump (GSHP) system that appears promising for UK application is a system where building heating is provided via one or more water-to-water heat pumps coupled to the space heating system. Building cooling may be provided by circulating water directly between the ground loop heat exchanger and chilled ceilings or beams. In this system, the heat extracted from the ground for heating is replaced during the cooling season.

This paper presents a simulation methodology for GSHP systems, and examines a potential system for a five-story office building in Newcastle. A building simulation model is used to provide hourly building heating and cooling loads for a typical meteorological year. The system simulation is then used to predict system performance and long-term temperature response of the ground. Energy consumption for this system is compared to that of a conventional system with condensing gas boilers and vapor compression refrigeration.

Introduction

Ground source heat pump (GSHP) systems are optimal for buildings where the annual cumulative heat extraction and heat rejection are at least roughly balanced. In the UK, it may be desirable to use GSHP systems in buildings that might otherwise be heated, but not cooled. One possibility for balancing the annual heat extraction/rejection is simply to provide space cooling. This could be done in one of two ways. First, the heat pumps may be reversed to provide cooling. Second, it may be possible to provide cooling by circulating water directly between the ground loop heat exchanger (GLHE) and chilled ceilings or beams. Comparing the two alternatives, the first should allow the use of a smaller GLHE, because higher summer loop temperatures will be acceptable. However, the first alternative will consume more electrical energy to power the heat pumps in cooling mode. Thus, there will be a trade-off between first cost and operating cost.

This paper presents a simulation methodology for direct cooling GSHP systems, and examines a potential system for a five-story office building in Newcastle. A building simulation model is used to provide hourly building heating and cooling loads for a typical meteorological year. The system simulation is then used to predict system performance and long-term temperature response of the ground. Energy consumption for this system is compared to that of a conventional system with condensing gas boilers and vapor compression refrigeration.

Building

Representative time series of heating and cooling plant demands were required for the analysis of the ground-coupled systems. These were obtained using a simulation modeling method applied to a five-story, 4640 m^2 campus building at Northumbria University in Newcastle upon Tyne, UK - the Northumberland Building (Figure 1). This building was selected since, besides having traditional construction details, it has many of the characteristics of a commercial inner-city UK buildings with a mix of cellular office-type space as well as larger open-plan spaces with significant IT equipment. The building has also been extensively monitored over recent years as part of a program of work on the development of building-integrated photovoltaics and is therefore well documented in the literature (1).

Methodology – Building and HVAC Plant Simulation

In order to generate loads for both heating and air conditioning at sufficient detail for the investigation of ground-coupled systems, a simulation modeling approach was adopted. This permitted the boundary conditions applicable to heat pump-based heating and chilled ceiling cooling to be applied, as well as the treatment of cooling loads. The building and HVAC Plant simulation, described in this section, gave hourly hot water heating and chilled water cooling loads. These loads were used as boundary conditions in the heat pump and ground loop heat exchanger simulation, described in the next section.

The simulation model used was a lumped parameter method based on five-parameter second-order element descriptions collectively making up a ninth-order model for each zone. The method has been extensively reported in the literature (2,3). Natural ventilation and infiltration was set at a nominal value of 1 air change per hour but was corrected using local wind-speed data during simulation. Previous extensive monitoring of the building suggested typical zone maxima for casual heat gains:

- Lighting - 8.0 W/m^2
- Equipment – 9.9 W/m^2
- Occupancy – 1 person per 10 m^2.

Profiling was used to describe the variations of these loads during a typical day (4).

The HVAC plant utilized hot water perimeter heating with chilled ceiling cooling. Hot water heating was set to operate at water circulating conditions appropriate to heat pump-based heating (i.e. 50 ^oC flow temperature and 40 ^oC return temperatures at design conditions). The operating temperatures for the chilled ceilings were set at 16^oC flow temperature and 18^oC return temperature in order to minimize the risk of surface condensation whilst attempting to minimize the resulting mean surface temperature through the use of a narrow temperature differential.

The plant was sized based on nominal zone design internal air temperatures of 20 ^oC (winter) and 22 ^oC (summer). However, as is customary, the chilled ceiling system was designed to cater for only a proportion of the anticipated design sensible heat gain. Thus, it is expected that, on occasions, the chilled ceiling option will experience upward drifts from nominal set point. Indeed when this occurs, it will be accompanied by an adventitious increase in chilled ceiling capacity and so that the upward drift experienced will be, to some extent, self-limiting.

Simulations were conducted using Matlab/Simulink with a variable step Rosenbrock stiff solver. Weather data were supplied from a file of locally monitored data (for 1993) comprising external air temperature, external air relative humidity, wind speed and global horizontal solar radiation.

Methodology – Heat Pump System Simulation

The simulation of the heat pump system covered three types of components – the heat pump, the ground loop heat exchanger, and circulating pumps. The simulations were performed in a modular simulation environment, HVACSIM+ embedded in a graphical user interface (5).

As shown in Figures 2 and 3, three heat pumps with nominal cooling capacity of 65 kW and nominal heating capacity of 100 kW are utilized.

It might be noted here that the system simulation has been kept very simple. Issues such as sequenced control of the heat pumps, variable speed pumping, use of antifreeze, etc., have not been simulated. Therefore, the current simulations, while feasible and physically realistic, may not represent the optimal designs.

The ground loop heat exchanger (GLHE) model is that described by Yavuzturk and Spitler (6). Ground thermal properties were estimated based on discussions with a UK consultancy (7) that has experience in measuring the properties in situ. The thermal conductivity was taken at 2 W/mK and the volumetric specific heat was taken as 2.16 MJ/m^3K.

Two options were considered for the GLHE – the first supports a direct cooling system, where the water temperature leaving the ground loop heat exchanger should be no higher than 16^{o}C; in the second, for the system where cooling is provided by the heat pumps, entering water temperature may be allowed to rise somewhat higher, say to 35^{o}C. For both cases, the minimum desired heat pump entering fluid temperature was held to 4.5^{o}C in order to avoid the need for antifreeze in the system. For the first option, the maximum entering water temperature was the limiting factor – a 10x10, 100-borehole configuration, with borehole-to-borehole spacing of 6.1 m, and a borehole depth of 90 m was chosen. For the second option, the minimum entering water temperature was the limiting factor, and a borehole depth of 65 m was chosen.

The heat pump model is very simple – given the building loads and entering water temperature, it computes the heat pump power consumption and exiting water temperature using polynomial fits of the COP and ratio of heat rejection/extraction to cooling/heating provided as functions of the source side entering water temperature and mass flow rate. The polynomial fits were done for commercially-available water-to-water heat pumps. It assumes that the load is always met, and that the load side entering conditions are constant in either heating or cooling mode.

Circulating pump power was also estimated in a simple manner. Separate pumps were modeled for cooling and heating modes, except for the conventional GSHP system, where the source-side pumping was done with a single pump, year-round. For each mode of operation, the pressure rise needed at the design mass flow rate was estimated. The base pumping power was then calculated assuming a 70% efficiency.

An hourly time step was used, and the pumps were assumed to switch on and off based on a calculated run-time fraction for the heat pumps.

Methodology – Conventional System Simulation

For comparison purposes, a conventional (non-ground-source) system was simulated with a condensing boiler and an air-cooled liquid chiller. Again, the HVAC plant for this system consisted of hot water perimeter heating and chilled ceilings. The boiler fuel consumption was estimated with an equation fit of efficiency as a function of load factor, taken from a manufacturer's catalog. The cooling electricity consumption was estimated with an equation fit of COP to outdoor air temperature. The data were taken from manufacturer's data provided for an air-cooled liquid chiller with four scroll compressors and a nominal 210 kW cooling capacity. Circulating pump power was estimated in a manner analogous to that described for the GSHP systems.

Results

The loads on the heat pumps (or boiler/chiller) were estimated with the simulation procedure described above. The resulting hourly loads (shown as positive for heating and negative for cooling) are shown in Figure 4. The total annual heating loads are approximately 135 MWh; the total annual cooling loads are approximately 104 MWh. With these loads, the annual heat rejection and heat extraction will be approximately balanced.

As described above, the direct cooling GSHP system was sized to have a maximum entering water temperature no higher than 16^{o}C. As can be seen from Figure 5, which shows the hourly entering water temperatures for ten years of operation, the temperature limit is exceeded only slightly for a few hours in the first year. Over time, the temperatures drift upward very slightly, so that in the tenth year a few more hours exceeding 16^{o}C can be seen. This is due to the annual heat rejection being slightly higher than the annual heat extraction.

Figure 6 shows heat pump entering water temperatures for the conventional GSHP system. The temperatures rise to a maximum of only about 23^{o}C. In this case, the GLHE size has been kept large to avoid the use of antifreeze. It is quite likely that, with antifreeze, the GLHE size could have been substantially reduced; the consequent tradeoffs in first cost and operating cost may be a topic of future research.

The total energy consumption is summarized for all three systems in Table 1. The annual heat pump energy consumption is given for both the 1st year and the 10th year, as the amount changes slightly, due to changes in the heat pump entering water temperature. As the EWT increases, the heat pump COP in heating mode increases, and hence slightly less energy is required in the 10th year than in the 1st year.

In addition, the annual energy costs and carbon emissions were calculated based on the 10th year of operation. An electricity cost of 3.0p/kWh and a gas cost of 0.8p/kWh were assumed (8). The carbon emissions were calculated based on carbon emissions of 0.055 kg C/kWh of gas, and 0.142 kg C/kWh of electricity (9). Given the relatively low utility costs, the total magnitude of the cost savings for the GSHP systems is not large. However, the direct cooling GSHP system saves 44% in energy costs, and 57% in carbon emissions when compared to the conventional boiler/chiller system. The conventional GSHP system uses significantly more electricity to provide cooling than the direct cooling GSHP system, and, hence, the savings are somewhat lower (12% in energy costs; 31% in carbon emissions compared to the conventional chiller/boiler system).

Conclusions

This paper has presented a preliminary investigation of the feasibility of using a direct cooling GSHP system for a UK office building with hot water perimeter heating and chilled ceilings. At least in this particular case, the natural balance of the building heating and cooling loads allows a direct cooling GSHP system to function with significant energy savings and carbon emission reductions compared to a conventional chiller/boiler system.

Issues for future research include:

- The degree to which the use of antifreeze in the conventional GSHP system would allow a significantly smaller ground loop heat exchanger.
- Whether or not sequenced control of the heat pumps, or variable speed pumping would have a significant impact on the electricity consumption.
- Whether or not a system which uses direct cooling when possible, and switches to heat pump cooling only when necessary might give annual operating costs approaching those of the direct cooling system, while reducing the required GLHE size, and hence, the first cost.

References

1. Horne M, Hill R and Underwood C (1997) Visualisation of photovoltaic clad buildings *Proc. International Conference on Information Visualisation* London 173-8

2. Gouda M M, Underwood C P and Danaher S (2002a) Modelling the robustness properties of HVAC plant under feedback control *Proc. SSB 2002: 6th Int. Conf. on System Simulation in Buildings, Liege*

3. Gouda M M, Danaher S and Underwood C P (2002b) Building thermal model reduction using nonlinear constrained optimization *Building and Environment* 37(12) 1255-65

4. Jones A D and Underwood C P (2003) A method for profiling lighting and small power demands in buildings *Building Serv. Eng. Res. Technol.* 24(1) 47-53

5. Varanasi, A. 2002. Visual Modeling Tool for HVACSIM+. M.S. Thesis. Oklahoma State University.

6. Yavuzturk, C., J.D. Spitler. 1999. A Short Time Step Response Factor Model for Vertical Ground Loop Heat Exchangers. ASHRAE Transactions. 105(2): 475-485.

7. Curtis, R. 2003. GeoScience, Ltd., Falmouth, UK. Personal communication.

8. CIBSE 1998. CIBSE Guide: Energy efficiency in buildings. London: CIBSE

9. http://www.dti.gov.uk/energy/inform/energy_prices/mar_03.shtml

Figure 1: Northumberland building (south façade)

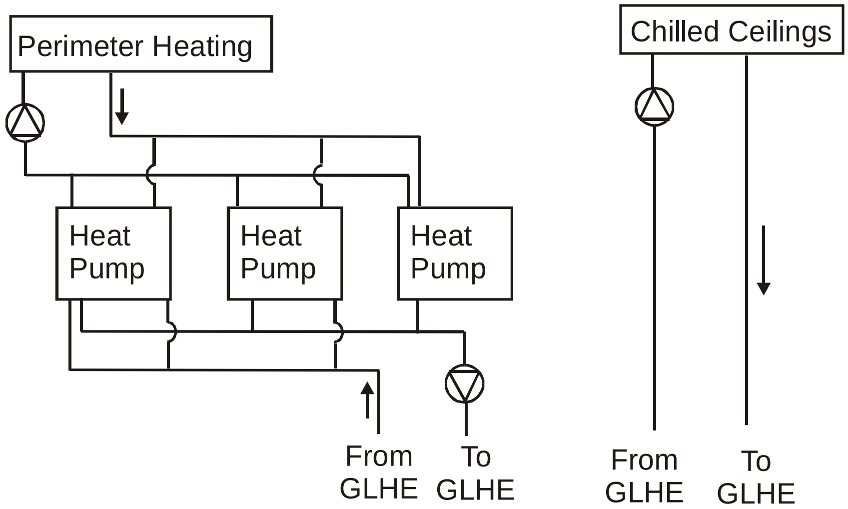

Figure 2: Direct Cooling GSHP System

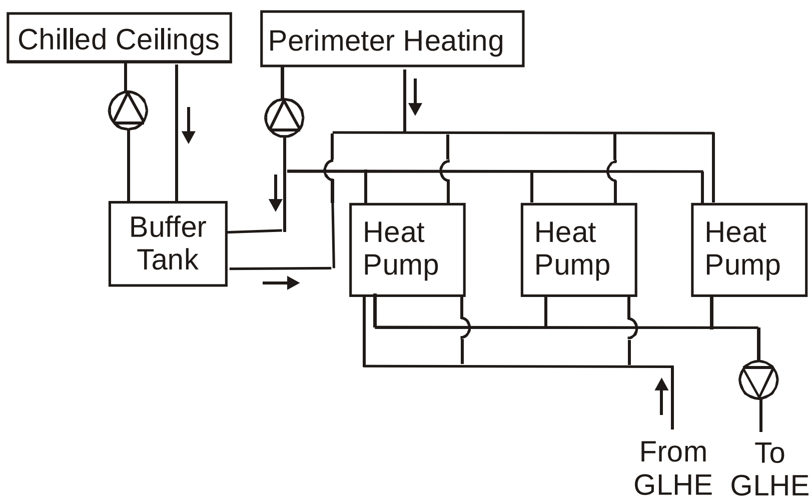

Figure 3: Conventional (Heating and Cooling) GSHP System

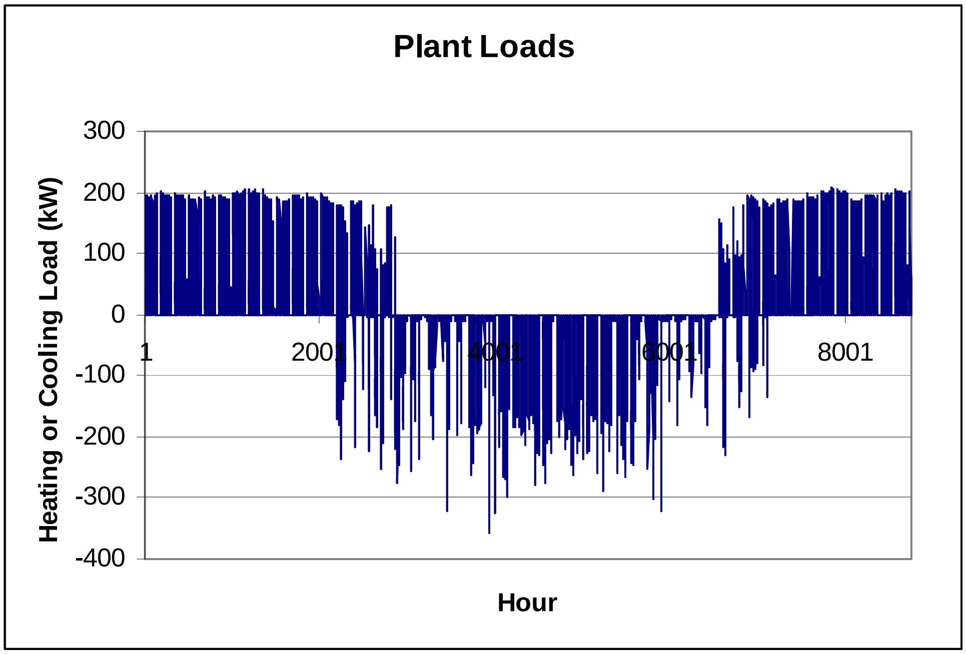

Figure 4: Hourly Plant Loads for Building

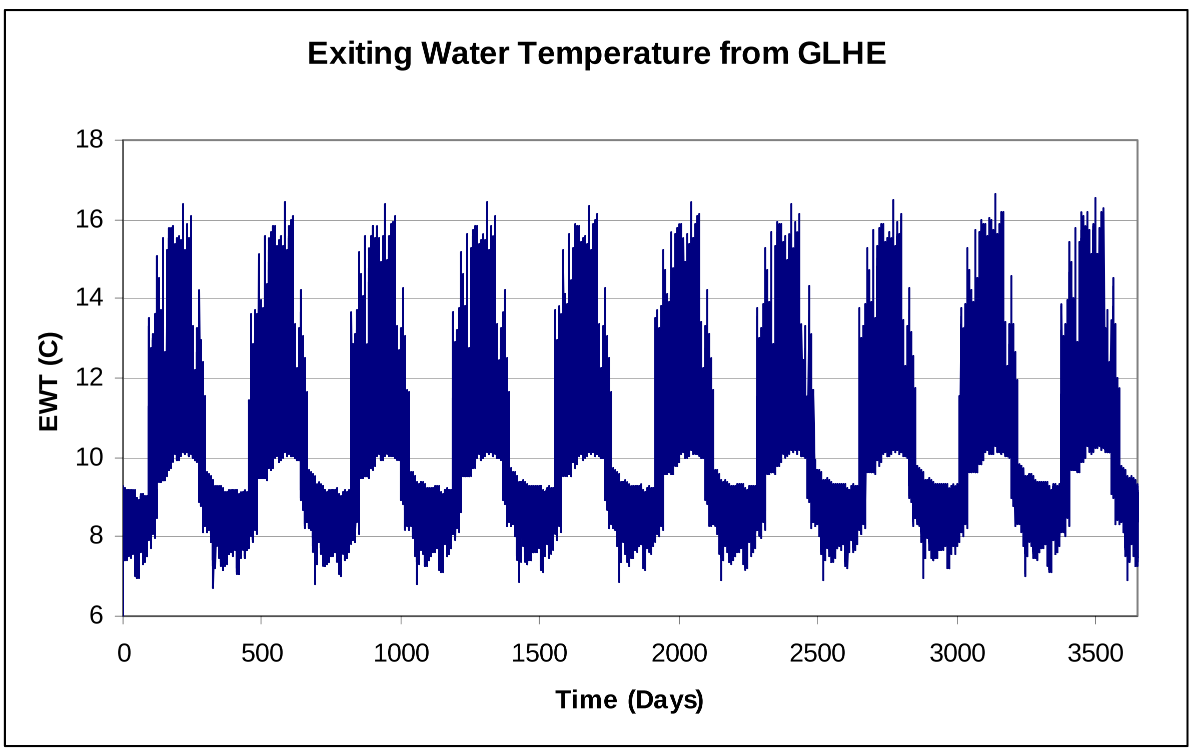

Figure 5: Entering water temperature to heat pump (in heating mode) or buffer tank (in cooling mode)

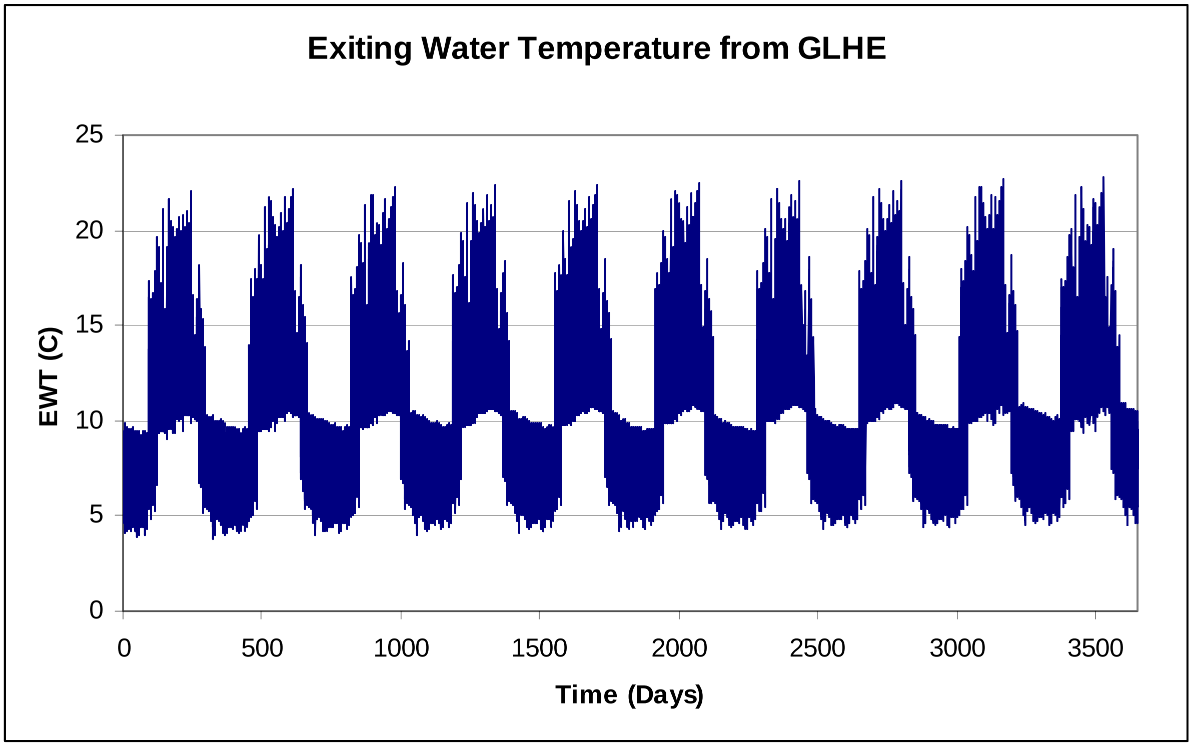

Figure 6: Entering water temperature to heat pump for conventional GSHP system

Table 1: Comparison of System Performance

	Direct Cooling GSHP System	Conventional GSHP System	Conventional Chiller/Boiler System
Annual Heat Pump Energy Consumption -1st year (kWh)	32932	52836	0
Annual Heat Pump Energy Consumption -10th year (kWh)	32773	52618	0
Annual Air-cooled Chiller Energy Consumption (kWh)	0	0	21102
Average Annual Circulating Pump Energy Consumption (kWh)	2244	2807	1874
Annual Gas Consumption (kWh)			149164
Annual Energy Cost (£)	1051	1663	1883
Annual Carbon Emissions (kg)	4972	7870	11467