12th IEA Heat Pump Conference 2017

Quantifying Systemic Efficiency using Exergy and Energy Analysis for Ground Source Heat Pumps: Domestic Space Conditioning and Water Heating Applications.

"Moonis Ally, Jeffrey Munk, Van Baxter, Anthony Gehl" *

"Oak Ridge National Laboratory, P.O. Box 2008 MS6070 (Oak Ridge, TN, USA)"

Abstract

Although air temperatures over land surfaces show wide seasonal and daily variations, the ground, approximately 10 meters below the earth's surface, remains relatively stable in temperature thereby serving as an energy source or sink. Ground source heat pumps can heat, cool, and supply homes with hot water efficiently by utilizing the earth's renewable and essentially inexhaustible energy resources, saving fossil fuels, reducing greenhouse gas emissions, and lowering the environmental footprint. In this paper, evidence is shown that ground source heat pumps can provide up to 79%-87% of domestic hot water energy needs, and up to 77% of space heating needs with the ground's thermal energy resources. The case refers to a 12-month study conducted at a 253 m^2 research house located in Oak Ridge, Tennessee, 36.01° N 84.26°W in a mixed-humid climate with HDD of 2218°C-days and CDD of 723°C-days under simulated occupancy conditions. A single 94.5m vertical bore interfaced the heat pump with the ground. The research shows that this technology is capable of achieving US DOE targets of 25 % and 35% energy savings in HVAC, and in water heating, respectively by 2030. It is also a viable technology to meet greenhouse gas target emissions under the IECC 2012 Standard, as well as the European Union (EU) 2020 targets of using renewable energy resources. The paper quantifies systemic efficiencies using Exergy analysis of the major components, clearly identifying areas for further improvement.

Selection and/or peer-review under responsibility of the organizers of the 12th IEA Heat Pump Conference 2017.

Keywords: Ground-source, heat pumps, exergy, sustainable, renewables

* Corresponding author. Tel.:+1+865-576-8003; fax: +1+865-574-9338.
E-mail address: allymr@ornl.gov.

This manuscript has been authored by UT-Battelle, LLC under Contract No. DE-AC05-00OR22725 with the U.S. Department of Energy. The United States Government retains and the publisher, by accepting the article for publication, acknowledges that the United States Government retains a non-exclusive, paid-up, irrevocable, world-wide license to publish or reproduce the published form of this manuscript, or allow others to do so, for United States Government purposes. The Department of Energy will provide public access to these results of federally sponsored research in accordance with the DOE Public Access Plan (http://energy.gov/downloads/doe-public-access-plan).

1. Introduction

The most recent data released in 2014 indicates that space heating (44.2%), space cooling (8.9%) and water heating (16.9%) accounted for almost 70% of all residential energy consumption in the United States[1]. Space conditioning currently accounts for 53% of energy consumption in U.S homes compared to 58% in 1993 [2].The factors responsible for this decrease include adoption of more efficient equipment, improved thermal insulation, better windows, and to some extent population shifts to warmer climates. The total U.S energy consumption in homes has remained relatively stable since 1985 at about 2931 TWh[3], despite an increase in the number and average size of housing units. The national average home energy expenditures is $2,024 (*ibid*.), and the average size of a residential home built in 2015 was 229 m^2 [4], approximately 61% bigger than those built 40 years ago [5]. In 2015, the residential primary electricity consumption was 4.24 x 10^{12} kWh, gas 1.46 x 10^{12} kWh, petroleum (heating oil),3.16 x 10^{11} kWh, and renewables only 1.49 x 10^{11} kWh [1]. Natural gas is the primary source for space heating, consuming 9.96 x 10^{11} kWh [1]. Renewable energy accounts for only 1.49 x 10^{11} kWh or merely 3.5% of the primary electric consumption of 4.24 x 10^{12} kWh. It is clear that the share of renewable energy resources should be increased to reduce dependence on fossil fuels for power generation. In order to meet binding treaty obligations of the United Nations climate conference of the parties (COP21) in Paris, the European Unions' (EU's) Intended Nationally Determined Contribution (INDC) commits the EU to a domestic reduction of greenhouse gas emissions by at least 40% below 1990 levels by 2030 [6]. The United States has committed to reduce its greenhouse gases (GHG) emissions by 26-28% below the 2005 level in 2025, and to make "best efforts" to reduce emissions by 28% [7].

The paper shows that thermal energy from the ground can be harvested to supply up to 77% and 87% of the energy for space conditioning and for water heating, respectively. Utilizing the ground's thermal energy is a viable alternative to provide utility heat as demonstrated in this paper. The technology has the capacity to save electricity and reduce GHG emissions. In this paper we summarize the measured performance of a ground-source heat pump (sized at 5.3 kW) for water heating, and a separate ground-source heat pump (sized at 7.56 kW) for space conditioning, both containing refrigerant R410A and both connected to the same single-bore vertical well, 94.5m deep. The study was conducted at a research home in Oak Ridge, Tennessee, USA, located at 36.01°N 84.26°W in a mixed-humid climate with heating degree days (HDD) of 2218°C-days and cooling degree days (CDD) of 723°C-days. The ground loop contained a mixture ("brine") of 20% (v/v) of propylene glycol and water, maintained at a pressure of 275.8 kPa with a bladder-inflated pressure tank. Energy balances are used to determine the extent of energy removal from the ground that is used for water heating and space conditioning. Sources of systemic irreversibility are identified by Second Law Analysis to understand and to quantify further improvements in efficiency. With current technology we demonstrate that thermal energy from the ground can be utilized as a practical and efficient renewable energy resource for consumers and that it is an enabling technology to mitigate climate change.

.

2. Single vertical-bore ground loop and heat pumps

The single vertical-bore consisted of a 1.9 cm ID high density polyethylene (HDPE) tube bent into a U-tube at the bottom of the vertical well. The space conditioning equipment was a two-capacity heat pump rated at 7.79 kW (cooling at full load), and 6.23 kW at part load with an Energy Efficiency Rating (EER) of 5.4 and 7.6, respectively, provided by the manufacturer[†]. The heating mode capacity was 5.80 kW (at full load) and 4.84 kW (at part load) with coefficient of performances of 4.0 and 4.6, respectively. The cooling capacity (full load) is based on entering air at 27°C dry bulb (DB); 19°C wet bulb (WB); entering water temperature (EWT)25°C. Part load cooling capacity is at the same entering air conditions but 20°C EWT. The heating capacity (at full load) is at 20°C DB, 15°C WB entering air temperature; 0°C EWT, and for part load at the same entering air conditions but at 5°C EWT. In the 2-capacity heat pump, full load corresponds to 100% compressor displacement, and part load at 67% displacement achieved by bypassing a small portion of the refrigerant back to the low side of the compressor by ports controlled via a solenoid valve [8]. The water heating heat pump was rated at a COP of 3.1 based on ground EWT of 0°C and utility entering water of 37.8°C. The hot water tank was a 303L seamless,

† http://www.climatemaster.com/share/Res_All_Products_CLM/Section_3_TT27.pdf, page 8

blow-moulded polybutylene tank equipped with 4.5 kW upper and lower heating elements for backup heating. The tank was rated at an energy factor (EF) of 0.92 for conventional heating with elements. The heat pump serviced 227 L/day hot water at 49°C for showers, dishwashers, clothes washing, and other uses in a typical household in the US. The thermostat set points for space cooling and for space heating were 24.4°C and 21.7°C, respectively.

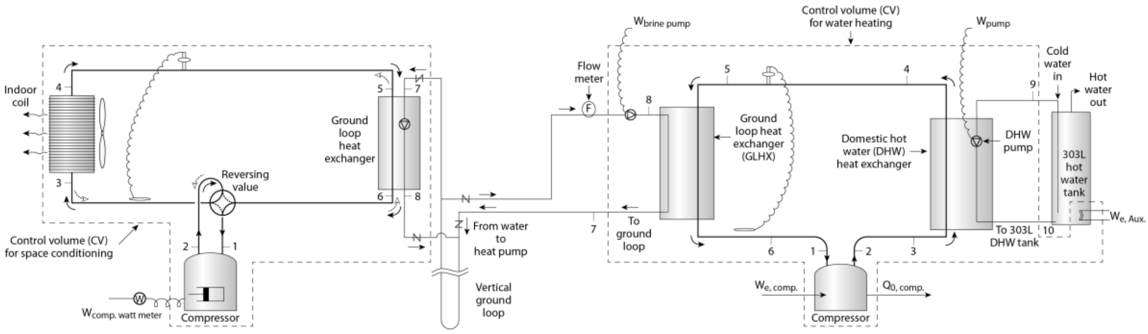

Figure 1 Schematic of the vertical-bore connected to the ground-source heat pumps for space conditioning (left) and for water heating (right).

Additional details of the equipment, instrumentation, data analysis, methodology, and thermo-physical properties at state points can be found elsewhere [9][10].

3. Analysis

Under assumption of quasi steady-state steady-flow (QSSF) the general mass, energy and entropy balances, between any two state points are given by Eqns. (1),(2) and (3) and the rate of irreversibility and total irreversibility, by Eqn. (4) .

$$\dot{m}_{in} - \dot{m}_{out} = \frac{dm}{dt} = 0 \tag{1}$$

$$\dot{Q}_0 + \sum_j \dot{Q}_j + \dot{W} + \sum_{in} \dot{m}\left(h + \frac{v^2}{2} + gz\right) - \sum_{out} \dot{m}\left(h + \frac{v^2}{2} + gz\right) = 0 \tag{2}$$

$$\dot{\sigma}_{Total} = \sum_{out} \dot{m}(s) - \sum_{in} \dot{m}(s) - \frac{\dot{Q}_0}{T_0} - \sum_j \frac{\dot{Q}_j}{T_j} \geq 0 \tag{3}$$

$$\dot{I} = \dot{\sigma} T_0 \;\; and \, I = \dot{I} \times t \tag{4}$$

Energy or work entering a control volume is positive and leaving is negative. The coefficient of performance is defined customarily as the ratio of the energy delivered to the total energy input.

4. Energy Balance and Systemic Irreversibility during space conditioning

The monthly energy balance is segmented into two parts: energy input and energy output, as shown in Table 1 and Table 2 with agreement within experimental 2.3% <error <3.92% where error = (E_{In}-|E_{out}|)/E_{In} expressed as a percentage. Note that March and October are “shoulder months” meaning that both heating and cooling are required in those months. During the entire year, there was virtually no need for any auxiliary (resistance) heat to supplement space heating. One of the main points in this paper is to show that thermal energy extracted from the ground is a viable renewable energy resource that can be utilized for space heating. This is exemplified by the data in the last column in Table 3 where it is shown that over 75% of the space heating needs is extracted from the ground, making this technology a good candidate for utilizing the ground's renewable energy resource. The data in Table 3 is derived from Table 1 and Table 2.

Areas of systemic inefficiency within the space conditioning control volume are determined by applying Eqns. (3) and (4) to the state points across each component with the results displayed in Figure 2.

Table 1 Total monthly energy input for each component of the ground-source heat pump for space conditioning, and yearly total

	$W_{Comp.}$	$W_{Fan + Controls}$	$W_{Brine\ Pump}$	$Q_{Brine\ HX}$	Q_{Indoor}	$W_{e,Aux}$	E_{In}
(yr. 2012)	(kWh)	(kWh)	(kWh)	(kWh)	(kWh)	(kWh)	(kWh)
Jan	317.9	60.3	50.9	1204.3		0.003323	1633.4
Feb	217.0	41.5	34.8	823.6		0.001842	1116.8
Mar (heat)	39.4	10.7	6.3	147.0		0	203.4
Mar (cool)	50.9	17.0	10.5		355.1		433.5
April (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
August	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		0.095415	181.0
Oct (cool)	12.2	4.1	2.4		73.7		92.4
Nov	199.1	64.5	30.0	808.3		0.001516	1101.9
Dec	268.3	50.2	43.2	1037.7		0.001038	1399.4
Total	**2568.7**	**507.3**	**429.4**	**4154.2**		**0.103134**	**16581.8**

Table 2 Total monthly energy output for each component of the ground-source heat pump for space conditioning and yearly total

	$Q_{Indoor\ HX}$	$W_{Fan + Controls}$	$Q_{Comp.}$	$Q_{Aux.}$	$Q_{Brine\ HX}$	E_{out}
(yr. 2012)	(kWh)	(kWh)	(kWh)	(kWh)	(kWh)	(kWh)
Jan	-1446.3	-60.3	-63.9	-0.0033		-1570.6
Feb	-983.8	-41.5	-48.5	-0.0018		-1083.8
Mar (heat)	-173.7	-10.7	-11.0	0.0000		-195.4
Mar (cool)		-17.0	-9.4		-396.7	-423.1
April (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
Jul		-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	-0.0954		-174.1
Oct (cool)		-4.1	-3.4		-82.5	-90.0
Nov	-960.2	-64.5	-39.3	-0.0015		-1063.9
Dec	-1246.2	-50.2	-49.2	-0.0010		-1345.6
Total	**-4967.0**	**-507.3**	**-408.1**	**-0.103**	**-10209.7**	**-16092.2**

Table 3 Percent of total space heating load derived from the ground

	Total Energy Delivered	Total Electrical Energy Used	Energy Extracted from Ground Loop	$\frac{W_{e,Total}}{Q_{BrineHX}}$	% Delivered Energy from Ground Loop
(yr. 2012)	(kWh_t)	(kWh_e)	(kWh_t)		
Jan	-1570.6	429.1	1204.3	0.356	76.7
Feb	-1083.8	293.2	823.6	0.356	76.7
Mar (heat)	-195.4	56.4	147.0	0.384	75.2
Mar (cool)					
Apr (cool)					
May					
Jun					
Jul					
Aug					
Sept					
Oct (heat)	-174.1	47.6	133.4	0.357	76.6
Oct (cool)					
Nov	-1063.9	293.6	808.3	0.363	76.0
Dec	-1345.6	361.7	1037.7	0.349	77.1

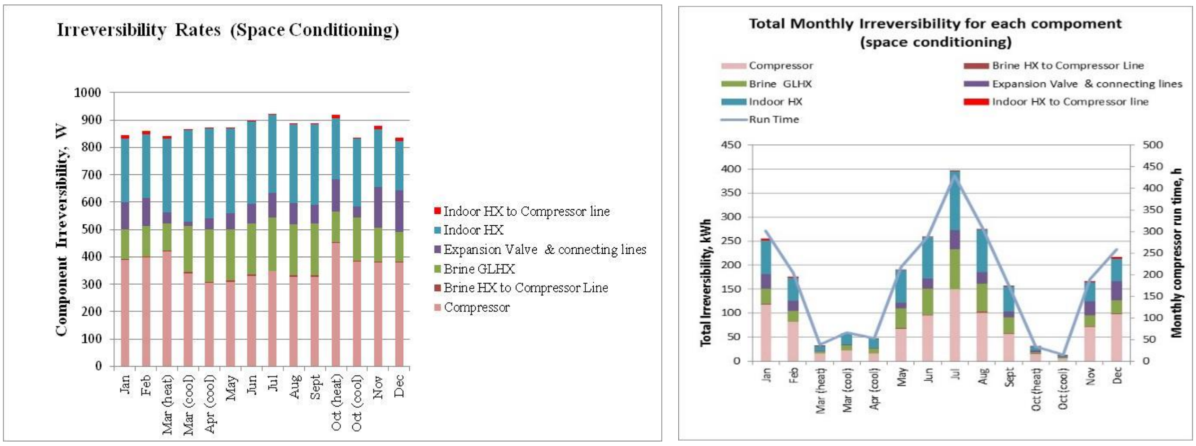

Figure 2 Component-wise irreversibility rates (left) and total irreversibility, I (Right) for space conditioning. The compressor run times (hours) are read on the right ordinate

The irreversibility rate remains fairly stable throughout the year (Figure 2 (left)) with the major sources of systemic irreversibility being the compressor, the indoor heat exchanger, and the brine ground loop heat exchanger. These are the areas for technological improvement in efficiency. The thermodynamic 'Lost Work' is the total irreversibility (I) computed by multiplying the irreversibility rate with the duration (same as the compressor run time) over which the irreversibility occurred. The total irreversibility or "Lost Work" shown in Figure 2 (right) represents the opportunity for improving space conditioning efficiency lost to systemic inefficiency. "Lost work" is high during the peak winter and summer months when the run times are high, and they are lowest during the Spring and Autumn months, when run times are low due to mild weather. Compressor run times are on the right ordinate in Figure 2. Throughout the year, the indoor relative humidity was between 50-55%.

5. Energy Balance and Systemic Irreversibility during Water Heating

The control volume for the water heating cycle is shown on the right hand side of Figure 1. By applying Eqns.

(1) and (2) across various state points, the component-wise energy balance is shown in Table 4. The refrigerant flow rate in the system varied slightly for each month with an averaged refrigerant mass flow rate of 0.02553 $kg \cdot s^{-1}$ and standard deviation of $\pm$ 0.0031 $kg \cdot s^{-1}$. Details of the methodology, thermodynamic state points, and analysis are referenced in [10]. The energy balance is established within 6% with standard deviation of +0.43%. The total energy input slightly exceeded the total energy output, with the balance attributed to heat losses. The thermal energy extracted from the ground and the thermal energy delivered to make domestic hot water are given in columns 8 and 9, respectively in Table 4. A better appreciation of how energy is distributed is presented in Table 5 where the percent of the energy extracted from the ground that goes to heat water and the percent of energy from the ground relative to the total energy input are tabulated in columns 7 and 4, respectively.

Table 5 clearly shows that greater than 79% of the thermal energy needed to produce domestic hot water came from the ground, demonstrating the effectiveness of the ground to be used as an energy resource. The lowest and highest water heating coefficient of performance was 2.69 and 3.57 in January and in July, respectively. Systemic irreversibility is calculated by invoking Eqns. (3) and (4) details of which are reported in [10] are shown in Table 6. The most irreversible component is the compressor. The total systemic irreversibility representing “Lost work” tabulated in the last column, is obtained by the product of the irreversibility rate and the compressor run time given in Table 5.

Table 4 Monthly Energy Balance of the ground source heat pump water heater

	Brine Pump	Compressor	DHW Pump	Auxiliary (Resistance heat)	Controls	Total Electrical Energy In	Ground Heat	Domestic Hot Water (DHW)	Heat from Compressor	Total Energy In (E_{in})	Total Energy Out (E_{out})	%Δ
2012	(kWh_e)	(kWh_e)	(kWh_e)	(kWh_e)	(kWh_e)	(kWh_e)	(kWh_t)	(kWh_t)	(kWh_t)	(kWh)	(kWh)	
Jan	12.0	121.5	4.5	0.0	2.0	140.0	298.8	-376.1	-39.3	438.9	-415.4	5.35
Feb	11.1	112.1	4.2	0.0	1.8	129.3	295.3	-371.7	-29.3	424.6	-401.0	5.5
Mar	9.8	101.0	3.6	0.0	1.7	116.1	290.9	-356.0	-28.6	407.0	-384.5	5.51
Apr	8.3	86.5	3.1	0.0	1.5	99.4	256.4	-311.1	-26.2	355.8	-337.4	5.18
May	7.6	81.6	2.8	0.0	1.4	93.4	261.9	-311.4	-26.9	355.2	-338.3	4.77
Jun	6.3	69.9	2.4	0.0	1.2	79.8	233.1	-273.9	-24.7	312.9	-298.6	4.59
Jul	5.5	66.1	2.3	0.0	1.2	75.1	232.6	-268.2	-25.4	307.7	-293.5	4.59
Aug	6.2	71.6	2.5	0.0	1.3	81.5	244.4	-284.6	-26.6	325.9	-311.2	4.50
Sept	6.5	71.0	2.5	0.0	1.2	81.1	234.5	-276.5	-24.4	315.7	-300.9	4.66
Oct	8.2	85.4	3.0	0.0	1.5	98.1	253.3	-304.5	-28.8	351.4	-333.3	5.16
Nov	9.3	96.3	3.5	0.0	1.6	110.7	257.9	-316.6	-32.0	368.6	-348.5	5.44
Dec	10.3	105.8	3.9	0.1	1.7	121.8	272.8	-337.5	-34.9	394.7	-372.3	5.66
					Total	1226	kWh				std. dev. =	0.43%

Table 5 Percent of total water heating energy required that is extracted from the ground and the COP: monthly basis

	COP	$\frac{W_{e,Comp.}}{E_{In}}$	$\frac{Q_{o,ground}}{E_{In}}$	Compressor Run time, t	$\lvert Q_{DHW} \rvert$	$\frac{\%Q_{GLHX}}{\lvert Q_{DHW} \rvert}$
(Yr. 2012)		(%)	(%)	(hrs)	(kWh)	
Jan	2.69	28%	68%	90.29	376.1	79.4%
Feb	2.88	26%	70%	83.11	371.7	79.5%
Mar	3.07	25%	71%	73.20	356.0	81.7%
Apr	3.13	24%	72%	62.13	311.1	82.4%
May	3.34	23%	74%	58.06	311.4	84.1%
Jun	3.43	22%	74%	49.40	273.9	85.1%

Jul	3.57	21%	76%	46.40	268.2	86.7%
Aug	3.49	22%	75%	50.50	284.6	85.9%
Sept	3.41	22%	74%	50.38	276.5	84.8%
Oct	3.10	24%	72%	61.46	304.5	83.2%
Nov	2.86	26%	70%	70.67	316.6	81.5%
Dec	2.77	27%	69%	78.35	337.5	80.8%

Table 6 Component irreversibility rate and total irreversibility for the ground source heat pump water heater: monthly averaged data. Run times are from Table 5.

	Compressor	Compressor Discharge to DHW HX	DHW HX	Expansion Valve & connecting lines	Refrig.-Brine GL HX	Ref.-Brine GL HX to Compressor suction	Total Irreversibility Rate,	Total Irreversibility $I = \dot{I} \times t$
State points →	[1→2]	[2→3]	[3→4]	[4→5]	[5→6]	[6→1]		
	$\dot{I}$	$\dot{I}$	$\dot{I}$	$\dot{I}$	$\dot{I}$	$\dot{I}$	$\dot{I}$	I
	(W)	(W)	(W)	(W)	(W)	(W)	(W)	(kWh)
Jan	512	6.52	91	167	85	-1.21	859	77.5
Feb	487	7.45	104	166	76	-1.26	839	69.7
Mar	486	7.94	114	169	97	-0.88	873	63.9
Apr	489	7.70	117	170	107	-0.83	889	55.3
May	489	7.92	127	165	142	-0.07	932	54.1
Jun	493	8.16	135	163	170	0.35	970	47.9
Jul	502	9.66	149	156	208	1.02	1025	47.6
Aug	498	8.61	140	158	195	0.78	1001	50.5
Sept	493	8.15	134	161	167	0.26	964	48.5
Oct	497	7.78	112	166	124	-0.47	907	55.8
Nov	498	7.38	99	170	97	-1.10	869	61.4
Dec	498	7.39	93	170	91	-1.30	858	67.2

6. Discussion

The electricity consumption for space conditioning (heating and cooling) from Table 1 is 2568.7 kWh (Compressor), 507.3 kWh (fan + controls), 429.4 kWh (brine pump) and a negligible 0.1 kWh (resistance) for a total of 3505 kWh. For water heating, the total electricity consumption from Table 4 is 1226 kWh. The total electricity consumption therefore is 4732 kWh.

Since the average sizes of residential homes have intra-country variances, it is instructive to display electricity consumption in space conditioning, water heating, and the total of the two on a unit area basis for the ground-source technology. This is shown in Figure 3 where it is clear that the water heating electricity consumption remains fairly stable throughout the year with a slight trough during peak summer months when the incoming city water temperature is higher than during the winter months. The space conditioning load has a “W” shape, with peaks during the winter and summer months. The fraction of water heating load to the total (space conditioning + water heating) shown on the secondary axis of Figure 3 varies seasonally. It peaks during the “shoulder” months (low space conditioning loads) and troughs during the peak summer and winter months (high space conditioning loads), since electricity consumption for water heating is stable throughout the year.

The average household electricity use in the United Sates is about 12,000 kWh/y of which space conditioning and water heating account for 70% [1] and the remaining 30% accounts for appliances, electronics, and lighting. Taking our figure of 4732 kWh and dividing by 0.7 yields 6760 kWh as the annual electricity consumption for the average US household, a 43.6% reduction relative to 12,000 kWh.

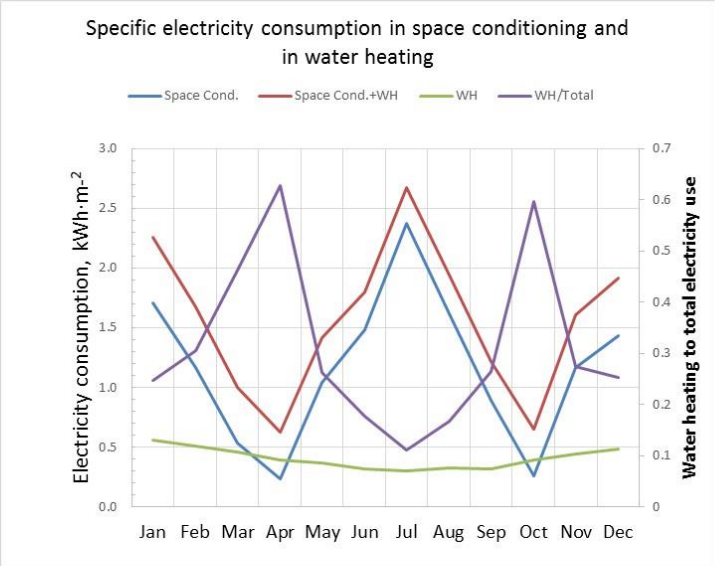

Figure 3 Specific electricity (kWh·m^{-2}) consumption and the fractional water heating electricity consumption.

For this single-family home, the vertical-bore serving both space conditioning and water heating applications was slightly under sized. The under sizing is apparent from the temperature profile of the ambient air and the entering water temperature (EWT) shown in Figure 4.

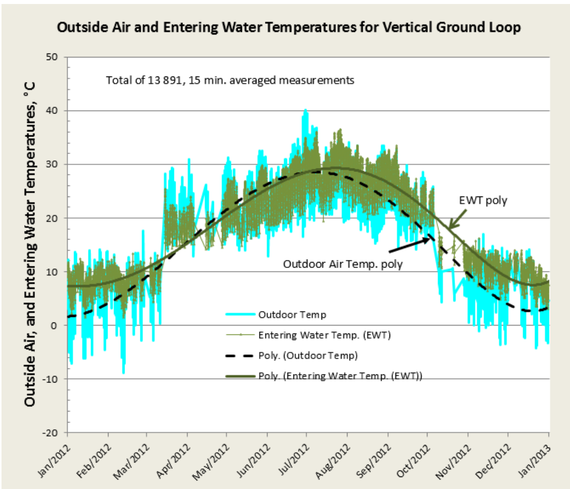

Figure 4 Measured out door air and entering water (brine) temperatures from the ground loop

A vertical bore approximately 10% deeper would have been a better choice because the EWT would be cooler in the summer and warmer in the winter. The EWT is not only dependent on the outside air temperature, but also on the heat rejected or extracted during space cooling and space heating, respectively, while for water heating the heat is always extracted from the ground. These thermal exchanges occur at different times within the 24-hour cycle, affecting the EWT. It is difficult to estimate accurately the soil characteristics and the thermal exchange with the ground for sizing purposes which vary with karst geology, soil moisture content, grout composition, and bore characteristics.

7. Conclusions

Data collected over a 12-month period at a research house located in a mixed-humid climate in the United States with HDD of 2218 °C-days and CCD of 723 °C-days shows that the electricity consumption for space conditioning and for water heating can be reduced by about 40% relative to the average US household by using ground-source heat pumps with existing technology.

The total annual electricity consumed for space conditioning and water heating was 3505 kWh and 1226 kWh, respectively. The heat pump delivered 227 L/day hot water at 49°C. More than 79% of the energy required to heat the water came from the ground (Table 5). The thermostat set points for space cooling and for space heating were 24.4°C and 21.7°C, respectively. Better than 75% of the thermal energy for space heating came from the ground (Table 3). This proves that the ground is a useful energy resource that may be utilized for providing thermal loads to building loads.

Systemic irreversibility represents a direct measure of “Lost Work” within the system is quantified using the Second Law of Thermodynamics. The rate of irreversibility and the total irreversibility for space conditioning are shown in Figure 2 and for water heating in Table 6. The components displaying the highest irreversibility are the compressor, indoor coil, and the ground loop heat exchanger.

Overall, ground loop heat pumps are a viable technology to reduce the dependence on electricity generated from fossil fuel and should be considered as part of the solution to mitigate anthropogenic atmospheric carbon dioxide emissions.

8. Acknowledgements

The authors are grateful to the U.S Department of Energy, Buildings Technology Office program manager, Mr. Antonio Bouza for supporting this work under a Cooperative Research and Development Agreement (CRADA) with Climate Master, Inc. The authors sincerely thank Ahmad Abu-Heiba and Kashif Nawaz at Oak Ridge National Laboratory for reviewing the manuscript and suggesting changes.

9. References

[1]. U.S. Department of Energy Buildings Energy Data Book, 2.1 Residential Sector Energy consumption, 2014. http://buildingsdatabook.eren.doe.gov/TableView.aspx?table=2.1.6

[2]. U.S. Energy Information Administration, Residential Energy Consumption Survey, 2009, http://www.eia.gov/consumption/residential/index.cfm

[3]. U.S. Energy Information Administration Residential Energy Consumption Survey, 2009, http://www.eia.gov/todayinenergy/detail.php?id=6570

[4]. United States Census Bureau. Characteristics of New Housing completed in 2015. http://www.census.gov/construction/chars/highlights.html

[5]. Perry, M. J. Today’s new homes are 1,000 square feet larger than in 1973, and the living space per perso has doubled over last 40 years. American Enterprise Institute, February 26, 2014, https://www.aei.org/publication/todays-new-homes-are-1000-square-feet-larger-than-in-1973-and-the-living-space-per-person-has-doubled-over-last-40-years/

[6]. European Parliament Briefing, EU position for COP 21 climate change conference. November 2015. http://www.europarl.europa.eu/RegData/etudes/BRIE/2015/572787/EPRS_BRI%282015%29572787_EN.pdf

[7]. Earth Institute, State of the Plant, Columbia University, December 11, 2015, http://blogs.ei.columbia.edu/2015/12/11/what-is-the-u-s-commitment-in-paris/

[8]. Emerson. Application Engineering Bulletin: ZPS20 to ZPS60K4 and ZPS20 to ZPS51K5 R-410A 1.5 to 5 Ton. Copeland Scroll UltraTech TM Compressors, 2011.

[9]. Ally, M. R., Munk, J. D., Baxter, V, D., Gehl, A. C, (2015). Exergy analysis of a two-stage ground source heat pump with a vertical bore for residential space conditioning under simulated occupancy. *Applied Energy* 155 (2015) 502-514. http://dx.doi.org/10.1016/j.apenergy.2015.06.004

[10]. Ally, M. R., Munk, J. D., Baxter, V, D., Gehl, A. C, (2015). Data, exergy, and energy analyses of a vertical-bore, ground-source heat pump for domestic water heating under simulated occupancy conditions. *Applied Thermal Engineering* 89 (2015) 192-203. http://dx.doi.org/10.1016/j.applthermaleng.2015.05.021

10. Nomenclature

Quantities

CDD	cooling degree days
COP	coefficient of performance (dimensionless)
DOE	United States Department of Energy
DHW	domestic hot water
EWT	entering water temperature (K)
GL	Ground Loop
GSHP	ground source heat pump
GWP	global warming potential
HDD	heating degree days
HX	Heat Exchanger
g	gravitational acceleration ($m \cdot s^{-2}$)
h	enthalpy ($kJ \cdot kg^{-1}$)
$\dot{I}$	rate of thermodynamic irreversibility (W, or kW)
I	total thermodynamic irreversibility (kWh)
LWT	leaving water temperature (K)
$\dot{m}$	mass flow rate ($kg \cdot s^{-1}$)
P	pressure (kPa)
$\dot{Q}$	thermal energy flow (W)
QSSSF	Quasi-Steady-State-Steady Flow
s	entropy ($kJ \cdot kg^{-1} \cdot K^{-1}$)
T	temperature (K)
U	internal energy in control volume (kJ)
V	velocity ($m \cdot s^{-1}$)
$\dot{W}$	rate of work (W)
%Δ	percent deviation
η	efficiency (dimensionless)
$\dot{\sigma}$	rate of entropy generation ($W \cdot K^{-1}$)

Subscripts

Aux.	Auxiliary heat or resistance heat from heating elements
b	brine
Brine-HX	brine heat exchanger
comp.	pertaining to the compressor
CV	control volume
EWT	entering water temperature
GLHX	Ground loop Heat Exchanger
Ground	the ground surrounding the vertical U-tube ground loop
i	inlet location
e	exit location or electrical energy
j	thermal reservoir other than the dead state or surroundings
o	dead state or surroundings
Indoor- HX	refrigerant-to-air heat exchanger