MEASURED SPACE CONDITIONING PERFORMANCE OFA VERTICAL-BORE GROUND SOURCE HEAT PUMP (GSHP) OVER TWELVE MONTHS UNDER SIMULATED OCCUPANCY LOADS

Moonis, Ally, Senior Research Staff, Oak Ridge National Laboratory, 1 Bethel Valley Road (Oak Ridge, TN, USA);
Jeffrey, Munk, Research Staff, Oak Ridge National Laboratory, 1 Bethel Valley Road (Oak Ridge, TN, USA);
Van, Baxter, Senior Research Staff, Oak Ridge National Laboratory, 1 Bethel Valley Road (Oak Ridge, TN, USA);
Anthony, Gehl, Research Staff, Oak Ridge National Laboratory, 1 Bethel Valley Road (Oak Ridge, TN, USA);

Abstract: This paper presents monthly performance metrics of a 7.56 kW (2.16 ton) GSHP serving the space conditioning loads of a 251m^2 (2700ft^2) residential home with a phase change material in its envelope, and a single vertical-bore 94.5m (310 ft) ground loop. The same ground loop also serviced a ground source heat pump water heater. Envelope characteristics are discussed briefly in the context of reducing thermal losses. Data on entering water temperatures, energy extracted from the ground, energy delivered/removed, compressor electricity use, COP, GSHP run times (low and high compressor stages), and the impact of fan and pump energy consumption on efficiency are presented for each month. Both practical as well as research and development issues are discussed. The findings suggest that GSHPs represent a practical technology option to reduce source energy reduction and greenhouse emissions under the IECC 2012 Standard, as well as the European Union (EU) 2020 target of generating over 25% of heat consumed in the EU from renewable energy.

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

1 INTRODUCTION

Nearly 5 million commercial buildings and 115 million residential households consumed nearly 40%, (11 x 10^{12} kWh) of total U.S energy demand in 2012, of which 6 x 10^{12} kWh was consumed in the residential sector, alone (DOE 2010; DOE 2012; EIA 2013). The EU countries show a similar trend where buildings represent approximately 40% of the total energy consumption and carbon production (Baden, Waide et al. 2006). Residential and commercial buildings account for approximately 33% of all energy related greenhouse gas emissions worldwide (UNEP 2007).

[1]1 450 MW as a typical power plant size was based on the range in size of power plants installed in 2006. Stock Generating Capacities, by Plant Type." http://buildingsdatabook.eere.energy.gov/TableView.aspx?table=6.2.7

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Unequivocally, the buildings sector offers the largest low-cost potential in all world regions for mitigating emissions (IPCC 2007) and energy consumption. The U.S DOE has established goals of 50% energy efficiency improvements compared to the 2006 IECC Standard (Taylor and Lucas 2010). Recent research (Belzer, Halverson et al. 2010) shows that energy consumption may be reduced by 10^3 TWh per year by 2030, the equivalent of 260 medium (450 MW) power plants[1]. In Europe alone, 7100 TWh or 610 Mtoe annually is emitted from land surfaces of which 260 Mtoe per year are produced under the surface of the EU 27, signally a huge potential resource (Dumas and Landolina 2012). The European Technology Platform on Renewable Heating and Cooling (RHC) thinks that shallow geothermal energy can be used by GSHPs almost everywhere in Europe without any geographical restrictions. At the present time, the ground is used for heating and for cooling residential (~5kW), commercial (30-500 kW), and larger structures greater than 500 kW, and for district heating in Europe. According to the RHC, the program is intentionally targeting the existing housing infrastructure which represents the largest share of the low temperature energy demand. RHC's current benchmark studies indicate that geothermal technology is probably the most effective option for this market in terms of carbon footprint and economics (Dumas and Landolina 2012). Clearly, there is a pressing need for technologies that utilize more renewable and less fossil-based energy resources to meet the higher efficiency and energy savings targets.

The goal of this research was to establish the 12-month performance of two discrete GSHPs – one for space conditioning and the other for water heating, both connected to the same vertical-bore ground loop. The internal loads were simulated according to the Building America Benchmark (Hendron and Engebrecht 2010) that captures the living habits of the average U.S household. Energy consumption for space heating loads is dependent on the envelope, equipment efficiency, and behavioral factors. A complementary paper discusses the GSHP for domestic water heating as part of the same IEA conference. The research house is located in Oak Ridge, Tennessee, 36° 0' 37" N / 84° 16' 11" W in Climate Zone 4 (Mixed-Humid).

2 ENVELOPE, GSHP EQUIPMENT AND VERTICAL-BORE

The research house is a two-storey structure with a conventional attic and crawl space and with a total conditioned space of 251m^2. Framing consists of a double-wall assembly with interior insulation acting as a thermal buffer against heat absorbed by a phase change material (PCM) in the exterior wall. The function of the PCM is to modulate the flow of thermal energy that would otherwise occur. The house roof is an infrared reflective (IRR) painted metal shake with a solar reflectance of 0.34 and measured thermal emittance of 0.85. The attic has an oriented stranded board (OSB) deck overlaid with a micro-perforated aluminium foil facing into the attic. Solar powered gable ventilators are installed on the interior of the attic gables to enhance attic ventilation. At solar noon with clear sky the fans will induce about 10 air changes per hour from the perforated fiber cement soffit panels and the gable vents. Total soffit and gable-end vent area exceeds the 1:150-code (Miller, Karagiozis et al. 2010). PCM materials added to the blown fiber insulation on the attic floor absorbs the remaining heat that escapes the reflective metal shake roof, the radiant barrier, and the solar powered attic ventilation.

Blower door tests yielded 2.2 air changes per hour (ACH) at 50Pa. The building has a Home Energy Rating System (HERS) Index of 51. In comparison, an EPA Energy Star home would have a HERS score of 85, and a Net Zero Energy Home would have a HERS Index of 0 while a house built to satisfy the IECC 2006 code would have 5.7 ACH@50Pa and a HERS rating of 93 (Miller, Karagiozis et al. 2010). The thermostat set point temperatures are 21.7

°C for heating and 24.4 °C for cooling and were maintained within 1 °C in each of two zones (top floor and living space). The roof assembly has an overall thermal resistance of about R_{SI}-8.8 (R_{US}-50). The walls are 0.13m thick and have thermal resistance of R_{SI}-5.28 (R_{US}-30). Windows are triple pane with a U-factor of 1.25 W/m2·K and a solar heat gain coefficient (SHGC) of 0.17.

A 7.16 kW GSHP with a variable speed blower was connected to a 94.5 m vertical bore containing a water-propylene glycol mixture of 80% water by volume. The GSHP was rated as cooling[2] and heating[3] COP of 5.4 and 4.0, respectively.

The vertical bore ground loop consisted of one U-tube 1.9 cm ID made of high density polyethylene (HDPE) with a total length of 94.5 x 2 m. A separate 5.3 kW water-to-water GSHP connected to the same loop serviced a 303 L water tank providing approximately 220 L/day hot water at 49°C for domestic use. This paper describes the space conditioning performance of the 7.15 kW GSHP. The water heating performance is discussed in a separate paper.

2 SENSORS AND DATA ACQUISITION

Compressor, indoor fan, electronic control, pump, and total power consumed by the GSHP were measured with watt/watt-hour transducers split-core current transformers (CT's) having a minimum accuracy of ± 0.45% of reading plus 0.05% of full scale (FS). Compressor suction and discharge pressures were measured with a pressure transducer with ±0.25% accuracy. Brine flow rate through the space conditioning heat pump was measured by a differential pressure measurement across the ground loop (GL) heat exchanger (brine side) to an accuracy of ±0.25% FS, complemented with a detailed pressure drop versus flow calibration from the factory. Pressure drop in connecting lines were estimated at 1-1.5% of the upstream pressure, based on equipment manufacturer's data. The pressure drop across the expansion valve included the pressure drop in the connecting lines between the indoor coil and the GL heat exchanger. The schematic of the equipment is shown in Figure 1 with the system boundary for mass and energy balances.

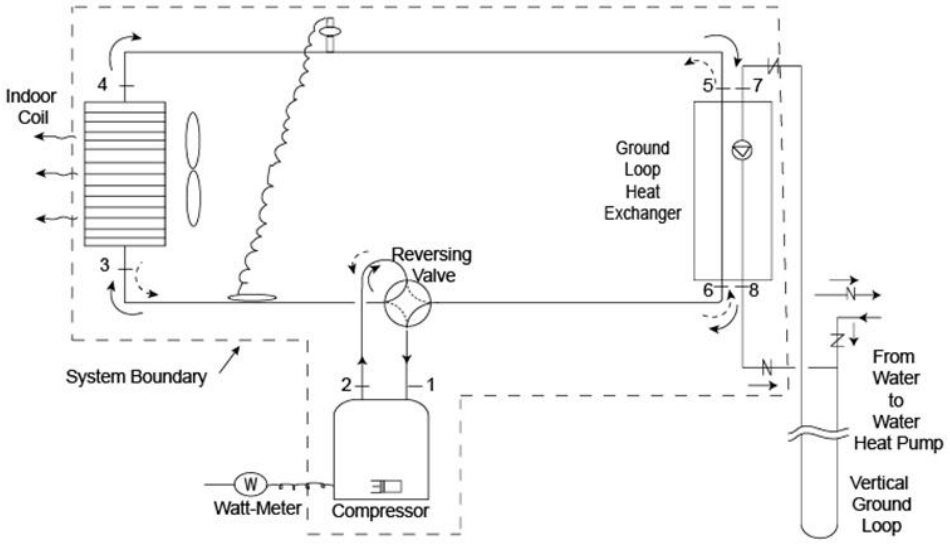


Figure 1: Schematic of GSHP for space conditioning connected to a vertical -bore ground loop. Numbered items indicate state points.

[2] Full Load Cooling based upon 27°C DB, 19°C WB entering air and EWT of 25°C
[3] Full Load Heating based upon 20°C DB, 15°C WB entering air temperature and EWT of -1.1°C

A data logger collected data every 15 seconds and was programmed to average the collected data over a 15 minute interval and download to a spreadsheet for post processing. 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).

3 ANALYSIS

Mass and energy balances are under the assumption of steady-state steady flow (SSSF) conditions justified on the basis that except during compressor start-up and shut down, or when cycling times are less than 5 minutes (rare occurrence), SSSF conditions prevail. Further, due to the sizing of the GSHP (7.56 kW) for the 251 m^2 house, we observe long compressor run times, an important factor in keeping cycling losses low. The expansion valve is assumed to be adiabatic. Compressor heat is rejected to the ambient (conditioned space) set at 24.4°C during the cooling season and at 21.7°C during the heating season by the thermostat whose accuracy is ±1°C.

3.1 Energy balance

For each respective month, the aggregate energy balance given by Eq. (1) is obtained by invoking the SSSF equation (Wylen, Sonntag et al. 1994). The energy balance equation in component form thus becomes,

$$\sum W_{Comp.} + W_{Fan+Controls} + W_{Pump} + W_{e,Aux} + Q_{Indoor\,Coil} + Q_{GL\,HX} + Q_{Comp.} + Q_{e,Aux.} = 0 \quad (1)$$

On this basis of Eq. (1), the total measured component-wise energy input to the GSHP for each month is shown in Table 1

Table 1: Energy input aggregate for each month

(yr. 2012)	$W_{Comp.}$	$W_{Fan+Controls}$	W_{Pump}	$Q_{GL\,HX}$	$Q_{Indoor\,Coil}$	$W_{e,Aux.}$	Total E_{In}
	(kWh)	(kWh)	(kWh)	(kWh)	(kWh)	(kWh)	(kWh)
Jan	317.9	60.3	50.9	1204.3		0.003	1633.4
Feb	217.0	41.5	34.8	823.6		0.002	1116.8
Mar (heat)	39.4	10.7	6.3	147.0		0.000	203.4
March (Cool)	50.9	17.0	10.5		355.1		433.5
Apr (cool)	41.2	9.2	8.7		314.4		373.5
May	189.8	37.1	36.1		1271.9		1534.9
Jun	275.7	49.7	48.1		1668.6		2042.1
July	452.8	74.0	71.6		2435.2		3033.6
Aug	303.3	52.8	51.7		1775.9		2183.7
Sept	165.2	30.2	29.3		1027.4		1252.1
Oct (heat)	35.8	6.0	5.7	133.4		0.095	181.0
Oct (cool)	12.2	4.1	2.4		73.7		92.4
Nov	199.1	64.5	30.0	808.3		0.002	1101.9
Dec	268.3	50.2	43.2	1037.7		0.001	1399.4

Similarly, the total measured component-wise energy output to the GSHP for each month is shown in Table 2.

Table 2: Energy output aggregate for each month

(yr. 2012)	$Q_{Indoor\ Coil}$	$W_{Fan+Controls}$	$Q_{Comp.}$	$Q_{e,Aux.}$	$Q_{GL\ HX}$	Total E_{Out}
	(kWh)	(kWh)	(kWh)	(kWh)	(kWh)	(kWh)
Jan	-1446.3	-60.3	-63.9	-0.003		-1570.6
Feb	-983.8	-41.5	-48.5	-0.002		-1073.8
Mar (heat)	-173.7	-10.7	-11.0	0.000		-195.4
March (Cool)		-17.0	-9.4		-396.7	-423.1
Apr (cool)		-9.2	-5.5		-350.0	-364.7
May		-37.1	-21.4		-1440.4	-1498.9
Jun		-49.7	-33.6		-1910.6	-1993.9
July		-74.0	-52.8		-2817.1	-2943.9
Aug		-52.8	-36.3		-2042.6	-2131.8
Sept		-30.2	-22.7		-1169.7	-1222.6
Oct (heat)	-156.8	-6.0	-11.3	-0.095		-174.1
Oct (cool)		-4.1	-3.4		-82.5	-90.0
Nov	-960.2	-64.5	-39.3	-0.002		-1063.9
Dec	-1246.2	-50.2	-49.2	-0.001		-1345.6

From Table 1 and Table 2, the goodness of the energy balances for each month is within 2-4% of the measured values.

A valid case for using GSHPs for space heating residential structures may be made by considering the percent of the energy extracted from the ground loop to the total energy needed to satisfy space heating loads. Table 3 clearly shows that over 75% of the space heating demand is derived from the ground loop. Since the GSHP shares the same ground loop as the WW-GSHP, the former is aided by the latter during the summer and penalized during the winter, when both pieces of equipment are competing for thermal energy extracted from the ground. Because the run time for the GSHP is between three to four times longer than it is for the WW-GSHP, the GSHP is the primary beneficiary of energy extracted from the ground during winter. Although a major fraction of the energy extracted from the ground during winter goes to heat the home (based on the GSHP run time and capacity compared to that of the WW-GSHP) (Ally, Munk et al. 2014), it is not possible to quantify the exact amount without knowing the overlap in the run times and the separation between run times of the GSHP and the WW-GSHP. But we can say with confidence that even in the depth of winter months, with competing demands for water heating, over 75% of the energy for space conditioning was derived from the ground loop.The total electrical energy used, $W_{e,\ Total} = W_{Comp.} + W_{Fan+Controls} + W_{Pump}$ is obtained directly from Table 1.

3.2 Heating and cooling capacity

The calculated GSHP heating and cooling capacities as a function of the entering water temperature (EWT) are shown in Figure 2. Also plotted on the same figure is the manufacturer's design point capacity. The design heating capacity of 4.5kW is rated at 0°C EWT, and the design cooling capacity of 6.18 kW is rated at 25°C EWT. Although our field

data when extrapolated to an EWT of 0°C agreed well with the manufacturer's heating design point rating, our cooling capacity fell short of what the manufacturer had specified. During summer (cooling season), the WW-GSHP helped mitigate an undersized vertical-bore heat exchanger by being an additional receiver of thermal energy rejected from the space cooling cycle. This explanation is supported by an analysis of the ambient and entering water temperatures as shown in Figure 3. We see that during peak summer months, the EWT lies above the outside air temperature OAT) trend line, signaling higher than expected EWTs. The present size of the vertical-bore apparently did not allow sufficient heat to be transferred to the ground and therefore, the EWT during the cooling season remained more elevated than otherwise would have been the case.

Table 3: Per cent of total space heating load obtained from the ground loop

(yr. 2012)	**Total Energy Delivered**	**Total Electrical Energy Used, $W_{e,Total}$**	**Energy Extracted from the Ground Loop, $Q_{GL\ HX}$**	**$\frac{W_{e,\ Total}}{W_{GL\ HX}}$**	**% Delivered Energy from Ground Loop**
	(kWh)	**(kWh)**	**(kWh)**		
Jan	-1570.6	429.1	1204.3	0.356	76.7
Feb	-1073.8	293.2	823.6	0.356	76.7
Mar (heat)	-195.4	56.4	147.0	0.384	75.2
March (Cool)					
Apr (cool)					
May					
Jun					
July					
Aug					
Sept					
Oct (heat)	-174.1	133.4	133.4	0.357	76.6
Oct (cool)					
Nov	-1063.9	808.3	808.3	0.363	760
Dec	-1345.6	1037.7	1037.7	0.349	77.1

Since the GSHP could not reject as much heat to the ground as it should have, the cooling capacity was lower than the manufacturer's design point. During the heating season, the EWTs as expected are higher than the OAT, with the cross-over point around July when the OAT dipped below the EWT. Notwithstanding this shortfall, the GSHP was able to maintain the summer time thermostat set point with good fidelity, so there was no sizing problem from a comfort point of view. Perhaps, a slightly larger ground loop would have further reduced compressor run times, and saved more energy.

3.3 Compressor run times

We have already alluded to the fact that longer run times reduce cycle losses, characterized by the cyclic loss coefficient. From this standpoint, it behooves us to show not only the total monthly run times, but also the fraction of the run time at which the compressor ran at high or at low speed. Figure 4 shows that the total compressor run times peaks in July for the cooling season, and in December/ January for the heating season. For this particular

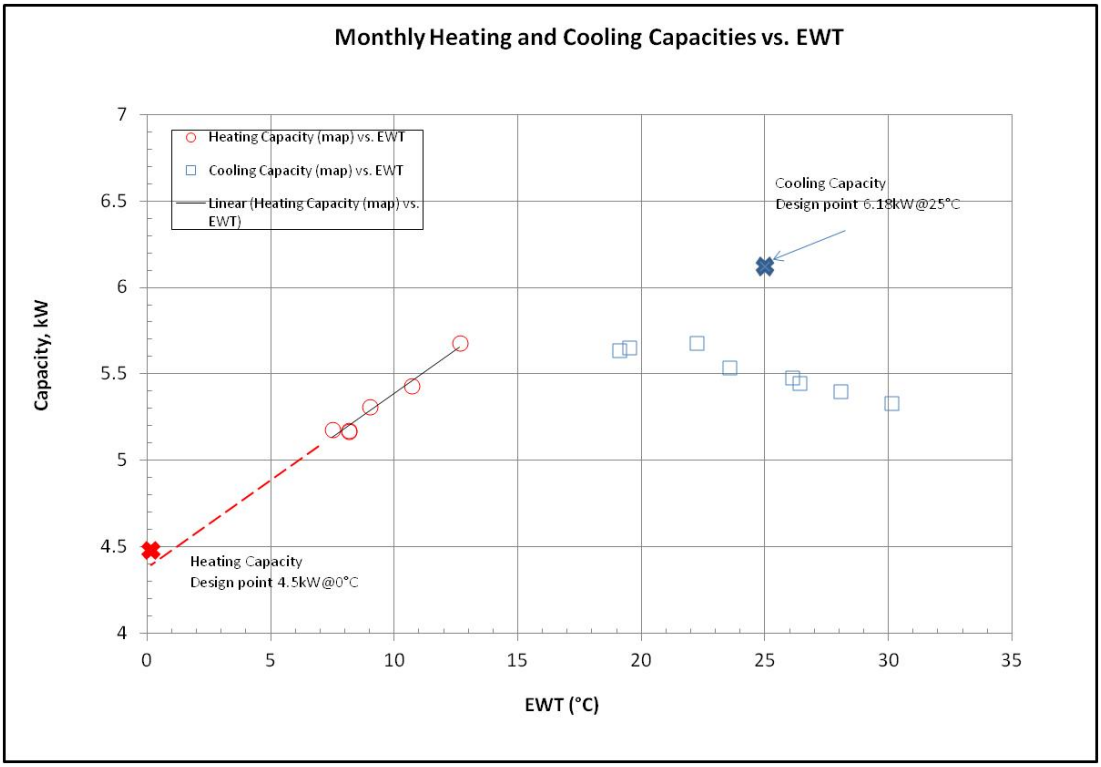


Figure 2: GSHP delivered capacity during heating and cooling seasons as a function of EWT

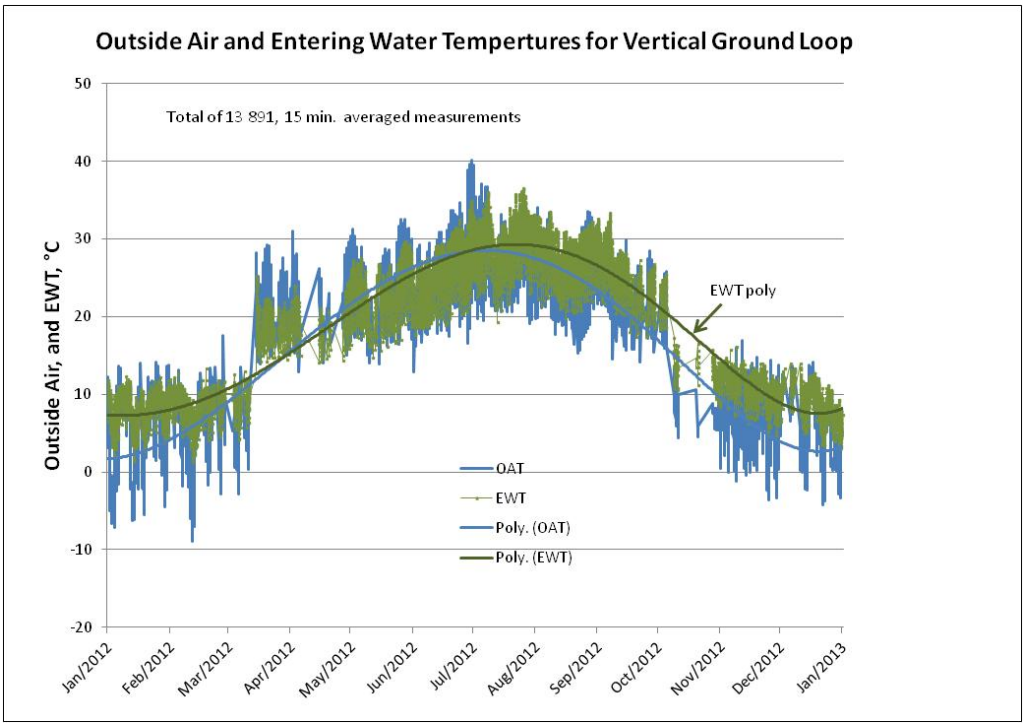


Figure 3: Variation of outdoor air temperature and EWT data taken when GSHP was operational

research house, the compressor run time during the peak cooling season is almost 33% higher than the run times for the peak heating season. In the US Climate Zone 4 where the research house is located, the heating and cooling loads are fairly balanced, and the longer run times during the cooling season is related to the vertical ground loop being undersized, as discuss in Section 3.2. However, despite the undersized loop, the percent of compressor run times at high speed is almost negligible. This is quite remarkable and shows that if the building envelope is well insulated, even a small (7.56 kW) GSHP unit with an undersized vertical ground loop is sufficient to serve space heating and space cooling needs. This has also been demonstrated for a shallow horizontal ground loop in the same climate region (Ally 2012).

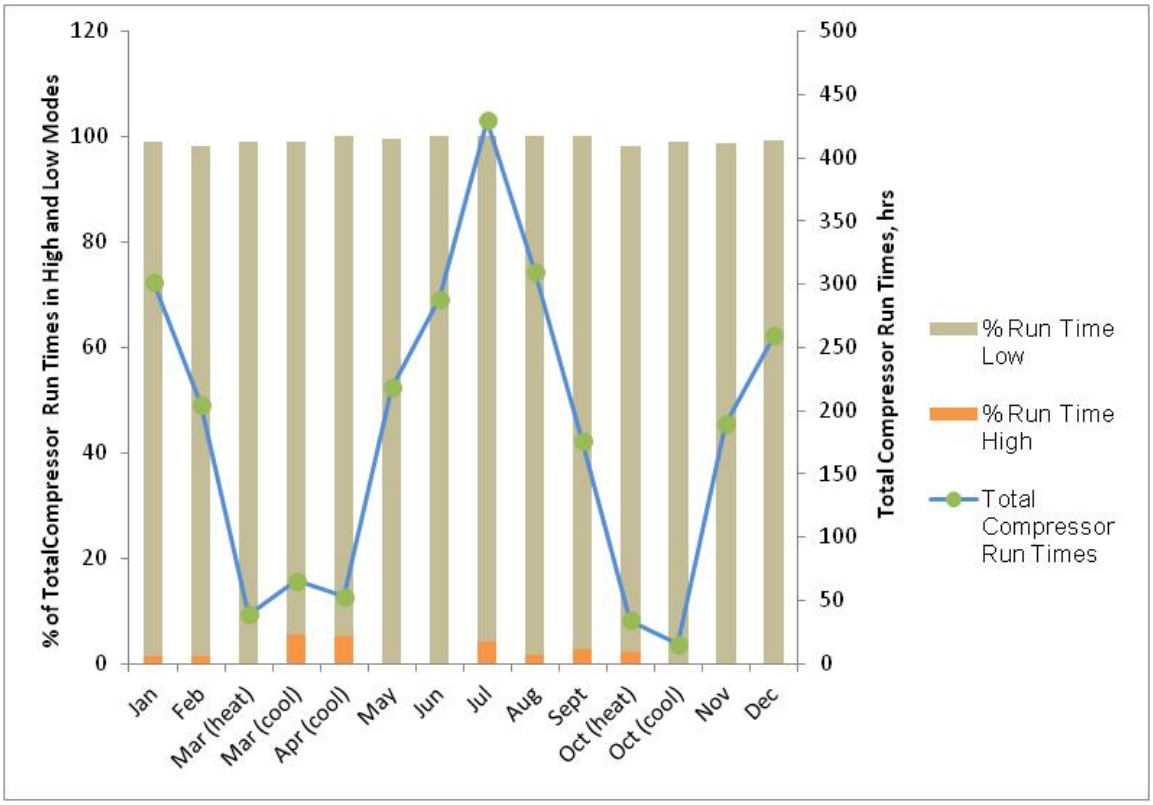


Figure 4: Compressor run times and the fraction of the total run time at which the compressor ran at low and at high speeds

3.4 Coefficient of performance

The heating season coefficient of performance, COP_h is given by Eq. (2)

$$COP_h = \frac{(Q_{GL\,HX} + W_{Comp.} + W_{Fan+Controls} + W_{Pump} + W_{e,\,Aux.})}{\left(W_{Comp.} + W_{Fan+Controls} + W_{Pump} + W_{e,\,Aux.}\right)} \quad (2)$$

Whereas, the cooling season coefficient of performance, COP_c is given by Eq. (3)

$$COP_c = \frac{\left(Q_{Indoor\,Coil} - W_{Fan+Controls} - Q_{Comp.}\right)}{\left(W_{Comp.} + W_{Fan+Controls} + W_{Pump}\right)} \quad (3)$$

With the above definitions, the monthly performances of the GSHP for the heating and cooling seasons are presented in Figure 5. For the cooling season, the COP decreases slightly with rising EWT, an expected trend. The highest cooling COP of 5.16 should be discounted because it occurred in the month of April when the OAT is close to the thermostat set point and the total compressor run time is only 53 hours. For peak summer months of May-September, the average cooling COP was 4.32. Similarly, for the heating season, the data for October and March are not very significant because in those months, there were only 34 hours and 30 hours of heating, respectively. For the high heating load months, the average heating COP_h is 4.11. The heating COP increases with EWT temperatures, although from our narrow range of EWT, this relationship is not very strong.

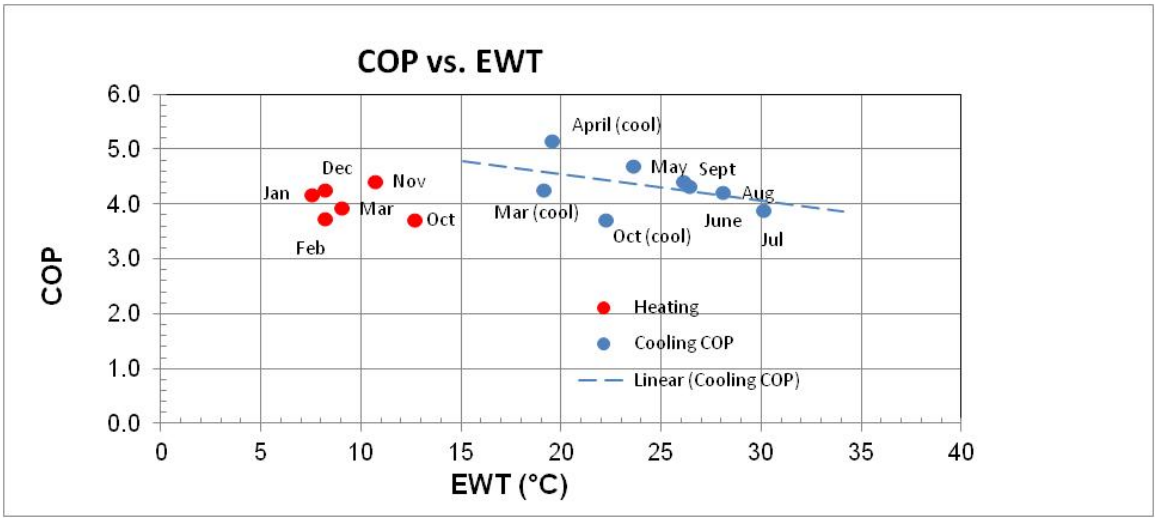


Figure 5: Variation of heating and cooling COP as a function of the EWT

4 CONCLUSION

This 12-month study suggests that GSHPs represent a practical technology option to reduce source energy reduction and greenhouse emissions under the IECC 2012 Standard, as well as the European Union (EU) 2020 target of generating over 25% of heat consumed in the EU from renewable energy. The 7.56 kW GSHP completely met the space conditioning loads of a 251m^2 residential research house with simulated occupancy loads. Practically no auxiliary (resistance) heat was needed in the winter months (Table 1). Most of the time the compressor ran at low speeds (Figure 4) and still met the space conditioning loads. The highest and lowest monthly cooling COP was in April (5.15) and in October (3.71), respectively. The average COP for all the cooling months was 4.33. The highest and lowest monthly heating COP was in November (4.43) and in December (4.27), respectively. The average COP for the entire heating season was 4.05. The COPs are calculated on the basis of total power input including fan, pumps, and control board power.

Most impressively, more than 75% of the energy for space heating was extracted from the ground loop (Table 3) demonstrating GSHP as a viable technology for reducing primary energy consumption.

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