Measured water heating performance of a vertical-bore water-to-water ground source heat pump (WW-GSHP) for domestic water heating over twelve months under simulated occupancy loads

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Abstract: This paper presents monthly performance metrics of a 5.275 kW (1.5 ton) WW-GSHP providing 227 L·day^{-1} domestic hot water at 49°C. Daily water use is simulated as stipulated in the Building America Research Benchmark Definition capturing the living habits of the average U.S household. The 94.5m vertical-bore ground loop is shared with a separate GSHP for space conditioning the 251m^2 residential home. Data on entering water temperatures, energy extracted from the ground loop, delivered energy, compressor electricity use, COP, WW-GSHP run times, and the impact of fan and pump energy consumption on efficiency are presented for each month. Factors influencing performance metrics are highlighted.

Key Words: ground source, heat pumps, water heating, renewable energy

1 INTRODUCTION

The practicality of using the ground as a resource of renewable thermal energy for domestic water heating throughout the year is described in this paper. The U.S consumed approximately 103 EJ in 2010, representing 19% of global consumption, ranking second after China in energy consumption (110.2 EJ) (DOE March 2012). Forty-one percent of U.S primary energy (42 EJ) was consumed by the buildings sector, compared to 30% by the industrial sector and 29% by the transportation sector. Of the 42 EJ consumed by the buildings sector, 54% was consumed in homes and the remaining 46% in commercial buildings. Seventy-five percent of the energy sources used by buildings came from fossil fuels, 16% from nuclear generation, and only 9% from renewable sources (DOE, *ibid*.). The top three end uses of energy in a residential building are space heating (45%), space cooling (9%), and water heating (18%). Although homes built between 2000 and 2005 are 14% more energy efficient per unit area than those built in the 1980s, and 40% more efficient than those built before 1950, the demand for larger homes have partially offset these efficiency improvements (DOE, *ibid*.). Water heating in buildings contributes 2 x 10^{11} kg of CO_2 to the atmosphere per year (DOE March 2012a). In its latest climate and energy outline until 2030, the EU Commissioners want a binding target to reduce carbon emissions by 40% from 1990

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levels. Renewable energy will need to provide 27% of EU energy by 2030, but while the target will be binding at the EU level, there will be no mandatory targets for member states (McGrath 2014). From an energy and environmental sustainability perspective, it is imperative that technologies be developed to mitigate anthropogenic impact on the environment and climate.

In this research, a 5.275 kW GSHP shared the same 94.5 m vertical-bore ground loop with another 7.15 kW GSHP used for space conditioning described in a separate paper in the same IEA conference (Ally, Munk et al. 2014). Since vertical-bores such as the one used in this study can extract thermal energy at relatively stable temperatures throughout the year, it makes sense to use ground source heat pumps for water heating in buildings. Much work (Neksa, Rekstad et al. 1998; Cecchinato, Corradi et al. 2005; Nanxi, Shi et al. 2005; Stene 2005; Hjerkinn 2007; Stene and Hjerkinn 2008; Minetto 2011) has been done in recent years on the use of various refrigerants for ground source heat pumps, but this study focuses on the efficiency of year-round water heating and quantifying the amount of thermal energy that can be extracted from the ground loop for domestic hot water production. The research house is located in Oak Ridge, Tennessee, 36° 0' 37" N / 84° 16' 11" W in Climate Zone 4.

2 VERTICAL-BORE GROUND LOOP AND EQUIPMENT

The vertical-bore consists of a 0.019m internal diameter high density polyethylene (HDPE) tube inserted into the ground to a depth of 94.5 m, giving a total length of 189 m and a total surface area of 11.3 m^2. The "brine" solution in the ground loop consists of 20% (v/v) of propylene glycol and water 80% (v/v) to exchange energy with the ground and transfer it to the refrigerant via the ground loop (GL) heat exchanger as shown in Figure 1. A 5.275 kW water-to-water ground source heat pump (WW-GSHP) rated at a COP of 3.1 was installed in the research house (Ally, Munk et al. 2014) to provide on an average of 227 $L \cdot day^{-1}$ hot water at 49°C for domestic use. The water use pattern for showers, laundry, and dishwashing constructed from (Hendron and Engebrecht 2010) is tabulated in Table 1. The dishwasher averages 15.3 L of hot water per wash cycle, running four cycles per week, whereas, the clothes washer averages 13.9 L of hot water per cycle, running 6 cycles per week. The WW-GSHP serviced a 303 L hot water tank via the domestic hot water (DHW) heat exchanger (see Figure 1).

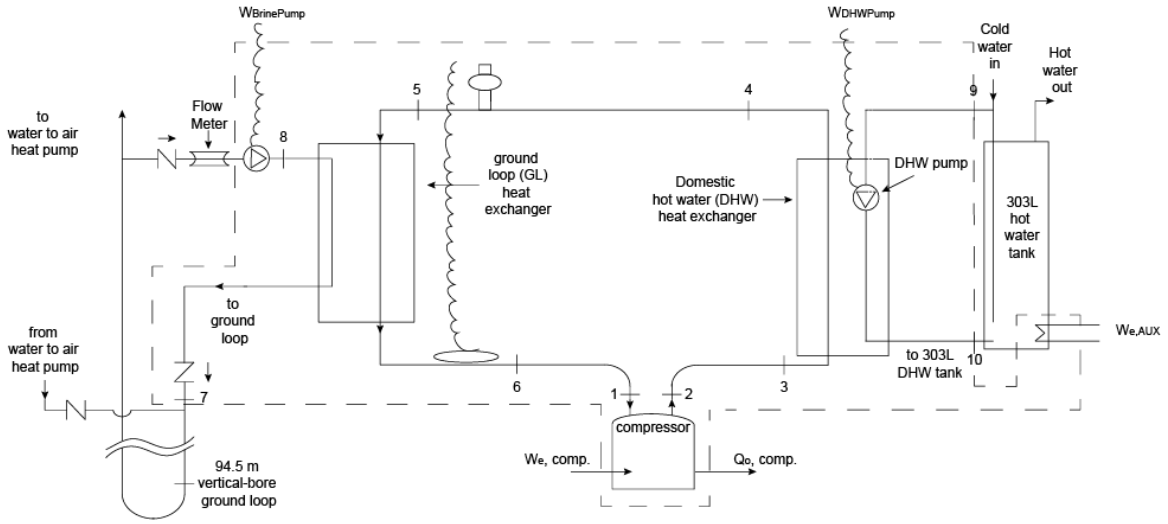


Figure 1: Equipment for water heating connected to a vertical-bore ground loop that also serviced a GSHP for space conditioning (not shown). The vertical-bore is 94.5 m deep, 0.019m in diameter with a surface area of 11.3 m^2.

The hot water tank was a seamless, blow-molded, polybutylene tank with a 4.5kW upper and lower heating elements for back up energy in case the WW-GSHP function was arrested due to a technical fault. The tank was rated at an energy factor of 0.92 when operated in the conventional electric resistance mode.

3 SENSORS AND DATA COLLECTION

The schematic of the equipment and the state points where measurements were taken are represented in Figure 1. Data was recorded every 30 seconds, averaged over 15 minute periods, and stored before exporting it to a spreadsheet for post-processing and analysis. Referring to Figure 1, the compressor suction line (1) and discharge (2) pressures, were measured with pressure transducers with an accuracy of ±0.25% of full scale (FS). The leaving (7) and entering (8) water temperatures in the ground loop were measured with immersion thermocouples. All other temperatures on the refrigerant and water lines were measured with surface mounted thermistors that were first coated with conductive paste, tightly attached to their respective connecting lines with electrical tape, insulated with cork and taped once again for good adherence. The thermistors had a 15 s time constant and an accuracy of ± 0.2°C. The brine flow rate was measured with a turbine meter whose accuracy was ±2% over the flow range. The DHW flow rate was measured via a differential pressure measurement with ±0.25% FS accuracy. Electrical power to the compressor, brine and DHW pumps were measured with instrumentation-grade transducers with split-core current transformers having a minimum accuracy of ±0.45% of reading plus 0.05% of FS. Pressure drops in connecting lines were 1-1.5% of the upstream pressure, (depending on line length and configuration) as provided by the equipment manufacturer. Thermodynamic state points were evaluated using the National Institute of Standards and Technology (NIST) Reference Fluid thermodynamic and Transport Properties Database (REFPROP) Version 8.0 (NIST). Uncertainty in the estimated COPs is estimated to be ± 0.254. The monthly COPs are weighted average values based on compressor run times for the particular month.

4 ANALYSIS

Analysis is based on the assumption of quasi steady state conditions in the refrigerant loop justified by observing that the thermodynamic state points reach stable values typically within 5 minutes after the compressor switches on. The time to reach steady state in the hot water lines (locations 9 and 10) in Figure 1 varies between 1-4.5 minutes depending on the temperature of the water in the tank at the start of water flow to the DHW heat exchanger. Since compressor run times are longer than 4 minutes, quasi steady state conditions are valid.

4.1 Mass and energy balances

Under quasi steady state conditions, the mass balances are trivial, with inflows equal to outflows. The energy balance equation in component form is given by Eq. (1)

$$W_{\text{Brine Pump}} + W_{\text{Comp.}} + W_{\text{Controls}} + W_{\text{e,Aux.}} + W_{\text{DHW Pump}} + Q_{\text{GL HX}} - (Q_{\text{HWTank}} + Q_{\text{Comp.}}) = 0 \quad (1)$$

Where $Q_{\text{GL HX}}$ is the heat extracted from the vertical ground loop; $W_{\text{Brine Pump}}$ is the electrical energy supplied to the brine pump; $W_{\text{Comp.}}$ is the electrical work to the compressor; W_{Controls} is the electrical energy consumed by the control board; $Q_{\text{HW Tank}}$ is the thermal energy delivered to the water tank; and $Q_{\text{Comp.}}$ is the thermal energy rejected by the compressor to the surroundings (conditioned space). The monthly energy inputs to the WW-GSHP for producing domestic hot water are shown in Table 2. Similarly, the total monthly energy outputs are shown in Table 3. The last columns of Table 1and Table 2 show reasonably good satisfaction of the energy balance given by Eq. (1) with the percent difference, Δ%

tabulated in Table 4 ranging from 4.50% (minimum in August) to 5.66% (maximum in December).

Table 1: Schedule and duration of use of water under simulated occupancy conditions in the research house

Shower Schedule		**Clothes Washer Schedule**			**Dishwasher Schedule**		
Start times (24 h basis)	**Daily Water use**	**Start times (24 h basis)**	**Days**	**Daily Water use**	**Start times (24 h basis)**	**Days**	**Daily Water use**
	L/day			**L/day**			**L/day**
7:00:00	76	7:30:00	Th	13.9	19:30:00	Mon	15.3
8:30:00	19	17:15:00	Th	13.9	19:30:00	Tues	15.3
12:00:00	19	8:00:00	Tues/ Wed	13.9 /13.9	19:30:00	Wed	15.3
17:00:00	38	10:00:00	Tues/ Wed	13.9 /13.9	19:30:00	Th	15.3
21:00:00	76						
TOTAL	228			83.4			61.2

Table 2: Measured energy inputs to the WW-GSHP for water heating

Month (Yr. 2012)	**$W_{Brine\ Pump}$**	**$W_{Comp.}$**	**$W_{Controls}$**	**$W_{e,Aux.}$**	**$W_{DHW\ Pump}$**	**$Q_{GL\ HX}$**	**Total Energy In, E_{in}**
	(kWh)	**(kWh)**	**(kWh)**	**(kWh)**	**(kWh)**	**(kWh)**	**(kWh)**
Jan	12.01	121.54	1.984	0	4.51	298.84	438.87
Feb	11.13	112.13	1.832	0	4.16	295.31	425.56
Mar	9.81	100.99	1.665	0	3.63	290.86	406.95
Apr	8.31	86.51	1.523	0	3.07	256.37	355.78
May	7.57	81.56	1.377	0	2.84	261.86	355.21
Jun	6.25	69.94	1.224	0	2.41	233.12	312.95
Jul	5.49	66.12	1.191	0	2.28	232.60	317.68
Aug	6.20	71.57	1.273	0	2.48	244.37	325.90
Sept	6.46	70.98	1.216	0	2.47	234.51	315.65
Oct	8.20	85.42	1.479	0	3.03	253.27	351.39
Nov	9.33	96.29	1.596	0	3.48	257.87	368.57
Dec	10.34	105.82	1.734	0.08	3.86	272.83	394.67

4.2 Water heating coefficient of performance and EWT

The coefficient of performance for the WW-GSHP is,

$$COP_{WW\text{-}GSHP} = \frac{Q_{HW\ Tank}}{W_{Brine\ Pump} + W_{DHW\ Pump} + W_{Comp.}} \quad (2)$$

Since $COP_{WW\text{-}GSHP}$ is primarily dependent on the entering water (brine) temperature (EWT), it is useful to know the monthly variation in EWT, the corresponding coefficient of performance, and also the leaving water (brine) temperature to gauge the difference between EWT and LWT, typically within 3°-5°C.

Table 3: Measured Energy Output from WW-GSHP

Month (Yr. 2012)	$Q_{HW\ Tank}$	$Q_{Comp.}$	Total Energy Out, E_{out}
	(kWh)	(kWh)	(kWh)
Jan	-376.14	-39.25	-415.39
Feb	-371.66	-29.35	-401.00
Mar	-355.97	-28.57	-384.54
Apr	-311.12	-26.24	-337.36
May	-311.35	-26.90	-338.25
Jun	-273.86	-24.71	-298.57
Jul	-268.16	-25.39	-293.55
Aug	-284.65	-26.60	-311.24
Sept	-276.53	-24.41	-300.94
Oct	-304.50	-28.77	-333.27
Nov	-316.56	-31.95	-348.51
Dec	-337.46	-34.88	-372.34

Table 4: Monthly EWT, LWT, coefficient of Performance, and the agreement in energy balance

Month (Yr. 2012)	EWT	LWT	Energy In, E_{in}	Energy Out, E_{out}	$\Delta\% = 100x(E_{in}-\lvert E_{out}\rvert)/E_{in}$	$COP_{WW\text{-}GSHP}$
	(°C)	(°C)	(kWh)	(kWh)	(%)	
Jan	7.43	4.24	438.87	-415.39	5.35	2.69
Feb	5.05	8.12	425.56	-401.00	5.55	2.88
Mar	12.61	9.50	406.95	-384.54	5.51	3.07
Apr	14.36	11.27	355.78	-337.36	5.18	3.13
May	19.09	15.40	355.21	-338.25	4.77	3.34
Jun	22.58	17.85	312.95	-298.57	4.59	3.43
Jul	28.03	19.64	317.68	-293.55	4.59	3.57
Aug	25.14	19.14	325.90	-311.24	4.50	3.49
Sept	21.70	17.27	315.65	-300.94	4.66	3.41
Oct	15.33	12.20	351.39	-333.27	5.16	3.10
Nov	10.66	7.40	368.57	-348.51	5.44	2.86
Dec	9.03	5.83	394.67	-372.34	5.66	2.77

4.3 EWT and compressor run times

The profile of compressor run times shows that it runs for the longest time during the heating season when the entering water temperatures are low. Lowest run times are observed during peak cooling months of July/ August, when EWTs are higher than during the heating season. In January/February, the compressor ran approximately 45% longer than in June/July because the EWT is much lower in the winter months than it is in summer. The inverse relationship between EWT and compressor run times is graphically shown in Figure 2.

Table 5: Monthly EWT and compressor run times

Month (Yr. 2012)	EWT	Compressor run time
	(°C)	(h)
Jan	7.43	90.29
Feb	5.05	83.11
Mar	12.61	73.20
Apr	14.36	62.13
May	19.09	58.06
Jun	22.58	49.40
Jul	28.03	46.40
Aug	25.14	50.50
Sept	21.70	50.38
Oct	15.33	61.46
Nov	10.66	70.67
Dec	9.03	78.35

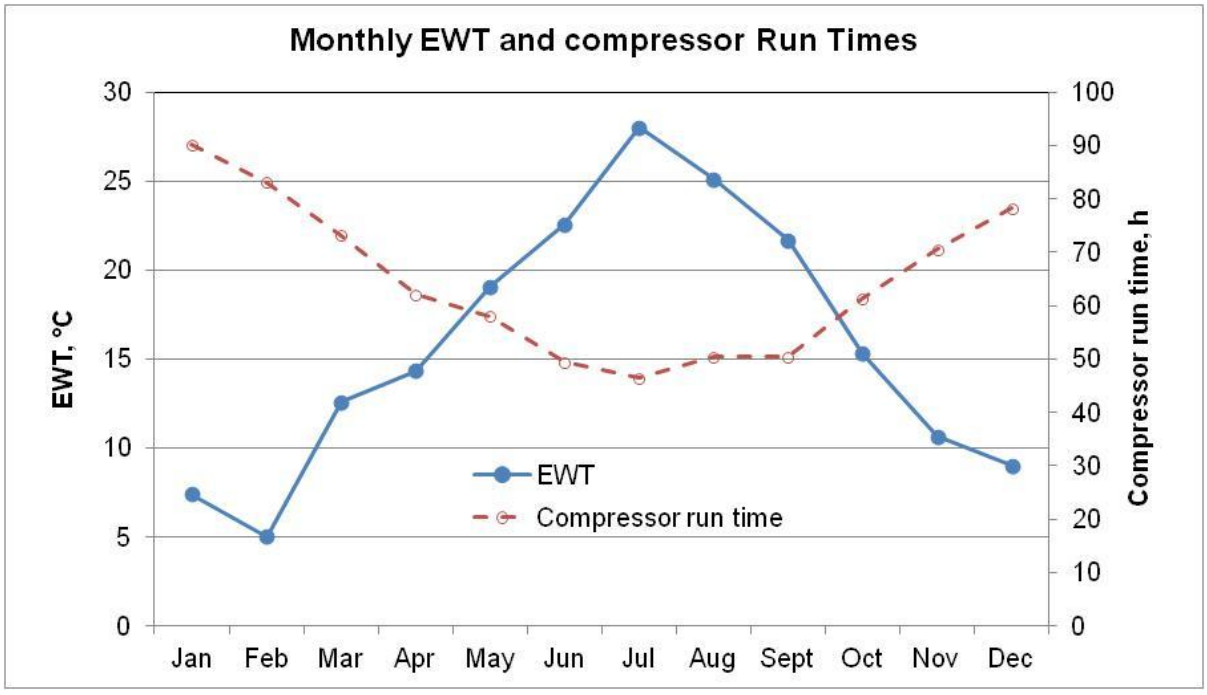


Figure 2: Monthly EWT and compressor run times are inversely related

Since the GSHP for space conditioning and the WW-GSHP for water heating share the same vertical loop, water heating is somewhat assisted by the GSHP in summer because heat rejection to the brine elevates EWT, making more energy available for water heating than would otherwise be the case. In winter, the GSHP and the WW-GSHP both compete for the same energy from the ground only when space conditioning and water heating demand is simultaneous. However, GSHP and the WW-GSHP do not always work simultaneously. Therefore, the benefit and the penalty on EWT due to the GSHP are not as distinct as may appear *prima facie*. In summer, the GSHP and the WW-GSHP complement each other, whereas in winter they act adversely to each other. For shared loops, it may be instructive to design the vertical bore on the basis of winter loads for space heating plus water heating.

4.4 Energy extracted from the ground loop

Energy extracted from the ground loop, $Q_{GL\ HX}$ and compressor run times are quantified in Table 1 and Table 5, respectively and plotted in Figure 3 to show that not only are compressor run times lower during peak summer months, but that the amount of thermal energy required from the ground troughs in July/August because of elevated entering water temperatures (see Figure 3). In colder months, the compressor run time is almost double relative to the summer trough because EWTs are lower. Despite the monthly variation in EWTs (Figure 2), energy demand on the ground loop for combined space conditioning and for water heating remained in the range of 230 kWh to 300kWh for this particular research house.

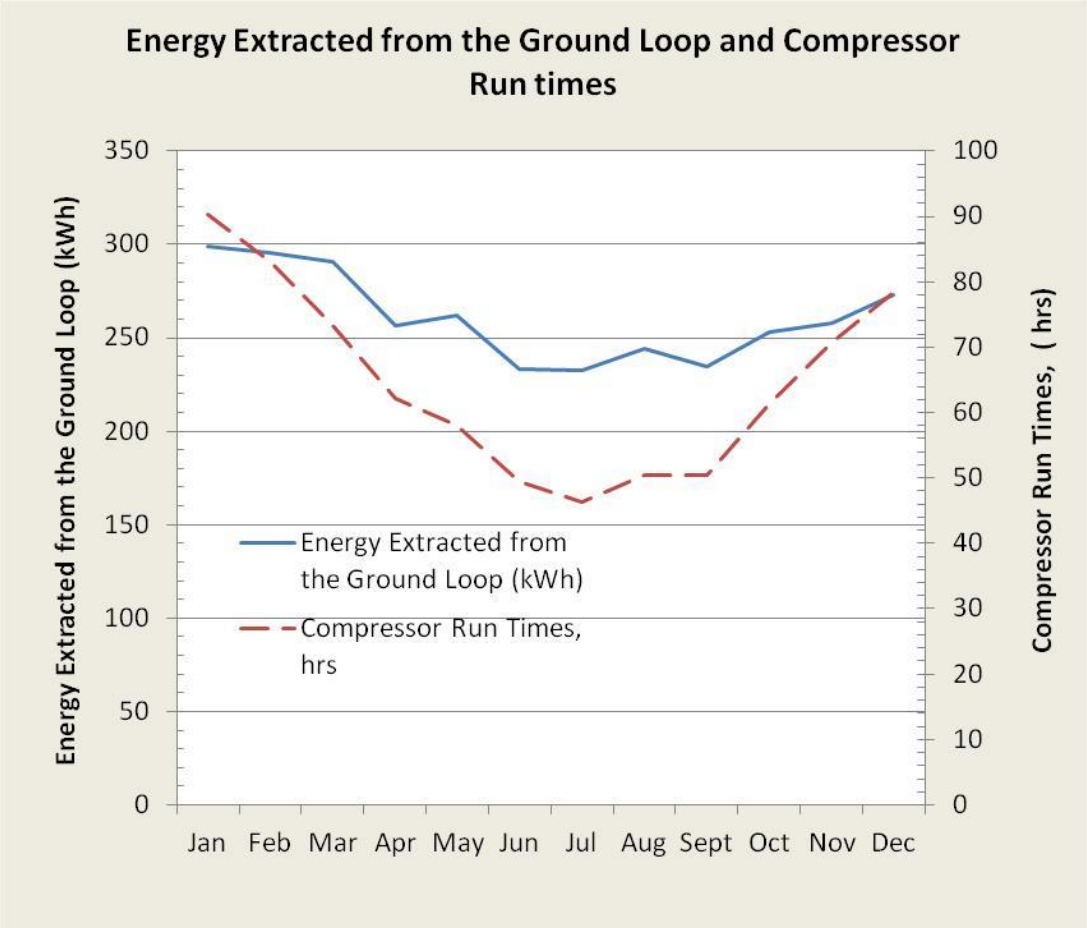


Figure 3: Energy extracted from the ground loop and compressor run times.

4.5 Energy from the ground loop dominates total energy required for water heating

Since the focus is to develop and to evaluate technologies that increasingly use renewable sources of energy and less electrical energy to service the utility load (water heating), it is incumbent to show what fraction of the total input energy came from the ground loop, and what fraction is consumed in the form of electricity by the compressor. These fractional metrics, derived from the data in Table 2 and plotted in Figure 4 show that the ground loop provided at least 68% of the total energy input in winter, and a maximum of 76% in peak summer months.

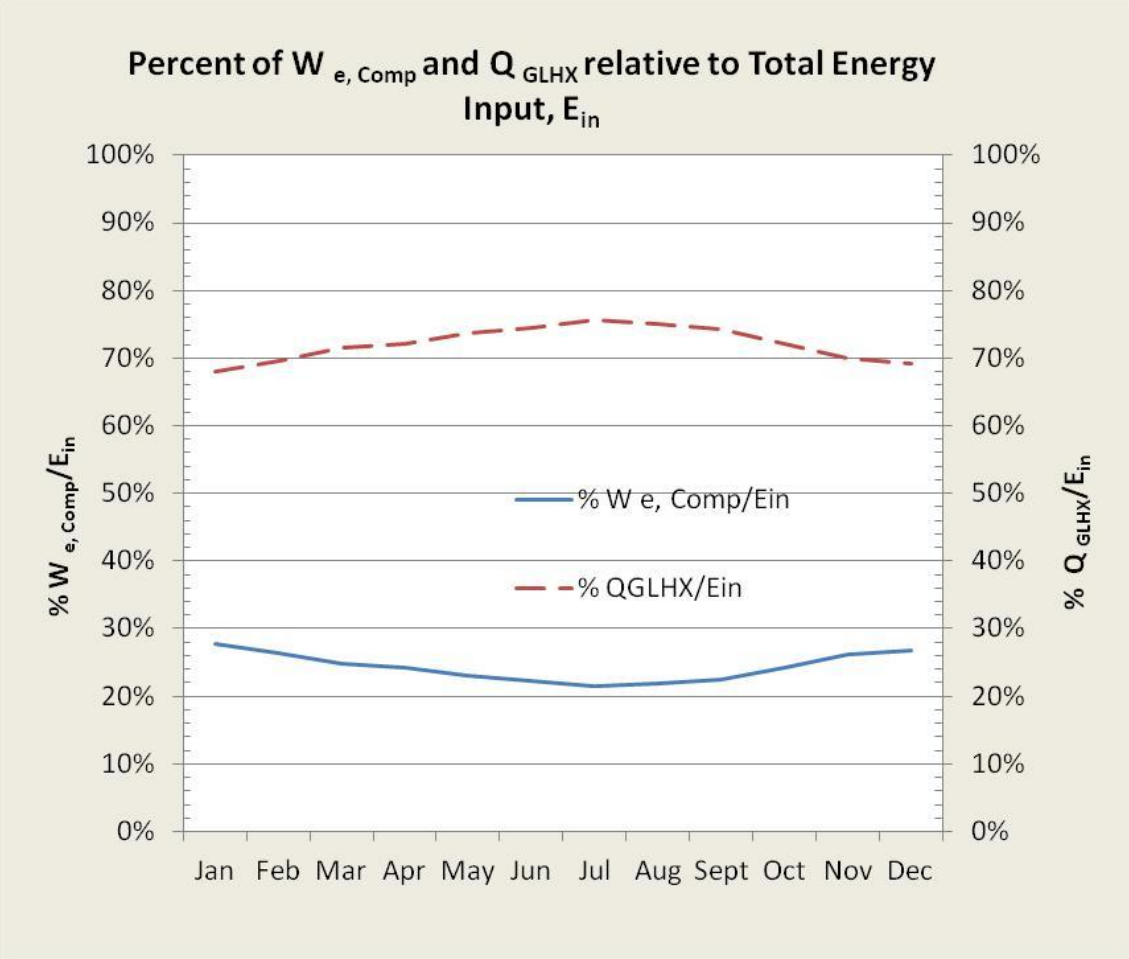


Figure 4: Percent of total input energy contributed by the ground loop and used by the compressor in the form of electricity.

Between 21% and 28% of the total energy input in the form of electricity was consumed by the compressor. As more energy was extracted from the ground loop in going from winter to summer, the compressor used less electrical energy. The reverse was true in going from summer to winter, but these variations remain small, within 10% of the annual range. Similar results were reported for shallow ground loops in residential construction (Ally, Munk et al. 2013).

5 CONCLUSIONS

- A 5.275 kW WW-GSHP was sufficient to service a 303 L hot water tank providing 227 $L \cdot day^{-1}$ hot water at 49°C to a research house operated under simulated occupancy loads. Hot water use patterns (Table 1) were developed under the Building America protocol that captures the living habits of US households.
- Essentially no auxiliary (resistance) heat was used over the 12-month study period (Table 2).
- The ground loop provided between 68% and 76% of the total energy input to the WW-GSHP (Figure 4).
- The compressor run times varied between a maximum of 90 h in January to a minimum of 46 h in July. Higher run times are needed in the winter than in the summer due to colder EWTs, respectively (Figure 2).
- The maximum COP was 3.57 (July) and the minimum COP was 2.69 (January). The COP values include all sources of electrical energy inputs as given by Eq. (2).
- Since 68%-76% of the total input energy was extracted from the ground loop, there is a valid case in favor of WW-GSHPs to be a practical and viable technology for reducing primary energy consumption for making domestic hot water. This is an enabling technology for curbing greenhouse gases emissions, and achieving the goals of the IECC and EU programs to combat climate change.

6 ACKNOWLEDGEMENTS

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