Exergy Analysis of a Two-Stage Ground Source Heat Pump with a Vertical Bore for Residential Space Conditioning under Simulated Occupancy[☆]

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Abstract

This twelve-month field study analyzes the performance of a 7.56W (2.16-ton) water-to-air-ground source heat pump (WA-GSHP) to satisfy domestic space conditioning loads in a 253 m² house in a mixed-humid climate in the United States. The practical feasibility of using the ground as a source of renewable energy is clearly demonstrated. Better than 75% of the energy needed for space heating was extracted from the ground. The average monthly electricity consumption for space conditioning was only 40 kWh at summer and winter thermostat set points of 24.4°C and 21.7°C, respectively. The WA-GSHP shared the same 94.5 m vertical bore ground loop with a separate water-to-water ground-source heat pump (WW-GSHP) for meeting domestic hot water needs in the same house. Sources of systemic irreversibility, the main cause of lost work, are identified using Exergy and energy analysis.

[☆] This paper is a collaborative effort.

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1 Quantifying the sources of Exergy and energy losses is essential for further systemic
2 improvements. The research findings suggest that the WA-GSHPs are a practical and viable
3 technology to reduce primary energy consumption and greenhouse gas emissions under the IECC
4 2012 Standard, as well as the European Union (EU) 2020 targets of using renewable energy
5 resources.

6 *Keywords:* Sustainability, Exergy, Availability, Geothermal, Heat Pump

7

1 **Nomenclature**

2 *Quantities*

3	ACH	Air Changer per hour (dimensionless)
4	ASHRAE	Amer. Soc. Heating Refrigeration and Air-Conditioning
5	CDD	Cooling degree days, °C-days or °F-days
6	COP	Coefficient of performance (dimensionless)
7	CT	Current Transformer
8	CV	Control Volume or System Boundary
9	DOE	U.S Department of Energy
10	EER	Energy Efficiency Ratio, (dimensionless)
11	EU	European Union
12	EWT	Entering Water Temperature, °C
13	FS	Full Scale
14	GLHX	Ground Loop Heat Exchanger
15	GSHP	Ground Source Heat Pump
16	g	gravitational acceleration ($\text{m}\cdot\text{s}^{-2}$)
17	h	enthalpy ($\text{kJ}\cdot\text{kg}^{-1}$)
18	HDD	Heating Degree Days, °C-days or °F-days

1	HERS	Home Energy Rating System
2	$\dot{I}$	rate of thermodynamic irreversibility (W)
3	I	irreversibility (kWh)
4	LWT	Leaving Water Temperature, °C
5	$\dot{m}$	mass flow rate (kg·s ⁻¹)
6	Mtoe	Million tons of oil equivalents
7	OEM	Original Equipment Manufacturer
8	P	pressure (kPa)
9	PCM	Phase change material
10	$\dot{Q}$	thermal energy flow (W)
11	QSSSF	Quasi Steady-State-Steady-Flow process
12	R _{SI}	R-value of insulation in SI
13	R _{US}	R-value of insulation in English units
14	s	entropy (kJ·kg ⁻¹ ·K ⁻¹)
15	SSSF	Steady-State-Steady Flow process
16	SHGC	Solar Heat Gain Coefficient, (dimensionless)
17	T	temperature (K)
18	U	internal energy (kJ·kg ⁻¹)

1	V	velocity ($\text{m}\cdot\text{s}^{-1}$)
2	$\dot{W}$	rate of work (W)
3	WA-GSHP	Water-to-air ground source heat pump
4	WW-GSHP	Water-to-water ground source heat pump
5	$\%\Delta$	percent deviation
6	η	efficiency (dimensionless)
7	$\dot{\sigma}$	rate of entropy generation ($\text{W}\cdot\text{K}^{-1}$)
8		
9	<i>Subscripts</i>	
10	<i>Comp.</i>	Compressor
11	<i>discharge</i>	compressor discharge port
12	<i>o</i>	dead state corresponding to the surroundings
13	<i>j</i>	thermal reservoir other than the dead state
14	<i>EWT</i>	entering water temperature ($^{\circ}\text{C}$)
15	<i>Brine HX</i>	<i>brine heat exchanger</i>
16	<i>Indoor HX</i>	Indoor refrigerant-to-air heat exchanger
17	<i>suction</i>	compressor suction port
18	Fans+Controls	fans and electronic controls on electronic boards

1	pump	brine pump
2	e,Aux	electrical energy of the auxiliary heating element
3	GLHX	ground loop heat exchanger
4	e,Total	total electrical energy
5	Total	aggregate sum of the elements under consideration
6		

1. Introduction

While the ultimate impact of climate change may be difficult to quantify, scientific evidence suggests a causal relationship between greenhouse gas emissions and its effects on global warming trends, food production, frequency and severity of weather patterns, water shortages, diseases, and environmental stress for flora and fauna [1]. Approximately 33% of total global greenhouse gas emissions come from buildings as a result of fossil fuel consumption for their operation [1]. In the United States, the buildings sector accounted for about 40.7% of primary energy consumption in 2013, compared to 30.5 % for industry, and 28.5% for transportation [2]. In the European Union (EU), buildings represent approximately 40% of the EU's energy consumption and carbon dioxide production [3].

The trends in coal, oil, and natural gas consumption and the emissions of greenhouse gases from combustion and other sources such as refrigerants are deemed unsustainable. Options for reducing fossil energy consumption include the use of renewable energy for producing electricity and thermal energy, summarized in a report, "Renewable Energy Sources and Climate Change Mitigation" by the Intergovernmental Panel on Climate Change [4]. Although in 2008, renewable energy contributed about 21% of the global electricity supply (16% hydroelectric, 3% other renewable energy, 2% biofuels of global transport fuel supply), both solar thermal and geothermal energy accounted for only 2% of the total global demand for heat [4], [5].

The buildings sector in the U.S. consists of 5 million commercial buildings and 115 million residential households, consuming 6×10^{12} kWh, approximately 40% of the total U.S. energy

1 demand (11×10^{12} kWh) [6] [7] in 2013. There is an abundance of thermal energy in the ground
2 that can be used for space and water heating, even from shallow sub-surfaces [8][9]. With
3 growing demand for space conditioning, and the parallel need to decrease fossil energy
4 consumption, researchers are looking at various hybrid options such as hybridizing ground
5 source heat pump (GSHP) systems with auxiliary systems [10], [11].

6
7 A comparison of ground-source heat pump technology versus conventional heating systems in
8 terms of costs and carbon dioxide emissions showed GSHPs to be a cleaner and practical
9 alternative [11]. The energy savings by deploying water source heat pumps in an apartment
10 complex with eighty-seven apartments each with 949 m^3 conditioned floor space was reported by
11 Chen et al. [13]. Estimates of energy and CO_2 saving with GSHPs in 10 states in India are
12 summarized by Sivasakthivel et al. [14]. Globally, interest in using the ground as a thermal
13 energy resource is increasing. Among the 28 EU countries alone, approximately 260 million tons
14 of oil equivalent (Mtoe) is emitted annually from land surfaces [15], signaling a huge potential
15 energy resource.

16
17 Recently, Exergy analysis of a three-circuit vertical ground- loop with two GSHPs [16] with total
18 cooling and heating capacities of 1324 kW and 1458 kW, respectively installed for a $14,000 \text{ m}^2$
19 hotel building showed that the greatest Exergy penalty occurred in the compressor and the
20 ground loop. Also, Exergy losses in the heating mode were higher than in the cooling mode. In
21 laboratory testing of GSHPs with R134a [17] during the heating months from January-May
22 showed that residential heating could be done in Erzurum, one of the coldest climatic regions in
23 Turkey, with heating COPs in the range of 2.07 – 3.04. Energy analysis of other vertical-bore

1 heat pump systems using R-22 and R-134a have been reported by [18]- [20], respectively. Healy
2 and Ugursal [21] investigated computationally the feasibility of using a GSHP in place of a
3 conventional heating (gas furnace) and cooling (air-source heat pump) system and concluded that
4 although system parameters have a significant effect on performance, the GSHP is economically
5 preferable to conventional systems in certain Canadian climates. They report that a typical GSHP
6 has a COP of 2.9 – 3.2 at 2°C EWT whereas, all the conventional systems have lower COPs. For
7 example, resistance heaters have a COP of 1; oil heating a COP between 0.65 – 0.7; and a high
8 efficiency natural gas furnace, a COP between 0.8 – 0.95. The COP of an air-source heat pump
9 is also lower than that of the GSHP given the fact that during winter, the air temperatures in
10 Canada are much lower than the ground temperatures. Their modeling estimated that the GSHP
11 may provide 65% – 70% energy savings relative to the conventional systems.

12
13 A meticulous examination of the practicality of using the ground as a source and sink for thermal
14 energy transfers in space conditioning applications is deliberated in this paper. Although thermal
15 energy from the ground is readily available as a renewable resource, its practical use, even in
16 developed economies, is not yet widespread. In this paper we provide metrics to show that by
17 using the ground as a thermal source for space heating, a large fraction of the space heating load
18 can be extracted from the ground. In space cooling, the ground acts as a thermal sink. During
19 the 12-month field test, the average monthly electricity consumption was determined to be only
20 360 kWh for a 253 m² (2721 ft²) residential house operated under simulated occupancy in a
21 mixed-humid climate zone 4 [22] in the U.S. Component-wise Exergy analysis identifies
22 systemic inefficiencies, the main cause of Lost Work. The study captures the energy use
23 patterns, seasonal variations, and electricity consumption for heating and cooling in a U.S

household located in Oak Ridge, Tennessee, 36.01° N 84.26° W in a mixed-humid climate with HDD of $2218^{\circ}\text{C-days}$ ($3993^{\circ}\text{F-days}$) and CDD of 723°C-days ($1301^{\circ}\text{F-days}$).

This paper presents measured performance of a vertical-bore ground-source heat pump, and uses the Second Law (Exergy) analysis to identify system components where efficiency improvements may be made.

Although energy and Exergy losses in GSHPs have been investigated by researchers in several countries, this study presents a comprehensive 12-month analysis of the WA-GSHP in a mixed-humid climate where latent cooling loads are relatively high during the peak cooling season.

Both the WA-GSHP as well as the WW-GSHP are connected to the same single bore vertical loop to satisfy both space conditioning as well as water heating loads of a typical U.S household.

A solitary, 94.5m (310 ft) vertical bore ground loop served both the water- to-air ground-source heat pump (WA-GSHP) for space conditioning, as well as a separate water-to-water ground-source heat pump (WW-GSHP) for domestic hot water production. Both heat pumps were connected to the same vertical-bore. The scope of this paper is limited to space conditioning and the results for domestic water heating will be published separately.

Since building loads are dependent on the envelope, this paper first describes the envelope characteristics followed by instrumentation, data acquisition, analysis, and measures of performance. Exergy analysis quantifies the discrepancy between actual and ideal performance, providing a true measure of systemic inefficiency, where they occur, and where engineers need to

focus to achieve higher efficiencies.

2. Experimental facilities and methodology

This research documents the performance and sources of inefficiencies of a vertical-bore GSHP over a continuous 12-month period in a house operated under simulated occupancy conditions.

2.1. Building envelope

The research house has a total conditioned space of 253 m², a conventional attic, and a crawl space. It is built with a double-walled framed assembly with interior insulation and phase change material (PCM) in the exterior wall. The thermal energy exchange between the house and its environs is modulated by the PCM. The house roof has an infrared reflective (IRR) painted metal shake with a solar reflectance of 0.34 and thermal emittance of 0.85. Attic ventilation is aided with solar-powered gable ventilators. The attic floor consists of oriented stranded board (OSB) deck overlaid with micro-perforated aluminum foil facing the attic roof, serving as a radiant barrier. At solar noon with clear skies, the gable ventilators expel air through the soffit vents to the outside at the rate of 10 air exchanges per hour. Attic insulation consists of 0.25m regular cellulose with 0.1m of 20 % by weight PCM-enhanced cellulose. The PCM material, dispersed in the blown fiber insulation on the attic floor hypothetically absorbs residual heat that has not been vented or radiated out. The R-value of the wall was $R_{SI} - 5.3$ ($R_{US} - 30$) and of the attic floor, $R_{SI} - 8.8$ ($R_{US} - 50$). Glazed, triple pane windows had a U-factor of $1.25 \text{ W} \cdot \text{m}^{-2} \cdot \text{K}$ and a SHGC of 0.17. Additional details of the building envelope may be found elsewhere [23].

The residential house has a Home Energy Rating System (HERS) Index of 51. In comparison, a Net Zero Energy home and an Environmental Protection Agency (EPA) Energy Star home would

1 have a HERS Indexes of 0, and 85, respectively. A blower door test yielded 2.2 air changes per
2 hour (ACH) at 50 Pa pressure difference. A house built to meet the IECC 2006 code would have
3 higher leakages of 5.7 ACH at 50Pa and a HERS Index of 93[23].

4
5 The house is demarcated into two space conditioning zones, upstairs and downstairs. The
6 ambient temperatures for the entire house was set at 24.4°C (76°F) for space cooling, and 21.7°C
7 (71°F) for space heating with a thermostat whose accuracy was $\pm 1^\circ\text{C}$. The relative humidity
8 varied between 40% and 50% throughout the year, well within the comfort region as defined by
9 ASHRAE [24].

10 11 *2.2 Vertical bore*

12 The vertical-bore consisted of one, 1.9 cm ID high density polyethylene (HDPE) tube bent into a
13 U-shape at the bottom of a 94.5 m vertical bore. The “brine” contained a water-propylene glycol
14 mixture of approximately 80%/20% vol. /vol. A separate 5.3 kW water-to-water ground source
15 heat pump connected to the same vertical-bore ground loop serviced a 303L (85 gal.) water tank
16 that provided approximately $220 \text{ L} \cdot \text{day}^{-1}$ hot water at 49°C (120°F) for domestic use.

17 18 *2.3. Space conditioning equipment*

19 The space conditioning equipment was a two-capacity, WA-GSHP with a rated capacity in the
20 cooling mode of 7.79 kW (full load) and 6.23 kW (part load) with Energy Efficiency Rating
21 (EER) of 5.4 and 7.6, respectively. Heating mode capacities were 5.80 kW (full load) and 4.84
22 kW (part load) with COPs of 4.0 and 4.6, respectively. Cooling capacity is based upon 27°C dry
23 bulb (DB), 19°C wet bulb (WB) entering air, and EWT of 25°C (full load), and the same entering

air conditions but 20°C at part load. The heating mode capacity is based upon 20°C DB, 15°C WB entering air temperature, and EWT of 0°C (full load) and 5°C (part load) . The full load corresponds to 100% compressor displacement, whereas part load is at 67%, attained by bypassing a portion of the refrigerant gas back to the low side of the compressor by ports controlled by solenoid action [25]. In reality, the compressor runs at a single stage, but mimics a two-stage system by by-passing the refrigerant to the low side. This was a design feature implemented by the OEM to closely approximate the two stage option at lower cost compared to variable speed compressors. The two-stage compressors are widely sold for HVAC systems and are not one-of-a-kind models. Note that the OEM calls these compressors two-stage, and not two-speed as would be the case if a drive mechanism was used to achieve the two speeds.

The blower motor is a 373 W electronically commutated motor (ECM). At the ISO rating points the blower draws are (a) in cooling mode: full-load 183 W; part-load 140 W, and (b) in heating mode: full-load 246 W, part-load 190 W. The brine pump is a 3-speed (low, medium, high) cast iron pump set on medium speed.

2.4. Data collection

Brine flow rates in the ground loop were monitored by measuring the differential pressure across the refrigerant-to-water heat exchanger, calibrated by the manufacturer within $\pm 0.25\%$ full scale (FS) accuracy. Compressor and pump power were measured with instrumentation-grade watt/watt-hour transducers, with split-core current transformers (CTs) having a minimum accuracy of $\pm 0.45\%$ of reading plus 0.05% of FS. Thermistors were surface- mounted by taping them to the specific location, wrapping in cork insulation, and then taping once again to insulate

1 against the environment. Surface- mounted thermistors used had a 15s time constant and $\pm 0.2^{\circ}\text{C}$
2 tolerance. Compressor suction and discharge pressures were measured with pressure transducers
3 measuring gauge pressure in the ranges 0-1.586 MPa gauge (0-230psig) and 0-5.171 MPa gauge
4 (0-750 psig), respectively, with $\pm 0.25\%$ FS accuracy. The standard errors of the calculated COP
5 in the heating and cooling modes are at most ± 0.0172 and ± 0.0204 , respectively.

6
7 The pressure drops in connecting lines between various components were taken as 1 – 1.5% of
8 the upstream pressure, based on the original equipment manufacturers (OEMs) data. A
9 schematic of the WA-GSHP is shown in Figure 1 with the control volume (CV) for mass and
10 energy balances, and Exergy analysis.

11
12 Data are recorded every 30 s and averaged over 15 min for subsequent analysis. To capture the
13 true performance, key variables are time-averaged over the interval when the GSHP is running
14 and are reported for each month. Some months have both heating and cooling days and these are
15 segregated in the respective tables. Thermodynamic irreversibility analysis identifies sources of
16 inefficiency where systemic improvements may be made.

17
18 Balancing accuracy and reliability imposed some experimental limitations. Although *in situ*
19 measurements would likely be more accurate than *ex situ* measurements, the former requires
20 penetrations that are prone to leaks whereas the latter may yield slightly less accurate readings
21 but are free from leaks. Therefore, apart from two *in situ* pressure measurements (transducers)
22 in the refrigerant lines, temperature measurement were all *ex situ* with surface-mounted
23 thermistors, except for entering and leaving brine temperatures which were *in situ*. Analysis is

based on quasi steady-state conditions in the refrigerant loop justified on the basis that small fluctuations in state variables are negligible. The state points (Figure 1) refer to the refrigerant and brine lines. The expansion valve is assumed to be adiabatic. Apart from the compressor whose suction and discharge pressures are measured *in situ* with pressure transducers, the pressure drop across each component is according to information supplied by the manufacturer. Thermodynamic state points are calculated from measured intensive variables from National Institute of Standards and Technology (NIST) Reference Fluid Thermodynamic and Transport Properties Database (REFPROP) Version 8.0 [26] as a Microsoft Excel add-in program.

3. Analysis

Mass and energy conservation equations under the assumptions of quasi steady-state steady flow (QSSSF) conditions applied to the control volume shown in Figure 1 are fully justified on the basis that except during compressor and indoor blower fan shut-down and start-up, or when the intra cycle times are less than 5 minutes (rare occurrences), QSSSF conditions prevail. Since the size of the WA-GSHP is only 7.56 kW for the 253 m² house, the compressor run times are relatively long, an important factor in keeping cyclic losses low. The term QSSSF instead of steady-state steady flow (SSSF) is used because the latter implies no fluctuations with time, whereas the former is tolerant of small variations (fluctuations) in process variables that cancel out over the time scale of a typical cycle, thereby approaching SSSF behavior.

The general mass balance applied between any two state points in Figure 1 is given by, Eq. (1)

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

and the general energy balance applied between any two arbitrary state points is expressed [27]

1 as,

$$\dot{Q}_o + \sum_j \dot{Q}_j + \dot{W} + \sum_{in} (h + \frac{V^2}{2} + gz) \times \dot{m} - \sum_{out} (h + \frac{V^2}{2} + gz) \times \dot{m} = \frac{dE_{cv}}{dt} \quad (2)$$

3 where $\dot{Q}_o$ is the heat exchanged with the surrounding at T_o and the quantities $\dot{Q}_j$ with j
4 additional thermal reservoirs at temperatures $T_1, T_2, T_3, \dots, T_j$, respectively; $\dot{W}$ is the rate of
5 work input; $h, V^2/2, gz$ represent enthalpy, kinetic, and potential energy terms, respectively; and
6 $\dot{m}$ is the mass flow rate into or out of the control volume. The term dE_{cv}/dt represents the rate
7 of accumulation of energy within the control volume (as demarcated in Figure 1); $E_{cv} = U +$
8 $(1/2) V^2 + gz$; and both $\dot{Q}_o$ and $\dot{W}$ are defined as positive for heat and work *inputs* to the
9 control volume. On a monthly basis, assuming QSSSF, the energy terms from Eq. (2) in
10 component form reduce to,

$$W_{Comp.} + W_{Fan+Controls} + W_{Pump} + W_{e,Aux.} + Q_{Indoor\ Coil} + Q_{GLHX} + Q_{Comp.} + Q_{e,Aux.} = 0 \quad (3)$$

13
14 On the basis of Eq. (3), the monthly aggregated component-wise energy inputs and outputs are
15 shown in Table 1 and Table 2, respectively, with agreement between 2.4-3.9%.

16
17 One of the benefits and a strong argument in favor of GSHPs for residential space heating is the
18 high percentage ($> 75\%$) of the total energy needed for space heating that is extracted from the
19 ground, shown explicitly in Table 3. Unlike an air-source heat pump, the GSHP operates
20 favorably with respect to its heat sources and sinks. In the heating season, the GSHP extracts
21 energy from the ground (source), which is warmer than the ambient air, and in the cooling
22 season, the GSHP rejects thermal energy to the ground (sink), which is cooler than the ambient

air. In this respect, the GSHP is in harmony with its environment. In contrast, the air source heat pump extracts energy from the colder air (source) during winter time, and rejects heat to warm air (sink) during the cooling season. The air source heat pump thus operates unfavorably against the environment. This is the reason why intrinsically, the GSHPs are more efficient than its counterpart, the air source heat pumps. The measured OAT and EWT temperature profiles are shown in Figure 2. Clearly, we want the EWT to be warmer than the OAT in winter.

The WA-GSHP under development operated nearly exclusively in the low-stage, year round, further conserving electricity use. The measured monthly electricity consumption, compressor run times, and the percent time the GSHP operated in the low and high stages and entering and leaving water (brine) temperatures are shown in Table 4 along with a plot of the electricity use profile shown in Figure 3. At \$0.11/kWh, the average monthly cost of space conditioning is only \$40/month.

Having shown the goodness in energy balances, and the low electricity use of the vertical-bore WA-GSHP, we now discuss the sources of systemic inefficiency.

4. Exergy Analysis

The general entropy balance equation applied to various state points within the control volume is given by [27],

$$\dot{\sigma}_{Total} = \sum_{out} \dot{m} \times s - \sum_{in} \dot{m} \times s - \frac{\dot{Q}_o}{T_o} - \sum_j \frac{\dot{Q}_j}{T_j} \geq 0 \quad (4)$$

where $\dot{\sigma}_{Total}$ is the sum of the entropy generation rate due to internal irreversibility and those due to irreversible transfer of thermal energy between the respective CV and its surroundings.

The *rate* of availability destruction $\dot{I}$ is given by $\dot{I} = T_o \dot{\sigma}$ or by $\dot{I} = T_j \dot{\sigma}$ depending on whether the CV boundary is at T_o or at T_j , respectively. In applying Eq. (2) and Eq. (4) to each segment in Figure 1 in piecemeal fashion, the final and initial state points depend on whether the heat pump is run in the heating or in the cooling mode, except in the case of the compressor, whose suction (state point 1) and discharge (state point 2) ports remain the same, irrespective of the operational mode. Thus, the initial and final state points for the indoor coil in the heating mode are 3 and 4, respectively, whereas in the cooling mode, they are 4 and 3, respectively. For the connecting line between the indoor coil and the compressor, the initial and final points are 3 and 1 in the cooling mode, and 2 and 3 in the heating mode, due to flow reversal in the reversing valve in switching modes. Similar considerations apply to the state points of other components when applying Eqs. (2) and (4).

All segments of the WA-GSHP are taken to exchange thermal energy with the surroundings at the indoor temperature set point, T_o except for the refrigerant line segment 5-6 in the ground loop heat exchanger which exchanges thermal energy with the temperature of the ground loop taken to be the average of the LWT and EWT in segment 7-8.

Starting with the compressor whose control volume extends to the surroundings and noting that no other thermal reservoir except for the indoor air (surroundings) is in contact with this control

volume implies that $\dot{Q}_j = 0$ but $\dot{Q}_o \neq 0$. With these stipulations and the assumption of QSSSF, Eqs. (1) and (2) and (4) applied to the compressor yields,

$$\dot{Q}_o + \dot{W}_{Comp.} + \dot{m} \times (h_{suction} - h_{discharge}) = 0 \quad (5)$$

$$\dot{\sigma}_{Comp.} = \dot{m} \times (s_{discharge} - s_{suction}) - \frac{\dot{Q}_o}{T_o} \quad (6)$$

$$\dot{I}_{Comp.} = T_o \dot{\sigma}_{Comp.} \quad (7)$$

The rates of compressor heat rejection, $\dot{Q}_o$, entropy production, $\dot{\sigma}_{Comp.}$, and irreversibility, $\dot{I}_{Comp.}$, are calculated from Eqs. (5) — (7) in conjunction with data tabulated in Table 5.

The discharge of thermal energy from the refrigerant to the brine loop occurs at the temperature of the brine, implying that $\dot{Q}_o = 0$, but that $\dot{Q}_j \neq 0$.

With the exception of the compressor whose suction and discharge are invariant with the cooling or heating cycle, the initial and final states of the refrigerant in other components change depending on the direction of refrigerant flow. For example, in the heating mode, the flow of refrigerant from the GLHX to the compressor is along the segment $6 \rightarrow 1$, whereas in the cooling cycle it is from $2 \rightarrow 6$. The initial and final states of the refrigerant in this particular segment are categorized as: initial state point (6: heating; 2: cooling) and final state point (1: heating; 6: cooling). Table 5 is constructed with these stipulations in mind where Eqs. (1), (2), and (4) are repetitively applied to each segment of the WA-GSHP yielding the component-wise monthly

entropy production and irreversibility given by the product of $\dot{I}$ and run times, as tabulated in Table 6 and Table 7 and their relative magnitudes shown in Figure 4. From an energy conservation point of view, the Lost Work is more important than the irreversibility rate because it represents the magnitude of systemic inefficiency. Figure 4 shows graphically that the compressor and the indoor coil are the two major contributors to systemic inefficiency in terms of irreversibility rates, and total irreversibility (Lost Work).

Further validity of the Exergy analysis is gleaned by showing that the total rate of electrical work ($\dot{W}_{Comp. meas.}$) supplied to the compressor must equal the work input rate of an ideal compressor (Carnot engine power), plus the rate of irreversibility (or rate of Lost Work) due to irreversibility within the compressor control volume. The Carnot engine power is calculated on the basis of a 2-T Carnot engine operating between the temperature of the brine loop and the surrounding temperature. If the irreversibility calculations are correct, then Eq. (8) must be satisfied within reasonable experimental error.

$$\dot{W}_{Comp. meas.} = \dot{W}_{Carnot} + \dot{\sigma}_{Total} T_o = \dot{W}_{Carnot} + \dot{I} \quad (8)$$

The experimentally measured quantities in Eq. (8) are T_o and $\dot{W}_{Comp. meas.}$. The variable on the right-hand side, $\dot{W}_{Carnot}$ is quantified by the Eq. (9)

$$\dot{W}_{Carnot} = \frac{\dot{Q}_o (Indoor HX) + \dot{Q}_{Comp.}}{COP_{Carnot}} \quad (9)$$

where $\dot{Q}_o (Indoor HX)$ is <0 when heat is rejected to the conditioned space (heating season), and is >0 when heat is absorbed from conditioned space (cooling season) as shown clearly in Table 5 for the Indoor HX. If the irreversibility calculations are correct, then Eq. (8) should be satisfied

within an acceptable margin of experimental error. The agreement is measured by the relative percent difference, $\Delta\% = [(\dot{W}_{Comp.} - \dot{W}_{Comp.meas.}) / \dot{W}_{Comp.}] \times 100$. The Carnot work as a fraction of the actual (measured) work input is given by the ratio, $\eta = (\dot{W}_{Carnot} / \dot{W}_{Comp.meas.})$, tabulated in Table 8. The Carnot COPs for the heating and cooling periods required to evaluate the first term on the right-hand side of Eq. (8) are calculated as follows:

$$\text{Heating mode: } COP_{Carnot} = T_o / (T_o - T_{EWT}) \quad (10)$$

$$\text{Cooling mode: } COP_{Carnot} = T_o / (T_{EWT} - T_o) \quad (11)$$

Agreement between the actual (measured) compressor power, $\dot{W}_{Comp.meas.}$ and that computed from Eq. (8) for each month validates the irreversibility calculations. Attention is drawn on some anomalous cases in Table 8 where the Carnot COP is negative. This peculiarity is because when the house set point temperature, T_o is greater than T_{EWT} , the traditional definition of COP, as given by Eq. (11) does not apply in the normal sense, and the Carnot work to drive heat from a warmer temperature (T_o) to a cooler environment (T_{EWT}) is spontaneous, not requiring any work input. Looking at it in another way, a *de facto* cooling is taking place in the direction of the natural flow of heat from the higher indoor air temperature, T_o , to the cooler brine at T_{EWT} . In fact, the natural flow of heat is such that useful work may be extracted rather than needed to cool the conditioned space. As a case in point, take the April data in Table 8. The Carnot COP as given by invoking Eq. (11) is -69.85, and the Carnot work given by Eq. (9) is $[5853.6 / -69.85] = -84.4$ W, where $\dot{Q}_{Indoor HX} + \dot{Q}_{Comp.} = 5853.6$ W from the data in Table 5. The actual compressor work by Eq. (8) is, $[-84.4 + 871] = 786$ W, within 0.61% of the measured compressor work of 782 W. The goodness of agreement, $\Delta\%$, for all 12 months is tabulated in Table 8 demonstrating that

Eq. (8) is satisfied within experimental error. Similar cases for a horizontal ground loop have been addressed extensively by Ally et al. [8].

5. Performance metrics

The First-Law efficiency of the WA-GSHP is defined by its COP which, according to ANSI/ARI/ASHRAE/ISO Standard 13256-1 [28] must include the compressor, the air handler, the water pump, and all other control as the “effective power input”. The flow resistance in the duct work, the brine loop, and power consumed by the electronic controls are reflected in our analysis. With these stipulations, the heating COP becomes,

$$COP_h = \frac{\text{Total Energy Delivered}}{\text{Total Energy Input}} \quad (12)$$

$$= \frac{(\dot{Q}_{Brine\ HX} + \dot{Q}_{Fan+Controls} + \dot{W}_{Comp.} + \dot{W}_{e,Aux.})}{(\dot{W}_{Comp.} + \dot{W}_{Fan+Controls} + \dot{W}_{Pump.} + \dot{W}_{e,Aux.})}$$

where, $\dot{Q}_{Brine\ HX}$ is obtained directly from Table 5, and Fan+Controls, compressor, and auxiliary electrical energy given in Table 9 must be multiplied by the run time in each month, respectively, and divided by 1000 to convert to kWh. Distinct from the ANSI/ARI/ASHRAE/ISO Standard 13256-1 [28] rating, most manufacturers do not include pump and fan power in their manufacturer’s data sheet because duct and loop losses vary from one installation to the other. To quantify the impact of pump, fan, controls, and any other “effective power inputs” as per ANSI/ARI/ASHRAE/ISO Standards on COP, we compute two additional COPs where pump energy, and pump plus fan and controls energy are excluded from Eq. 12.

The results are tabulated in Table 9. Measurements from our field tests show that the COP penalty from including the pump, and pump plus fan and controls is about 11% and 25%,

respectively. Generally, COP measurements in actual field tests may be lower than rated values from the manufacturer because duct and pumping losses are not known precisely, *a priori*.

Applying Eqs. (1) and (2) to the cooling cycle, the rate of cooling the conditioned space given by:

$$\dot{Q}_{air} = -(\dot{Q}_{Comp.} + \dot{W}_{Comp.} + \dot{Q}_{Brine\ HX} + \dot{W}_{fan+controls}) \quad (13),$$

and the cooling cycle coefficient of performance, COP_c , is given by

$$COP_c = \frac{\text{Total Rate of Energy Removed from Conditioned Space}}{\text{Total Rate of Energy Input}} \quad (14)$$

$$= \frac{\dot{Q}_{air}}{(\dot{W}_{Comp.} + \dot{W}_{fan+controls} + \dot{W}_{Brine\ Pump})}$$

where, once again, the terms in the numerator and denominator are multiplied by the run time and divided by 1000 to convert to kWh. As for the heating cycle, we calculate the COPs when pump, and pump plus fan and controls are excluded. The cooling cycle results are displayed in Table 10. In the cooling season for months with significant run times, the penalty on COP due to the pump and pump plus fan and controls can vary from 13% to 25%, respectively. Table 5 through Table 10 contain a comprehensive set of data and performance metrics taken over a continuous 12-month period for a vertical-bore ground source heat pump that may be benchmarked in comparative studies.

6. Discussion

This 12-month field study unequivocally demonstrates that the entire thermal load of a 253 m² PCM house with $R_{SI} = 5.3$ ($R_{US} = 30$) walls, $R_{SI} = 8.8$ ($R_{US} = 50$) attic floor, glazed, triple pane

1 windows with a U-factor of $1.25 \text{ W} \cdot \text{m}^{-2} \cdot \text{K}$ and a SHGC of 0.17, low leakage rates of 2.2
2 ACH at 50Pa, and a HERS Index of 51 was fully met with a 7.56 kW WA-GSHP consuming
3 negligible auxiliary heat during the heating season. The highest and lowest COPs in the cooling
4 season were 5.16 and 3.72 in April and October, respectively, although in these months the total
5 run times were lower than during peak summer season. Discounting the shoulder months when
6 both heating and cooling loads are low, the cooling COPs were still mostly greater than 4.
7 During the heating season, the highest and lowest COPs were 3.75 and 3.49 in December and
8 March, respectively. These COPs are calculated on the basis of total power input including full
9 power from pump, fan, controls, exceeding the ASHRAE/ISO 13256-1 Standards which require
10 that only a portion of the pump and fan power sufficient to overcome the internal resistance shall
11 be included in the effective power input. Delivered energy and pump power are partly dependent
12 on brine pressure which we had to adjust periodically. This was part of the learning process in
13 getting the loop to operate as best as we could. Such adjustments influences the energy drawn
14 from the brine loop, the pump and fan power which in turn influences the overall COP. These
15 nuances including a problem with a zone damper explain the slightly lower heating COP (3.49)
16 in March compared to that in December even though the EWTs are close in value. We quantify
17 the differences in COP that can emerge if the power from the pump, fan, and controls are
18 discounted which can be as much as 30%. Remarkably, most of the time the compressor operated
19 in the low stage (Table 9 and Table 10) while still meeting loads, and strikingly, virtually no
20 auxiliary heat was needed during the heating season (Table 2 and Table 9) except for a negligible
21 spurious reading of 2.794 W in October. Exergy analysis provides deep insight in identifying
22 sources of systemic inefficiency which the First Law cannot. The main source of systemic
23 inefficiency is the compressor, followed by the indoor HX, followed by the brine GLHX as

shown in Figure 4. Without Exergy analysis it would not be possible to identify sources of irreversibility. Although the rate of irreversibility $\dot{I}_{Total}$ is fairly constant in each month (Table 6), the total irreversibility, or $I = \dot{I}_{Total} \times \text{run time}$ (Figure 4 and Table 7) is highest during the peak cooling and peak heating seasons (due to longer run times). The rates of irreversibility can be ordered as follows: compressor>indoor HX>brine HX,>expansion valve>line connections. Notably, the irreversibility across the expansion valve was not as great as first anticipated, probably due to the lower pressure drop across it with R410A refrigerant as would be the case with other refrigerants. The research findings suggest that the WA-GSHPs are a practical and viable technology to reduce primary energy consumption and greenhouse gas emissions under the IECC 2012 Standard, as well as the European Union (EU) 2020 targets of using renewable energy resources.

7. Conclusions

Measured data from a 12-month field test located in Oak Ridge, Tennessee, 36.01° N 84.26°W in a mixed-humid climate with HDD of 2218°C-days (3993°F-days) and CDD of 723°C-days (1301°F-days) clearly showed that all the space conditioning loads of a 253 m² house with a HERS rating of 51 could be met with a 7.56 kW ground source heat pump, virtually eliminating the need for any auxiliary heat (Table 2). More than 75% of the entire heating load was extracted from the ground (Table 3) proving that the ground's resource of thermal energy can be extracted for a large portion of space heating load. Although the irreversibility rate did not show much variation from month to month (Table 6), the total irreversibility, or Lost Work which depends on run times, was highest during the peak cooling and heating seasons when run times are high (Figure 4). Exergy analysis identifies sources of systemic inefficiency. The rates of

irreversibility is in the order, compressor>indoor HX>brine HX,>expansion valve>line connections. The sum total of total Lost Work is the opportunity for improving the technology. Irreversibility analysis was verified by showing that the measured electrical input to the compressor is the sum of the Carnot work and the Lost Work as dictated by Eq. (8), within reasonable experimental error of $\pm 3\%$ (Table 8). Performance metrics indicated that: (1) the highest and lowest COPs during the heating season were 3.75 and 3.49, respectively and (2) highest and lowest COPs in the cooling season were 5.16 and 3.72, respectively. COPs are calculated on the basis of total power input including full power from pump, fan, and controls. The penalty of pump, fan, and control board power consumptions on COP shown in Table 9 and Table 10 can vary from 11%-30%. The data and performance metrics show that although 75% of the heating load was extracted from the ground, there is still room for technical improvements based on irreversibility analysis by eliminating Lost Work within technical and economic limitations. The research findings suggest that the WA-GSHPs are a practical and viable technology to reduce primary energy consumption and greenhouse gas emissions under the IECC 2012 Standard, as well as the European Union (EU) 2020 targets of using renewable energy resources.

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3 **Figure 1. Schematic of the WA-GSHP connected to the 95.4 m vertical-bore ground loop.**
 4 **The numbered items refer to state points, and the dashed boundary refers to the system**
 5 **boundary.**

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3 **Figure 2 Measured Outside Air and Entering Water Temperature profiles.**

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Figure 3. Monthly electricity use (kWh)

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Figure 4. Monthly averaged irreversibility rates and Lost Work for each component of the WA-GSHP.

Table 1: Aggregated monthly energy inputs for each component and yearly totals.

(yr. 2012)	$W_{Comp.}$ (kWh)	$W_{Fan+Controls}$ (kWh)	$W_{Brine Pump}$ (kWh)	$Q_{Brine HX}$ (kWh)	Q_{Indoor} (kWh)	$W_{e,Aux.}$ (kWh)	E_{In} (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
Apr	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
Jul	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		0.095415	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: Aggregated monthly energy outputs for each component and yearly totals.

(yr. 2012)	$Q_{Indoor\ HX}$	$W_{Fan+Controls}$	$Q_{Comp.}$	$Q_{Aux.}$	$Q_{Brine\ HX}$	E_{Out}
	(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		-1073.8
Mar (heat)	-173.7	-10.7	-11.0	0.0000		-195.4
Mar (cool)		-17.0	-9.4		-396.7	-423.1
Apr		-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 obtained from the ground (renewable resource).

(yr. 2012)	Total Energy Delivered (kWht)	Total Electrical Energy Used $W_{e,Total}$ (kWhe)	Energy Extracted from Ground Loop $Q_{Brine\ HX}$ (kWht)	$\frac{W_{e,Total}}{Q_{Brine\ HX}}$	% Delivered Energy from Ground Loop
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
Mar (cool)					
Apr					
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

Table 4: Electricity use profile, compressor run times, and percent operation in low and high stage.

	Electrical Energy Use	Run Time	% Run Time High	% Run Time Low	EWT (°C)	LWT (°C)
	(kWh)	(hrs)				
Jan	429.1	302.1	1.4	97.7	7.50	5.00
Feb	293.2	204.6	1.3	97	8.17	5.56
Mar (heat)	56.4	38.6	0	99.1	9.00	6.52
Mar (cool)	78.4	65.8	5.6	93.4	19.11	23.58
Apr	59.1	52.8	5.2	94.8	19.50	24.39
May	263.1	218.1	0	99.5	23.56	28.30
Jun	373.4	288.5	0	100	26.39	31.38
Jul	598.4	429.7	4.1	95.9	30.11	35.01
Aug	407.9	310.2	1.7	98.3	28.06	32.86
Sept	224.7	175.8	2.7	97.3	26.11	31.28
Oct (heat)	47.6	34.2	2.2	96.1	22.22	26.63
Oct (cool)	18.7	15.1	0	98.9	12.65	9.89
Nov	293.6	189.5	0	98.8	10.71	7.95
Dec	361.7	259.6	0	99.2	8.17	5.57

Table 5: Energy flows, entropy, and irreversibility rates calculated from averaged monthly measured thermodynamic properties at respective state points for the WA-GSHP.

Compressor															
	Discharge				Suction				$\dot{Q}_o$ (W)	$\dot{W}$ (W)	$\dot{m}$ (kg/s)	$\dot{\sigma}$ (W/K)	$\dot{I}$ (W)	T_o (K)	
	T	P	h	s	T	P	h	s							
	(°C)	(kPa)	(kJ/kg)	kJ/kg K	(°C)	(kPa)	(kJ/kg)	kJ/kg K							
Jan	63.9	2031	323.2	1.08983	7.3	711	286.8	1.06365	-211.7	1052.2	0.0231	1.3226	389.9	294.9	
Feb	63.2	2038	322.2	1.08668	7.8	718	286.9	1.06319	-237.1	1060.4	0.0233	1.3521	398.6	294.8	
Mar (heat)	58.2	2042	305.8	1.03159	8.4	753	274.0	1.01174	-283.7	1019.6	0.0231	1.4218	419.0	294.8	
Mar (cool)	45.0	1621	297.2	1.02455	14.8	920	275.9	1.00169	-142.3	773.9	0.0296	1.1566	340.8	297.0	
Apr	48.6	1650	454.2	1.84775	15.8	934	432.4	1.82556	-103.4	781.5	0.0311	1.0389	306.2	296.9	
May	53.6	1837	456.7	1.84613	16.1	952	432.2	1.82341	-98.1	870.2	0.0316	1.0485	309.1	296.9	
Jun	57.1	1979	458.3	1.84487	16.7	966	432.3	1.82222	-116.5	955.7	0.0322	1.1220	330.8	297.0	
Jul	57.1	1979	460.8	1.84420	16.7	966	432.3	1.82078	-122.8	1053.8	0.0327	1.1799	347.9	297.1	
Aug	61.9	2170	459.1	1.84349	17.2	986	432.5	1.82150	-117.1	977.7	0.0324	1.1063	326.2	297.0	
Sept	59.0	2062	457.7	1.84331	17.2	984	432.7	1.82241	-129.0	939.6	0.0324	1.1118	327.8	297.0	
Oct (heat)	56.4	1969	428.5	1.71464	17.5	980	399.6	1.69816	-329.5	1049.2	0.0250	1.5286	450.5	294.8	
Oct (cool)	56.7	2142	409.1	1.65758	11.5	832	389.0	1.63912	-225.4	808.9	0.0291	1.2960	381.9	297.0	
Nov	44.4	1778	463.6	1.85630	16.8	980	429.9	1.83305	-207.3	1051.1	0.0251	1.2859	379.1	294.9	
Dec	63.0	2075	465.5	1.86430	9.9	776	429.3	1.83684	-189.5	1033.7	0.0233	1.2834	378.3	294.9	

Brine HX to Compressor Line															
	Final state point (1:heating; 6:cooling)				Initial state point (6:heating; 2:cooling)				$\dot{Q}_o$ (W)	$\dot{W}$ (W)	$\dot{m}$ (kg/s)	$\dot{\sigma}$ (W/K)	$\dot{I}$ (W)	T_o (K)	
	T	P	h	s	T	P	h	s							
	(°C)	(kPa)	(kJ/kg)	kJ/kg K	(°C)	(kPa)	(kJ/kg)	kJ/kg K							
Jan	7.3	710.8	286.8	1.0637	4.8	710.9	284.4	1.0548	56.2	0.0	0.0231	0.0129	3.8	294.9	
Feb	7.8	718.3	286.9	1.0632	5.2	718.7	284.6	1.0546	55.1	0.0	0.0233	0.0124	3.7	294.8	
Mar (heat)	8.4	752.9	274.0	1.0117	5.3	715.0	271.9	1.0041	48.9	0.0	0.0231	0.0103	3.0	294.8	
Mar (cool)	43.9	1544.4	295.9	1.0207	45.0	1621.1	297.2	1.0246	-37.2	0.0	0.0296	0.0117	3.5	297.0	
Apr	47.5	1648.7	452.9	1.8440	48.6	1650.3	454.2	1.8478	-38.9	0.0	0.0311	0.0126	3.7	296.9	
May	52.4	1835.5	455.3	1.8420	53.6	1837.1	456.7	1.8461	-43.9	0.0	0.0316	0.0161	4.8	296.9	
Jun	55.9	1976.1	456.9	1.8406	57.1	1978.6	458.3	1.8449	-46.4	0.0	0.0322	0.0184	5.5	297.0	
Jul	60.6	2168.7	458.0	1.8346	61.9	2170.5	460.8	1.8442	-92.5	0.0	0.0327	0.0000	0.0	297.1	
Aug	57.8	2060.8	457.6	1.8391	59.0	2062.0	459.1	1.8435	-47.9	0.0	0.0324	0.0196	5.8	297.0	
Sept	55.3	1967.0	456.3	1.8391	56.4	1968.8	457.7	1.8433	-45.7	0.0	0.0324	0.0178	5.3	297.0	
Oct (heat)	11.5	832.5	399.6	1.6982	7.8	768.8	397.8	1.6915	46.4	0.0	0.0250	0.0082	2.4	294.8	
Oct (cool)	43.4	1597.3	407.9	1.6540	44.4	1777.7	409.1	1.6576	-34.1	0.0	0.0291	0.0118	3.5	297.0	
Nov	9.9	776.1	429.9	1.8330	7.6	775.6	427.8	1.8252	54.8	0.0	0.0251	0.0110	3.2	294.9	
Dec	7.8	723.3	429.3	1.8368	5.4	723.2	426.9	1.8282	55.8	0.0	0.0233	0.0125	3.7	294.9	

Brine HX

	Final state point (6:heating; 5:cooling)				Initial state point (5:heating; 6 cooling)				$\dot{Q}_j$ (W)	$\dot{W}$ (W)	$\dot{m}$ (kg/s)	$\dot{\sigma}$ (W/K)	$\dot{I}$ (W)	T_j (K)
	T	P	h	s	T	P	h	s						
	(°C)	(kPa)	(kJ/kg)	kJ/kg K	(°C)	(kPa)	(kJ/kg)	kJ/kg K						
Jan	4.8	710.9	284.4	1.0548	0.9	724.9	111.8	0.4236	3986.8	0.0	0.0231	0.3770	105.8	280.7
Feb	5.2	718.7	284.6	1.0546	1.2	732.7	112.1	0.4246	4024.7	0.0	0.0233	0.3978	111.9	281.3
Mar (heat)	5.3	715.0	271.9	1.0041	1.4	728.4	107.4	0.4058	3807.0	0.0	0.0231	0.3560	100.4	282.1
Mar (cool)	22.2	1528.9	92.3	0.3436	43.9	1544.4	295.9	1.0207	-6033.7	0.0	0.0296	0.5806	169.7	292.3
Apr	24.8	1632.3	239.8	1.1366	47.5	1648.7	452.9	1.8440	-6633.1	0.0	0.0311	0.6510	190.5	292.7
May	28.7	1817.2	246.4	1.1579	52.4	1835.5	455.3	1.8420	-6603.2	0.0	0.0316	0.6318	187.5	296.7
Jun	31.5	1956.3	251.3	1.1735	55.9	1976.1	456.9	1.8406	-6623.2	0.0	0.0322	0.6245	187.1	299.5
Jul	35.2	2147.1	257.7	1.1938	60.6	2168.7	458.0	1.8346	-6556.7	0.0	0.0327	0.6456	195.8	303.3
Aug	33.3	2040.2	254.2	1.1830	57.8	2060.8	457.6	1.8391	-6584.2	0.0	0.0324	0.6198	186.7	301.2
Sept	31.4	1947.3	251.1	1.1730	55.3	1967.0	456.3	1.8391	-6653.2	0.0	0.0324	0.6337	189.6	299.3
Oct (heat)	7.8	768.8	397.8	1.6915	3.5	781.4	241.2	1.1279	3906.4	0.0	0.0250	0.3955	113.0	285.8
Oct (cool)	23.1	1581.3	219.8	1.0355	43.4	1597.3	407.9	1.6540	-5474.0	0.0	0.0291	0.5336	157.6	295.4
Nov	7.6	775.6	427.8	1.8252	3.3	789.1	257.6	1.2084	4266.1	0.0	0.0251	0.4352	123.5	283.9
Dec	5.4	723.2	426.9	1.8282	1.3	736.7	255.7	1.2030	3997.3	0.0	0.0233	0.3867	108.8	281.3

Expansion Valve and connecting lines

	Final state point (5:heating; 4:cooling)				Initial state point (4:heating; 5:cooling)				$\dot{Q}_o$ (W)	$\dot{W}$ (W)	$\dot{m}$ (kg/s)	$\dot{\sigma}$ (W/K)	$\dot{I}$ (W)	T_o (K)
	T	P	h	s	T	P	h	s						
	(°C)	(kPa)	(kJ/kg)	kJ/kg K	(°C)	(kPa)	(kJ/kg)	kJ/kg K						
Jan	0.9	724.9	111.8	0.4236	33.2	2009.1	111.8	0.4089	0.0	0.0	0.0231	0.3414	100.6	294.9
Feb	1.2	732.7	112.1	0.4246	33.3	2015.6	112.1	0.4099	0.0	0.0	0.0233	0.3432	101.1	294.8
Mar (heat)	1.4	728.4	107.4	0.4058	31.2	1930.2	107.4	0.3998	0.0	0.0	0.0231	0.1403	41.3	294.8
Mar (cool)	7.2	894.1	92.3	0.3453	22.2	1528.9	92.3	0.3436	0.0	0.0	0.0296	0.0500	14.8	297.0
Apr	9.4	969.7	239.7	1.1409	24.8	1632.3	239.8	1.1366	0.0	0.0	0.0311	0.1326	39.4	296.9
May	10.1	992.4	246.4	1.1641	28.7	1817.2	246.4	1.1579	0.0	0.0	0.0316	0.1962	58.2	296.9
Jun	10.6	1009.2	251.2	1.1810	31.5	1956.3	251.3	1.1735	0.0	0.0	0.0322	0.2408	71.5	297.0
Jul	11.3	1031.9	257.7	1.2032	35.2	2147.1	257.7	1.1938	0.0	0.0	0.0327	0.3062	90.9	297.1
Aug	11.2	1027.7	254.2	1.1912	33.3	2040.2	254.2	1.1830	0.0	0.0	0.0324	0.2639	78.3	297.0
Sept	11.0	1022.7	251.1	1.1801	31.4	1947.3	251.1	1.1730	0.0	0.0	0.0324	0.2316	68.7	297.0
Oct (heat)	3.5	781.4	241.2	1.1279	32.0	1971.6	241.2	1.1114	0.0	0.0	0.0250	0.4104	117.2	285.8
Oct (cool)	7.8	910.1	219.8	1.0402	23.1	1581.3	219.8	1.0355	0.0	0.0	0.0291	0.1385	41.1	297.0
Nov	3.3	789.1	257.6	1.2084	34.1	2052.5	257.6	1.1879	0.0	0.0	0.0251	0.5132	151.3	294.9
Dec	1.3	736.7	255.7	1.2030	32.9	1995.5	255.7	1.1809	0.0	0.0	0.0233	0.5148	151.7	294.9

Indoor HX

	Final state point (4:heating; 3:cooling)				Initial state point (3:heating; 4:cooling)				$\dot{Q}_o$	$\dot{W}$	$\dot{m}$	$\dot{\sigma}$	$\dot{I}$	T_o
	T	P	h	s	T	P	h	s						
	(°C)	(kPa)	(kJ/kg)	kJ/kg K	(°C)	(kPa)	(kJ/kg)	kJ/kg K	(W)	(W)	(kg/s)	(W/K)	(W)	(K)
Jan	33.2	2009.1	111.8	0.4089	60.4	2029.4	319.1	1.0776	-4788.1	0.0	0.0231	0.7886	232.5	294.9
Feb	33.3	2015.6	112.1	0.4099	59.8	2036.0	318.1	1.0745	-4807.7	0.0	0.0233	0.7941	234.1	294.8
Mar (heat)	31.2	1930.2	107.4	0.3998	54.9	1949.6	301.8	1.0198	-4500.5	0.0	0.0231	0.9135	269.3	294.8
Mar (cool)	12.1	873.9	274.6	0.9970	7.2	894.1	92.3	0.3453	5400.8	0.0	0.0296	1.1255	334.2	297.0
Apr	14.6	934.8	431.1	1.8211	9.4	969.7	239.7	1.1409	5957.0	0.0	0.0311	1.1078	329.0	296.9
May	14.7	953.0	430.8	1.8185	10.1	992.4	246.4	1.1641	5830.6	0.0	0.0316	1.0441	310.0	296.9
Jun	15.1	967.0	430.8	1.8172	10.6	1009.2	251.2	1.1810	5784.4	0.0	0.0322	1.0121	300.5	297.0
Jul	15.5	986.2	430.8	1.8154	11.3	1031.9	257.7	1.2032	5667.7	0.0	0.0327	0.9609	285.5	297.1
Aug	15.7	984.8	431.1	1.8164	11.2	1027.7	254.2	1.1912	5724.3	0.0	0.0324	0.9656	286.8	297.0
Sept	15.8	981.0	431.3	1.8175	11.0	1022.7	251.1	1.1801	5843.9	0.0	0.0324	0.9918	294.6	297.0
Oct (heat)	32.0	1971.6	241.2	1.1114	54.0	1991.5	425.2	1.7048	-4590.2	0.0	0.0250	0.7634	225.1	294.8
Oct (cool)	12.5	874.5	387.9	1.6352	7.8	910.1	219.8	1.0402	4892.1	0.0	0.0291	0.8407	249.7	297.0
Nov	34.1	2052.5	257.6	1.1879	59.8	2073.3	459.7	1.8448	-5068.1	0.0	0.0251	0.7134	210.4	294.9
Dec	32.9	1995.5	255.7	1.1809	60.2	2015.6	461.3	1.8520	-4800.8	0.0	0.0233	0.6101	179.9	294.9

Indoor HX to compressor line

	Final state point (3:heating; 1:cooling)				Initial state point (2:heating; 3:cooling)				$\dot{Q}_o$	$\dot{W}$	$\dot{m}$	$\dot{\sigma}$	$\dot{I}$	T_o
	T	P	h	s	T	P	h	s						
	(°C)	(kPa)	(kJ/kg)	kJ/kg K	(°C)	(kPa)	(kJ/kg)	kJ/kg K	(W)	(W)	(kg/s)	(W/K)	(W)	(K)
Jan	60.4	2029.4	319.1	1.0776	63.9	2031.0	323.2	1.0898	-95.5	0.0	0.0231	0.0410	12.1	294.9
Feb	59.8	2036.0	318.1	1.0745	63.2	2037.7	322.2	1.0867	-95.5	0.0	0.0233	0.0404	11.9	294.8
Mar (heat)	54.9	1949.6	301.8	1.0198	58.2	2041.7	305.8	1.0316	-83.2	0.0	0.0211	0.0339	10.0	294.8
Mar (cool)	14.8	919.8	275.9	1.0017	12.1	873.9	274.6	0.9970	38.5	0.0	0.0296	0.0089	2.6	297.0
Apr	15.8	934.3	432.4	1.8256	14.6	934.8	431.1	1.8211	39.5	0.0	0.0311	0.0066	1.9	296.9
May	16.1	952.4	432.2	1.8234	14.7	953.0	430.8	1.8185	44.3	0.0	0.0316	0.0072	2.1	296.9
Jun	16.7	966.3	432.3	1.8222	15.1	967.0	430.8	1.8172	46.3	0.0	0.0322	0.0072	2.1	297.0
Jul	17.2	985.6	432.3	1.8208	15.5	986.2	430.8	1.8154	50.4	0.0	0.0327	0.0074	2.2	297.1
Aug	17.2	984.3	432.5	1.8215	15.7	984.8	431.1	1.8164	47.4	0.0	0.0324	0.0070	2.1	297.0
Sept	17.5	980.2	432.7	1.8224	15.8	981.0	431.3	1.8175	44.9	0.0	0.0324	0.0067	2.0	297.0
Oct (heat)	54.0	1991.5	425.2	1.7048	56.7	2142.4	428.5	1.7146	-82.3	0.0	0.0250	0.0345	10.2	294.8
Oct (cool)	16.8	980.3	389.0	1.6391	12.5	874.5	387.9	1.6352	32.5	0.0	0.0291	0.0051	1.5	297.0
Nov	59.8	2073.3	459.7	1.8448	63.0	2074.7	463.6	1.8563	-96.7	0.0	0.0251	0.0407	12.0	294.9
Dec	60.2	2015.6	461.3	1.8520	63.7	2017.2	465.5	1.8643	-96.6	0.0	0.0233	0.0411	12.1	294.9

Table 6: Monthly averaged irreversibility rates of the WA-GSHP by component.

	Compressor Line	Brine HX to Compressor GLHX Line	Brine GLHX	Expansion Valve & connecting lines	Indoor HX	Indoor HX to Compressor line	Total
	$\dot{i}$	$\dot{i}$	$\dot{i}$	$\dot{i}$	$\dot{i}$	$\dot{i}$	$\dot{i}$
yr. 2012	(W)	(W)	(W)	(W)	(W)	(W)	(W)
Jan	390	3.80	105.80	100.59	232.53	12.08	845
Feb	399	3.66	111.90	101.13	234.15	11.90	861
Mar (heat)	419	3.04	100.44	41.34	269.30	10.00	843
Mar (cool)	341	3.48	169.68	14.84	334.23	2.63	866
Apr	306	3.74	190.51	39.36	328.95	1.95	871
May	309	4.79	187.46	58.21	309.95	2.13	872
Jun	331	5.45	187.08	71.45	300.54	2.13	897
Jul	348	0.00	195.78	90.89	285.45	2.19	922
Aug	326	5.84	186.69	78.34	286.80	2.07	886
Sept	328	5.30	189.64	68.73	294.58	1.99	888
Oct (heat)	450	2.42	113.02	117.21	225.07	10.17	918
Oct (cool)	382	3.49	157.60	41.10	249.69	1.51	835
Nov	379	3.23	123.54	151.28	210.41	12.00	880
Dec	378	3.67	108.79	151.73	179.94	12.12	835

Table 7: Monthly averaged irreversibility (Lost Work) of the WA-GSHP by component.

	Compressor to Compressor Line	Brine HX GLHX	Brine & connecting lines	Expansion Valve & connecting HX	Indoor HX	Indoor Compressor HX	tc Total	Run Time
yr. 2012	<i>I</i> (kWh)	<i>I</i> (kWh)	<i>I</i> (kWh)	<i>I</i> (kWh)	<i>I</i> (kWh)	<i>I</i> (kWh)	<i>I</i> (kWh)	hrs
Jan	117.8	1.1	32.0	30.4	70.2	3.6	255	302.1
Feb	81.6	0.7	22.9	20.7	47.9	2.4	176	204.6
Mar (heat)	16.2	0.1	3.9	1.6	10.4	0.4	33	38.6
Mar (cool)	22.4	0.2	11.2	1.0	22.0	0.2	57	65.8
Apr	16.2	0.2	10.1	2.1	17.4	0.1	46	52.8
May	67.4	1.0	40.9	12.7	67.6	0.5	190	218.1
Jun	95.4	1.6	54.0	20.6	86.7	0.6	259	288.5
Jul	149.5	0.0	84.1	39.1	122.6	0.9	396	429.7
Aug	101.2	1.8	57.9	24.3	89.0	0.6	275	310.2
Sept	57.6	0.9	33.3	12.1	51.8	0.4	156	175.8
Oct (heat)	15.4	0.1	3.9	4.0	7.7	0.3	31	34.2
Oct (cool)	5.8	0.1	2.4	0.6	3.8	0.0	13	15.1
Nov	71.8	0.6	23.4	28.7	39.9	2.3	167	189.5
Dec	98.2	1.0	28.2	39.4	46.7	3.1	217	259.6

Table 8: Average monthly irreversibility rate and total irreversibility (Lost Work) for each component of the WA-GSHP.

	$\dot{\sigma}_{Total}$	$\dot{I}_{Total}$	Monthly Run Time	COP_{Carnot} (heating)	COP_{Carnot} (cooling)	$\dot{W}_{Carnot}$	$\dot{W}_{Carnot}$ + $\dot{I}_{Total}$	$\dot{W}_{Comp.meas.}$	% Δ	$\eta = \dot{W}_{Carnot} / \dot{W}_{Comp.meas.}$
	(W/K)	(W)	(hrs)			(W)	(W)	(W)		
Jan	2.883	845	302.07	20.73		241.1	1086	1052	3.1	0.229
Feb	2.940	861	204.63	21.79		231.5	1093	1060	3.0	0.218
Mar (heat)	2.876	843	38.60	23.29		205.4	1049	1020	2.8	0.201
Mar (cool)	2.933	866	65.75		-63.09	-83.3	782	774	1.1	0.108
Apr	2.949	871	52.77		-69.35	-84.4	786	782	0.6	0.108
May	2.944	872	218.14		-1781.23	-3.2	868	870	-0.2	0.004
Jun	3.025	897	288.47		114.94	49.3	947	956	-0.9	0.052
Jul	3.100	922	429.66		47.92	115.7	1038	1054	-1.5	0.110
Aug	2.982	886	310.23		70.76	79.2	965	978	-1.3	0.081
Sept	2.993	888	175.81		132.06	43.3	931	940	-0.9	0.046
Oct (heat)	3.141	918	34.15	32.69		150.5	1069	1049	1.8	0.143
Oct (cool)	2.826	835	15.07		-181.52	-25.7	810	809	0.1	0.032
Nov	2.999	880	189.46	26.62		198.2	1078	1051	2.5	0.189
Dec	2.849	835	259.59	21.65		230.5	1065	1034	2.9	0.223

Table 9: Heating Cycle: Monthly averaged COPs and change resulting from including fan, control, and pump power.

	Fan + Ctrl Pwr. (W)	Pump Power (W)	Run Time (hrs)	% Run Time High	% Run Time Low	Total Energy Delivered (kWh)	Auxiliary Heat (W)	Total Energy Used (kWh)	EWT (oC)	LWT (oC)	(EWT-LWT) (oC)	COP (Total)	COP w/o Pump	COP w/o Pump & Fan	Carnot COP (heating)
Jan	199.7	168.7	302.1	1.4	97.7	1582.5	0.011	429.1	7.50	5.00	2.50	3.69	4.18	4.50	20.73
Feb	202.6	169.8	204.6	1.3	97	1082.0	0.009	293.2	8.17	5.56	2.61	3.69	4.19	4.52	21.79
Mar (heat)	277.9	164.2	38.6	0	99.1	197.0	0	56.4	9.00	6.52	2.48	3.49	3.93	4.96	23.29
Mar (cool)	258.8	160.4	0.0	0	0		0								
Apr	174.5	164.5	0.0	0	0		0								
May	170.1	165.6	0.0	0	0		0	0.0							
Jun	172.2	166.6	0.0	0	0		0	0.0							
Jul	172.2	166.6	0.0	0	0		0								
Aug	170.3	166.8	0.0	0	0		0								
Sept	171.9	166.4	0.0	0	0		0								
Oct (heat)	175.8	167.1	34.2	2.2	96.1	178.0	2.794	50.3	12.65	9.89	2.77	3.54			32.69
Oct (cool)	271.3	161.8	0.0	0	0		0	0.0							
Nov	340.3	158.5	189.5	0	98.8	1071.9	0.008	293.7	10.71	7.95	2.76	3.65			26.62
Dec	193.4	166.4	259.6	0	99.2	1356.2	0.004	361.7	8.17	5.57	2.59	3.75	4.26	5.05	21.65

Table 10 : Cooling cycle: Monthly averaged COPs including the effect of pump, fan and controls.

	Fan + Control Power	Pump Power	Run Time	% Run High	% Run Low	Total Energy	Total Energy	EWT	LWT	COP	COP	COP
	(W)	(W)	(hrs)			Removed (kWh)	Used (kWh)	(°C)	(°C)		w/o Pump	w/o Pump & Fan
Jan	NA											
Feb	NA											
Mar (heat)	NA											
Mar (cool)	258.8	160.4	65.75	5.6	93.4	338.2	78.44	19.11	23.58	4.31	4.98	6.65
Apr	174.5	164.5	52.77	5.2	94.8	305.0	59.13	19.50	24.39	5.16	6.05	7.40
May	170.1	165.6	218.14	0	99.5	1234.9	263.05	23.56	28.30	4.69	5.44	6.50
Jun	172.2	166.6	288.47	0	100	1618.8	373.44	26.39	31.38	4.33	4.98	5.87
Jul	172.2	166.6	429.66	4.1	95.9	2343.1	598.37	30.11	35.01	3.92	4.45	5.17
Aug	170.3	166.8	310.23	1.7	98.3	1722.8	407.89	28.06	32.86	4.22	4.84	5.68
Sept	171.9	166.4	175.81	2.7	97.3	997.0	224.68	26.11	31.28	4.44	5.10	6.03
Oct (heat)	NA	NA	NA									
Oct (cool)	271.3	161.8	15.07	0	98.9	69.6	18.72	22.22	26.63	3.72	4.28	5.71
Nov	NA			0	100		0.00					
Dec	NA											

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