Energy Savings and Collateral Impacts of a DHW Water-to-Water Heat Pump System with Subslab Earth Coupling

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ABSTRACT

This paper presents research results from simulations and a field investigation of a domestic hot water (DHW) system incorporating a water-to-water heat pump and a subslab earth-coupled loop. The research stems from a need for nonsolar, high-efficiency, DHW system alternatives to help reach the whole-house energy savings goals of 50% and higher. A schematic diagram of the system is used to explain the system concept, emphasizing daily average DHW loads and the resulting load density imposed on the slab. A comparison to air-to-water heat pumps summarizes the primary advantages and market barriers of this system type. A summary of the analysis using a transient system simulation model is reviewed, and sensitivity to key system parameters and user schedules is discussed. The relative importance of subhourly simulation time steps and hot water demand schedules is explained. Collateral energy impacts of individual system components are then presented on monthly and annual bases and compared to results from a base-case simulation incorporating a conventional electric resistance water heater. Additional or reduced heat flux through the slab and standby tank losses/gains are of particular interest, as are variations due to foundation type and climate. The calculation of the net whole-house coefficient of performance is explained and total annual energy consumption is presented.

INTRODUCTION

Domestic hot water (DHW) energy use accounts for an average of 17% of all residential site energy use in a typical single-family home (EIA 2001). This percentage can be as high as 40% in an energy-efficient home as the energy efficiency of the building envelope and space-conditioning system increases. Current federal minimum energy factors for water heaters (DOE 2001) are 0.59 for a typical 40-gallon gas-fired tank-type water heater and 0.90 for a 50-gallon electric water heater. Maximum energy factors (GAMA 2007) are 0.85 for gas-fired water heaters and 2.28 for air-to-water heat pump water heaters (HPWHs).

Although it may appear possible to attain large energy savings by using an HPWH, there are two other factors one must consider. Current Building America goals (NREL 2007) target source energy savings. Applying national average site-to-source energy multipliers, the best gas-fired water heater still has a source-energy factor of 0.83, and an HPWH drops to only 0.72. The second factor that is not normally considered in the calculation of the energy factor is the net energy impact on the space-conditioning system. Most water heaters only add a small amount of heat to the space, but an HPWH removes about as much thermal energy from the space as electrical energy it consumes. For example, an air-to-water HPWH operating in a space heated by an air-to-air heat pump with a heating seasonal performance factor (HSPF) of 9.90 has a net source-energy factor of 0.51 during the heating season. On the other hand, heat removed from the space during the cooling season increases the net source-energy factor to 0.94. These interactions tend to decrease the net efficiency of HPWHs in heating-dominated climates and increase it in cooling-dominated climates. A more noticeable effect of these interactions is the cooler space temperature surrounding the HPWH year round.

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This research focuses on a water heating system that incorporates a water-to-water heat pump system with a source/sink exterior to the conditioned space of the house while requiring minimum additional excavation. The target applications are projects in climates or situations in which solar DHW systems would not be practical or economically feasible. Preliminary system cost estimates, based on retail costs of commercially available air-to-water HPWHs and piping costs, put the installed cost of a mature, market-ready, water-to-water heat pump system at approximately $2,500. Although this is several hundred dollars more than the installed cost of a high-efficiency, tankless, gas-fired water heater, it is less than many solar DHW systems (Burch and Salasovich 2005).

EXPERIMENTAL DESIGN

Evaluation of the system shown in Figure 1 began with computer simulations using TRNSYS (TESS 2006b) to determine the net annual energy efficiency of a water-to-water HPWH and the impact of such a system on subslab earth temperatures. Simulations were conducted for both warm and cold climates and two foundation types: slab-on-grade and full basement. Simulation results from the prototype system were then compared to simulation results from similar systems, with the primary difference being that the earth-coupled piping loop was modeled remotely from the building.

Results from the simulations were then used to design and construct a prototype system for field testing. Measured data were then compared to simulation results to determine the sensitivity of the simulation output to various system parameters.

System Description for Simulations

A water-to-water heat pump extracts heat from under an insulated slab to heat water via the integral heat exchanger in the hot water tank. Components shown in Figure 1 are described in the following paragraphs.

Hot Water Tank. The hot water tank is a solar-ready, 300-liter (80-gallon) storage tank with a 4,500 W heating element in the upper part of the tank and an integral heat exchanger in the lower part. The tank is modeled in TRNSYS as a stratified storage tank using Type 534 (TESS 2000). For simulation purposes, the tank volume is divided into eight vertical nodes and the heat exchanger is divided into four nodes equally distributed within the bottom half of the tank. The auxiliary heating element setpoint temperature is 50.0°C (122°F) with a deadband of 10.0°C (18.0°F). To minimize energy use of the auxiliary heating element, it is important that the setpoint temperature be equal to or lower than the heat pump setpoint temperature and that the temperature deadband be larger.

The integral heat exchanger consists of a 40 m (131 ft) coiled-copper pipe with an inside diameter of 16 mm (0.625 in.).

Heat Pump. The heat pump is a water-to-water heat pump with a nominal output capacity of 5.3 kW (1.5 ton) designed for DHW service. Catalogued performance data in Figures 2 and 3 illustrate the range in efficiency and capacity with respect to inlet load and source water temperatures. The ellipses in each figure identify the region of operation for the system. The fluid in the source piping loop is pure water and the fluid in the load loop is 50% propylene glycol and 50% water. The heat pump is controlled by an aquastat with a setpoint temperature of 50.0°C (122°F) and a deadband of 5.0°C (9.0°F) located in the bottom of the hot water storage

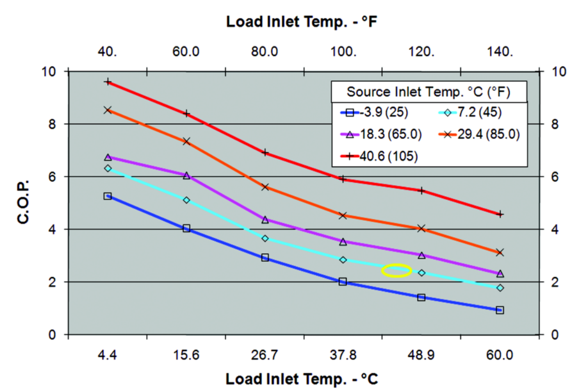

Figure 2 *Heat pump efficiency.*

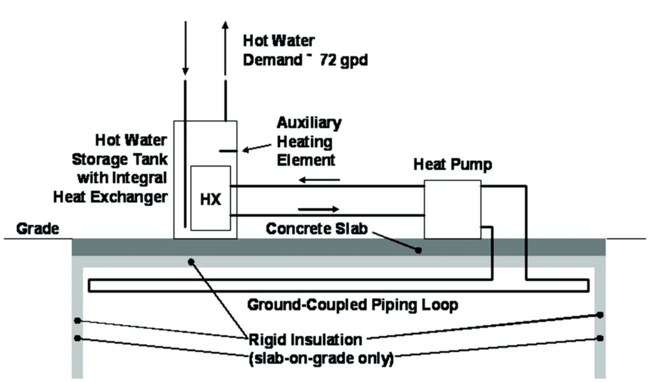

Figure 1 *System schematic.*

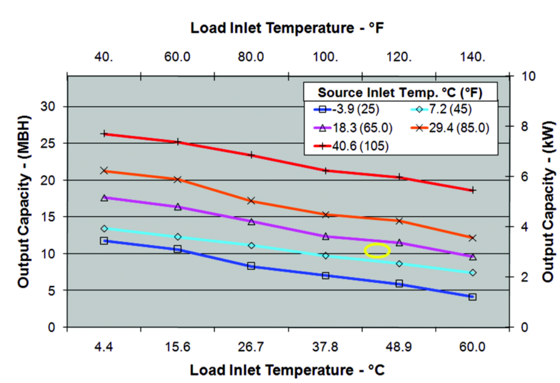

Figure 3 *Heat pump capacity.*

tank. A freeze stat prevents operation of the compressor when the source outlet temperature is below 1.7°C (35°F). The heat pump is simulated using TRNSYS Type 668 (TESS 2001).

Earth-Coupled Piping Loops. The subslab piping loop was modeled as a single-circuit serpentine of PEX piping in a 0.2 m (7.9 in.) thick sand bed under the insulated slab. Figure 4 shows the layout and Table 1 lists the parameter values used in the simulations.

Identical piping parameters were used when simulating earth-coupled loops that are remote from the slab for comparison purposes. In these cases, TRNSYS Type 952 (TESS 2006a) was used to model the buried pipe in two horizontal circuits 2 m (6.6 ft) below the surface. In the Type 952 model, the piping is assumed to be spaced far enough apart to disregard interactions between parallel pipe sections. Far-field ground temperatures are based on the Kusuda correlation (Kusuda and Achenbach 1965) so that long-term temperature changes due to the heat flux into the pipe are not considered.

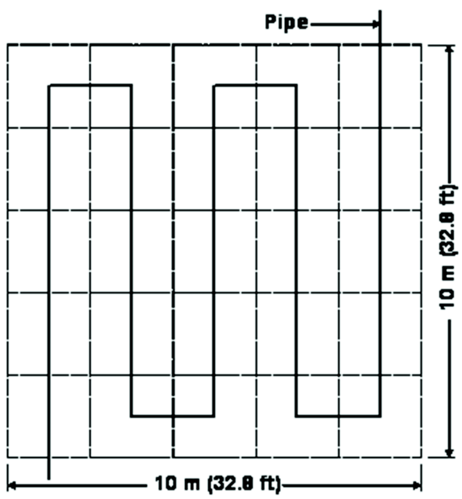

Figure 4 *Subslab piping layout.*

Foundations. The system was simulated using two different foundation conditions: slab-on-grade and full basement. For both simulations, TRNSYS Type 706 (TESS 2003) was used. This component is intended to model a radiant floor-heating slab embedded in soil and containing a number of fluid-filled pipes. The heat transfer within the slab and surrounding soil is assumed to be conductive only, and moisture effects are not accounted for in the model. The model relies on a three-dimensional finite difference method, solving the resulting interdependent differential equations using an iterative approach. Slab nodes are defined by the user in each geometric plane and extend beyond the slab in all directions except above. The floor that was modeled in both cases was a 0.10 m (3.9 in.) concrete slab over 0.51 m (2 in.) of rigid insulation with a thermal resistance of 0.49 h·m^2·K/kJ (10 h·ft^2·°F/Btu). In the slab-on-grade case, vertical insulation also extends to a depth of 0.7 m (28 in.) into the ground around the perimeter of the slab. For these simulations, radiant heat transfer at the earth's surface was included in the slab-on-grade case but not in the full basement model. Soil temperatures at the slab plane in the full basement model were described using the Kusuda correlation (Kusuda and Achenbach 1965) and the soil and ground temperature properties are listed in Tables 2 and 3. The Kusuda correlation was also used to describe the earth surface temperature of the slab-on-grade model because the energy balance surface mode does not consider evaporation, partial shading due to grass, and seasonal changes in ground reflectance due to snow.

DHW Temperature and Usage Schedule. Hot water events were described as discrete fixture flow rates and temperatures rather than specific hot water volumes to account for variations in flow due to seasonal variations in mains water temperature and instantaneous variations in hot water temperatures due to nonuniform temperatures in the storage tank. Parameter values for each event are shown in Table 4.

Table 1. Earth-Coupled Piping Parameters

Pipe total length	150 m (492 ft)
Pipe inside diameter	13 mm (0.51 in.)
Pipe outside diameter	15 mm (0.59 in.)
Pipe material thermal conductivity	1.661 kJ/h·m·K (0.266 Btu/h·ft·°R)
Pipe material density	200 kg/m^3 (12.5 lb/ft^3)
Pipe/earth contact resistance	0.0001 h·m·K/kJ (0.002044 h·ft·°R/Btu)
Fluid thermal conductivity	2.066 kJ/h·m·K (0.332 Btu/h·ft·°R)
Fluid density	1000 kg/m^3 (62.4 lb/ft^3)
Fluid specific heat	4.19 kJ/kg·K (1.00 Btu/lb·°R)
Fluid viscosity	5.468 kg/m·h (3.228 lb·s/ft^2)

Table 2. Ground Temperature Properties

Climate	Mean Surface Temperature, °C (°F)	Surface Amplitude Temperature, °C (°F)	Shift, Days
Cold (Chicago, IL)	9.8 (49.1)	14.1 (25.7)	15
Warm (Miami, FL)	24.8 (76.7)	4.3 (7.8)	21

Table 3. Slab and Soil Properties

Material	Density, kg/m^3 (lb/ft^3)	Conductivity, kJ/h·m·K (Btu/h·ft·°R)	Specific Heat, kJ/kg·K (Btu/lb·°R)
Slab	2400 (150)	2.15 (0.345)	0.90 (0.215)
Soil	2091 (131)	1.50 (1.10)	0.84 (0.201)

Table 4. DHW Event Temperature and Schedule

Event	Event Type	Flow Rate, L/m (gpm)	Duration, min	Temperature, °C (°F)	Start Time
Shower	Temperature dependent	9.5 (2.5)	7.5	40.0 (104)	7:00 a.m., 7:30 a.m., 7:00 p.m., 7:30 p.m.
Laundry	Temperature independent	9.5 (2.5)	7.5	Varies*	11:00 a.m.

* Equals instantaneous temperature of storage tank outlet.

SIMULATIONS

Simulations where performed for two cities, Chicago and Miami, and two foundation types, slab-on-grade and full basement. Due to the impact of the system on long-term ground temperatures, each simulation was run for a period of five years, and the results are shown for the last 12 months. A simulation time step of 7.5 min was used so as not to exceed the duration of the shortest hot water event. This is necessary to maintain acceptable hot water delivery temperatures, which would not be possible using a time step of one hour.

SIMULATION RESULTS

Results were calculated on a monthly basis. Several different coefficients of performance (COPs) are shown so that performance can be compared to the heat pump manufacturer's data, efficiencies of other water heating systems, and the whole-house or integrated efficiency.

Heat Pump COP. The heat pump COP represents the heat pump energy output divided by the energy input of the compressor.

System COP. The system COP is calculated by dividing the DHW-delivered energy by the sum of the energy input of the compressor, circulating pumps, and auxiliary electric heating element.

Integrated COP. The calculation of the integrated COP goes two steps further than the system COP and includes the heat gain to the space from the storage tank and the net heat loss/gain through the floor slab. To determine these values, the same model was run without the heat pump to benchmark monthly heat losses through the floor slab. For heating season months, heat gains to the space caused by the DHW system are considered part of the output energy of the DHW system. For the cooling season, heat gains/losses to the space are divided by the COP of an efficient space-conditioning system (5.57) and that value is then added to the energy input sum of the DHW system.

Sensible and latent loads from hot-water end uses, such as showers and baths, were not simulated and, therefore, were not included in this COP due to the unknown and complex interactions of these loads with bathroom exhaust fans.

Cold Climate (Chicago)

Simulation results indicate that the proposed concept provides energy efficiency levels that are equal to or better than a conventional system in which the earth coupling is remote from the foundation of the house. Analysis performed using Chicago weather data indicates that the annual average integrated COP is approximately 2.0 for both slab-on-grade and full basement foundation application types (Figure 5), whereas a similar system with a remote earth-coupling (also Figure 5) yielded an annual COP of 1.7 to 2.1. (The remote system was modeled in two different ways, which are discussed later in the paper.) A comparison of the system COP and the integrated COP of the prototype system illustrates the negative effect of the additional heat loss through the slab during the heating months and the beneficial effect in the summer months. The same comparison of the remote system illustrates the opposite effect due to the lack of any additional

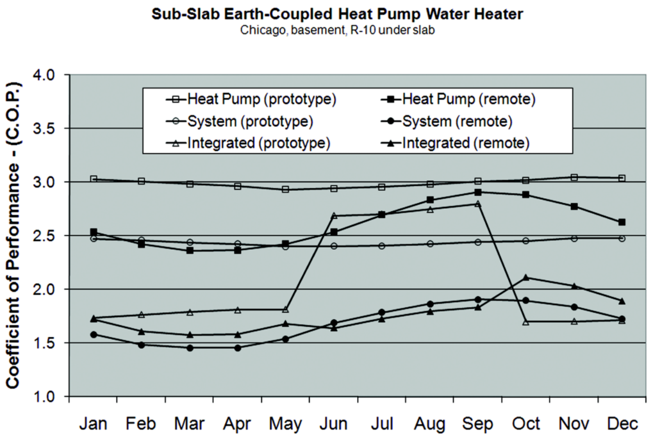

Figure 5 *Simulated monthly system efficiencies—Chicago (full basement).*

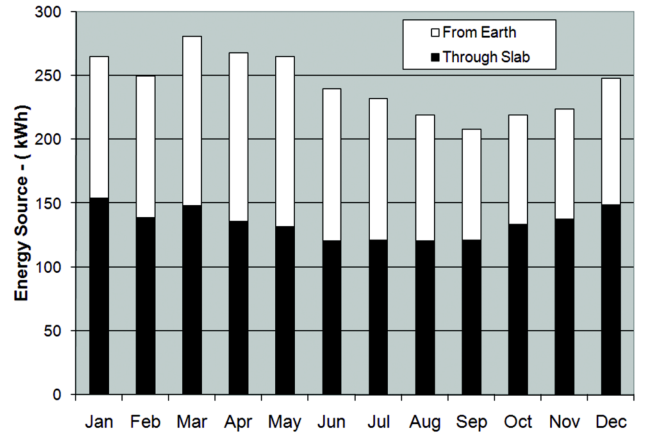

Figure 6 *Heat pump ground loop heat energy sources—Chicago.*

heat flux through the slab. The difference between the system COP and the integrated COP for the remote system is purely due to the tank losses.

Figure 6 illustrates the heat energy components of the heat pump ground loop. On a monthly basis, 50% to 61% of the ground loop energy going into the system comes through the slab. The annual average is 55%.

Results from one case, in which slab insulation levels were doubled to 0.98 $h \cdot m^2 \cdot K/kJ$ (20 $h \cdot ft^2 \cdot °F/Btu$), indicated a slight decrease in annual aggregate performance of the DHW system due to lower subslab soil temperatures. This is not to say that a house with more slab insulation would use more total energy, but it does indicate that this type of DHW system may be less feasible as slab insulation is increased. Comparing the nominally insulated case to the double-insulated case, space-conditioning energy usage would decrease about 135 kWh/yr, while DHW energy usage would increase 95 kWh/yr for a net savings of 40 kWh/yr.

Long-term subslab temperatures were evaluated to assess the risk of freezing the soil beneath the slab. Figure 7 illustrates the average monthly subslab temperatures with and without the prototype system, with an annual delivered hot water energy total of approximately 4,920 kWh/yr (167.8 therm/yr).[1] Although there does not appear to be any risk of freezing the soil, the long-term operation of the system decreases the subslab soil temperature an average of 17.2°C (30.9°F).

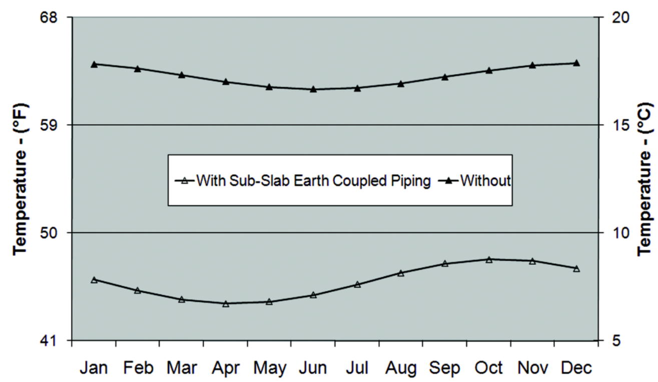

Figure 7 *Simulated monthly subslab earth temperatures—Chicago (full basement).*

Warm Climate (Miami)

Analysis performed using Miami weather data indicates that the annual average integrated COP is approximately 2.9 for slab-on-grade applications (Figure 8), whereas an identical system with a remote earth-coupling (also Figure 8) only

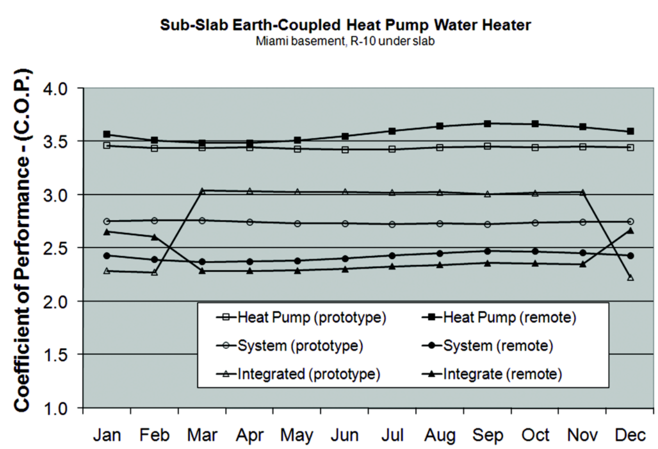

Figure 8 *Simulated monthly system efficiencies—Miami (slab-on-grade).*

1. This is about 12% greater than the U.S. DOE standard of 4,391 kWh/yr (149.8 therm/yr).

yields an annual COP of 2.3. Most of the difference is realized from the fact that the additional heat losses through the slab imposed by the proposed system help to offset cooling loads for 9 out of 12 months of the year.

The annual delivered hot water energy total, using the same hot-water end-use schedule, was approximately 2,656 kWh/yr (90.62 therm/yr) or 54% less than in Chicago.[2]

Equipment Runtime

Equipment runtime for the prototype system was less than the remote system in both climates. The cold climate prototype system ran 1,560 h/yr—31% less than the remote system. The warm climate prototype system ran 725 h/yr— 13% less than the remote system.

FIELD INVESTIGATION

A prototype of the system was constructed in an unoccupied new home near Knoxville, Tennessee, using an off-the-shelf water-to-water heat pump and a hot water tank with an integral heat exchanger for solar applications. Approximately 140 m (460 ft) of 13 mm (0.5 in.) PEX tubing was installed in two circuits under the insulated concrete floor slab. The hot water tank has a capacity of 300 L (80 gal) with 36.6 m (120 ft) of 16 mm (0.625 in.) copper piping serving as a heat exchanger. A 4,500 W auxiliary heating element is located three quarters of the way up the tank. It is set to 46.1°C (115°F) with an estimated deadband temperature of 11.1°C (20°F). The heat pump is controlled by a digital controller via a thermistor, with a setpoint temperature of 46.1°C (115°F) and a deadband of 2.8°C (5°F) in the bottom of the tank. A freeze stat prevents operation of the compressor when the source outlet temperature is below 1.7°C (35°F). The load side of the system is filled with a 50/50 water/propylene glycol mix because it is shared with a solar collector on the roof.

Operating Schedules

The system is set up with a solenoid valve and a thermostatic mixing valve to repeat the same temperature-dependent hot water events each week using three different daily schedules. Each daily schedule is designed to approximate a different total daily volume of hot water centered on the U.S. DOE water heater test standard of 237 L (62.3 gpd) or 12.030 kWh/day (41,045 Btu/day). The low hot water use is designed to be approximately 30% less than the DOE standard and draws 9.5 L/m (2.5 gpm) of mixed water from a mixing valve set at 40°C (104°F) four times each day for eight minutes starting at 6:30 a.m., 7:00 a.m., 9:00 p.m., and 9:30 p.m. The average hot water use includes an additional 15-minute event at the same temperature and flow and is designed to be approximately equal to the DOE standard. The high hot water use is approximately 30% above the DOE standard and includes another 15-minute event in addition to the average daily hot water volume.

[2] This is about 40% less than the U.S. DOE standard of 4,391 kWh/yr (149.8 therm/yr).

Measured Data

Data are measured every minute using two thermocouples and one flowmeter for each of the three fluid flows in the system. Energy output and heat transfer rates are calculated for the DHW and the source and load fluid flows of the heat pump. Two watt transducers record electrical energy usage of the heat pump and auxiliary water heater.

Efficiency. Figure 10 illustrates the COP at various levels within the system for a single twenty-four-hour period in late May, with subslab earth temperatures of 18.3°C (65.0°F). Three different end-use schedules were evaluated. The average mixed end-use water volume is 444 L/day (118 gpd). High and low end-use volumes are ±142 L/day (38 gpd) of the average. As expected, the efficiency at each system level increases as the total daily hot water volume increases.

The compressor COP, defined as the measured heat pump output divided by the measured electrical input of the compressor, ranged from 2.60 to 2.73. This is much lower than

Figure 9 *Prototype DHW Heat Pump System*

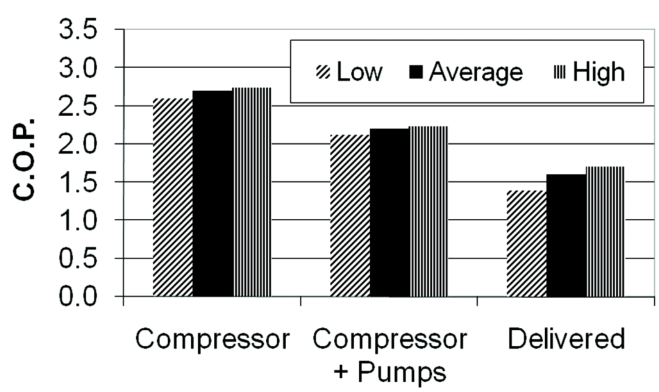

Figure 10 *Twenty-four-hour average system efficiencies.*

the simulated performance for similar ground temperatures, which yielded a COP of 3.33.

The COP of the compressor plus the pumps includes the electrical input of the circulator pumps in the denominator, in addition to the compressor, and ranges from 1.98 to 2.01. This efficiency is somewhat lower than expected due to short-cycling of the compressor, which allowed the circulator pumps to run more than the compressor. Calculation of this COP, using only the pumping energy that is coincident with the compressor energy, yields a COP range of 2.12–2.23, which is the value shown in the chart.

The delivered COP is calculated as the net delivered DHW energy divided by the electrical input of the compressor, pumps, and auxiliary water heater. Again, the performance was worse than expected due to the added runtime of the circulating pumps because of the short cycling of the heat pump. Correcting for the short cycling problem increases the COP range to 1.39–1.70, which are the values shown in the chart.

Figure 11 illustrates the energy balance of the hot water storage tank at three different daily loads. Even for the highest volume case, the heat pump provided 100% of the water heating—an end-use load of 9.70 kWh/day (33,300 Btu/day). Although this load is 19% below the DOE test standard of 12.030 kWh/day (41,045 Btu/day), it is important to remember that the load is based on a fixed volume of temperature-dependent, hot-water end uses at an incoming mains water temperature of 14.4°C (58°F). The actual average mains water temperature of 22.4°C (72.4°F) resulted in an end-use hot water load of 351 L/day (92.8 gpd). The system maintained an acceptable minimum delivery temperature of 45.8°C (114.4°F) using no auxiliary energy. System losses represented 3.01 kWh/day (10,300 Btu/day) or 24% of the energy input into the tank.

Figure 12 illustrates the energy balance of the heat pump. Heat pump source energy for the maximum daily hot water volume amounts to only 6.60 kWh/day (22,400 Btu/day), which represents a heat flux beneath the slab of only 2.7 W/m (0.77 Btu/hr/ft).

DISCUSSION

TRNSYS Simulation

Ground Models. The primary ground model used to simulate the heat transfer from the subslab piping loop was TRNSYS Type 706 (TESS 2003), which is intended to be used to model radiant slab-on-grade applications with insulation below the slab. Because the model is based on first-principal thermodynamics, it can also be used for applications such as this in which heat is removed from the earth from any one of the layers beneath the slab. However, the slab surface is not intended to be below grade, so the Kusuda correlation was used to approximate undisturbed earth temperatures at a depth of 2.5 meters, which were then used as input for the ambient sky near-field and far-field temperatures of the Type 706 radiant slab model.

Convective heat transfer of the soil at the plane of the slab surface is set to zero; the radiant heat transfer component is used to compensate for the difference in conductive heat transfer between earth-to-air and earth-to-earth. Because this approach does not involve the mass of soil above the plane of the basement slab, the near-field soil temperatures near the edge of the slab are likely to be higher in this scenario than in reality. This would tend to result in more optimistic simulated system performance.

The performance of the system using the remote earth-coupled piping loop was modeled in two different ways using two different TRNSYS types as a method of double-checking the output. Both simulations used the same pipe/soil/fluid properties, installation depth, and the Kusuda correlation for soil temperatures. The first model used Type 706d (TESS 2003) over a 20 × 20 m area to reduce thermal interactions between nearby sections of pipe. The second model used Type 952 (TESS 2006a), which models a single pipe with no interactions between sections of pipe. Type 952 does not allow for user input of pipe/soil interface conductivity.

Comparing the simulated performance using measured earth and mains water temperatures as inputs to the model, the measured heat pump performance is as much as 18% lower than the simulation would predict. Although increasing the resistance of the pipe/soil interface was found to bring the simulated efficiency in line with measured efficiency, it also

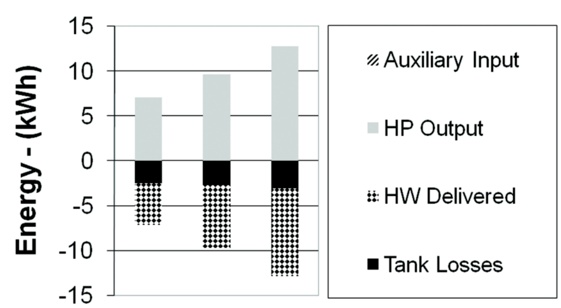

Figure 11 *Twenty-four-hour energy balance of hot water storage tank.*

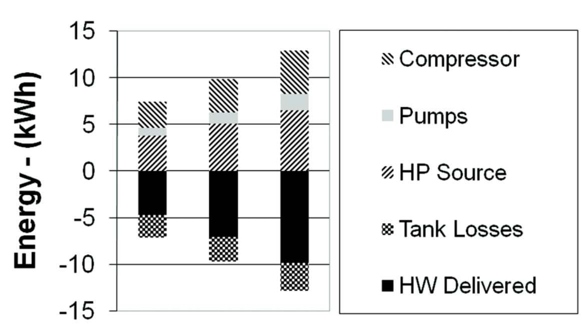

Figure 12 *Twenty-four-hour energy balance of the heat pump.*

produced unrealistically low source fluid temperatures. Other parameters, such as the size and conductivity of the hot water heat exchanger and fluid flow rates, were also changed, but they had little or no effect on the COP. This is probably because the manufacturer's performance data and flow factors are based on equal load and source fluid flows, and the measured flows are unequal and both somewhat different than the rated flows. Because the measured source flow is somewhat below the rated flow and the load flow is above, it is difficult to determine how much, if at all, the heat pump performance should be derated.

The only model parameter that did provide some ability to calibrate the model was the Rayleigh Number exponent of the hot water tank heat exchanger. It was discovered that decreasing this value from 0.25 to 0.11 helped bring the load-side fluid temperatures inline with measured temperatures, which in turn reduced the COP, bringing it closer to the measured COP.

Field Investigation

Heat Pump Operation. The overall system performance is less than expected due to compressor faults that allow the circulation pumps to operate when the compressor is shut off. The faults account for only 8% of the system runtime and have been mathematically removed from the calculations of system COPs. The faults are believed to be due to excessive refrigerant head pressure, even though the hot water setpoint is relatively low.

Parasitic Loads. Pumping energy reduces the system COP considerably. The pump power for the load fluid circulator is approximately 120 W and the source side is 172 W. It is estimated that pumping power could be reduced to 73 W if the subslab piping diameters were increased by one pipe size and the load side pump were eliminated altogether. A more market-ready system could have a direct expansion heat exchanger wrapped around the hot water tank, which would obviate the need for the load-side circulator pump. This would reduce the input energy and result in delivered COPs in the range of 1.59–1.94.

Slab Heat Flux. Due to the difficulty and expense of measuring the average heat flux of the entire floor slab, the heat flux of the floor was not measured.

Comparisons to Existing Technologies

Efficiency. The cold climate simulation system level COP of 2.4 and the field investigation potential system level COP of 2.2 are comparable with efficiencies of existing and past air-source HPWHs. A field study of 20 air-source HPWHs by Murphy and Tomlinson (2002) concluded that the average COP across all test locations was 2.0. In that study, there was no noticeable correlation to climate, and whole-house integrated efficiencies were not calculated. Simulated whole-house COPs in this study ranged from 2.0 in a cold climate to 2.9 in a warm climate.

CONCLUSIONS

TRNSYS Simulation

Although the integrated results indicate that this concept decreases soil temperatures beneath the slab by an average of 17.2°C (30.9°F) in a cold climate (Figure 7), the resulting soil temperatures still remain well above freezing, and the integrated whole-house efficiency is 17% better than a similar system using an earth-coupled piping loop remote from the house. The decrease in average monthly slab surface temperatures was less than 0.6°C (1°F) for all cases.

The following is a list of attributes relative to a conventional system with a remote earth-coupling:

- Eliminates need for excavation (major cost and construction barrier for earth-coupled systems)
- Up to 17% more efficient than conventional earth-coupled heat pump system
- 140% more efficient than conventional electric-resistance water heater
- Anti-freeze unnecessary (soil and fluid temperatures well above freezing)
- Extends equipment life (25% decrease in annual operating hours)
- Improves thermal comfort compared to typical uninsulated foundations (slab surface temperatures within 0.6°C [1°F] of room temperature)

Field Investigation

Heat pump efficiencies are less than expected but are acceptable for an all-electric system. Compressor control problems need to be addressed and piping pressure drops need to be reduced to increase system efficiencies to a point where the source energy consumption can be competitive with high-efficiency gas-fired systems. The current best estimate of the delivered COP is 1.94, while the target COP to be competitive with the source energy consumption of gas is 2.63, given site-source factors of 3.16 for electric and 1.02 for natural gas.

Operation of the system at an average daily hot water demand met the load without any use of auxiliary power or drop in delivered hot water temperature.

ACKNOWLEDGMENTS

The author would like to acknowledge the opportunity, efforts, and resources provided by Jeff Christian of Oak Ridge National Laboratory, Eric Helton of IBACOS, Inc., and the Loudon County Habitat in making this project a success.

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